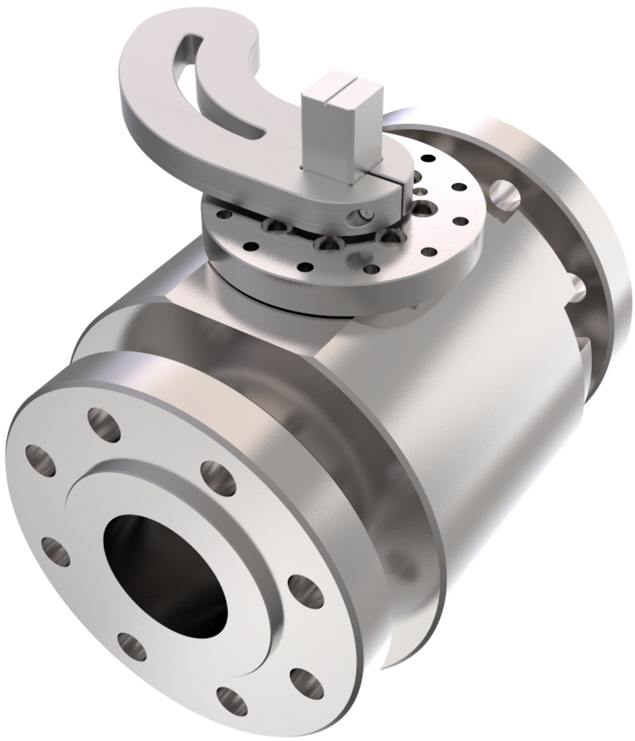
STORD/HAUGESUND UNIVERSITY COLLEGE

Resolving gasket- and bearing failure in a TBV96S valve



Bachelor thesis conducted at Stord/Haugesund university college
Mechanical engineering - Design of Marine Structures

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Helene F. Skage Candidate 02

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7

BACHELOR'S THESIS

Students names:	Helene F. Skage Håkon Malkenes
Study programme:	Mechanical engineering - Marin structural engineering
Thesis title:	*Resolving gasket- and bearing failure in TBV96S valve*

Assignment text:
The actuator controlled subsea ball valve TBV96S, which controls mud flow, have several bearings that were overloaded. The students are given the assignment of:

- Identify the load on the bearings.
- Design a component to transfer the moment from the actuator to the valve.
- Dimension the bearings such that they can handle the load applied.

The values used in dimensioning are found with known mathematical formulas along with computer aided analysis.

On the same valve there are problems with leakage in the seal between the ball and the seat. The students will:

- Give a theory for the cause of the leakage.
- Research for a better material and type to be used in this application.
- Give a recommendation accordingly

Final assignment given:	Wednesday 2th. march 2016
Submission deadline:	Wednesday 4th .may 2016 12 PM
Internal tutor:	Einar Arthur Kolstad – HSH
External tutor:	Vladimir Mitin – Valves of Norway
Approved by program coordinator: Date:	15/4 -16 J. C. Lindaas

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Thesis title: Resolving gasket- and bearing failure in a TBV96S valve	
Written by: Håkon Malkenes & Helene F. Skage	Submission date: May 4, 2016
Profile: Mechanical engineering - Design of maritime structures	Study: Mechanical
Supervisors: Einar A. Kolstad & Vladimir Mitin	Grading: Open

Abstract

This report is written for Valves of Norway and addresses the selection process for gaskets and bearings for the failed valve TBV96S. In addition a mount is designed and optimized for this specific usage. The changes made on the gasket and bearings will effect some components and these will be stressed analyzed with both hand and computer aided calculations and recommendations for optimization is suggested based on the result.

Preface

In the final semester of the bachelor degree for the students at Stord/Haugesund University College they are required to write a thesis. The thesis will treat issues related to the fields that have been lectured trough the whole bachelor degree. The thesis is written for a company with the intention to show that the students can use new information in combination with existing knowledge to plan and execute an independent task, formulate questions and analyze the information at hand to answer these.

The task planned along with the firm involves choosing materials for both bearings and gaskets, stress analyzing and additionally designing a mount for the TBV96S valve. This will give the students challenges in several areas. The students will have to get both a theoretical and practical understanding of the subjects at hand, and furthermore applying their creative skills to the task.

The courses which are especially applied for this task is:

- Mechanical Design I and II
- Statics and Strength of Materials
- Process Technology I
- Materials and Manufacturing
- Underwater Technology and Hydraulic Systems

A sincere appreciation and gratitude goes to the external supervisor from Valves of Norway, Vladimir Mitin, and the internal supervisor at Stord/Haugesund University College, Einar Arthur Kolstad.

Håkon Malkenes

Helene F. Skage

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List of Variables

Variable	Description	Unit
D	Outer diameter	mm
d	Inner diameter	mm
p	Pressure	N/mm^2
A	Area	mm^2
I	Moment of inertia	mm^4
W	Section modulus	mm^3
L	Length	mm
P	Load capacity	N/mm^2
V	Velocity	m/s
F	Applied load	N
c	Cyclic velocity	cpm
θ	Oscillating angle	rad
k	Spring constant	N/mm
l_0	Length at start	mm
l_1	Length at end	mm
μ	Coefficient of friction	
β	Seat incline	$^\circ$
n	Amount	
γ	Safety factor	
σ_y	Yield stress	N/mm^2
σ_t	Tensile strength	N/mm^2
σ_p	proof strength	N/mm^2
σ_a	Allowed stress	N/mm^2
τ	Shear stress	N/mm^2
P	Pitch	mm
N	Bolt count	
K	Bolt constant	
A_s	Tensile stress area	mm^2
F_i	Pre-tension force	N
J	Torsional constant	mm^4

List of Subscripts

Subscripts is the lower letters behind a variable to describe further what exactly is calculated. These subscripts are often self explanatory since the subscripts and the abbreviation of the part names often are the same. Many subscripts indicates positions on the valve, these are shown in figure 4.3 on page 23.

Table: Subscripts used in the report.

Subscript	full form	Refers to the:
g	gasket	Gasket placed in the seat
b	ball	Hollowed out sphere in the middle of the valve
pb	pressure ball	Pressure working on the ball
ls	lower stem	Position with the lower stem bearing
us	upper stem	Position with the upper stem bearing
t	trunnion	Position with the upper stem bearing
h	hexagon	Hexagon section on the stem
lc	lower circular	Lower circular section on the stem
gr	glide ring	Position of the top glide ring on the stem
tb	thrust bearing	Position of the thrust bearing on the stem
sg	spring gasket	Force from the springs on the gasket
sh	stem hole	Stem hole diameter

Glossary

alloy A mixture of metals or a metal and another element. 15, 34

brine A solution of salt in water. 10

CAD Computer-aided design. 3

CAE Computer-aided engineering. 3

CAM Computer-aided manufacturing. 3

centroid Center of a geometry, equal to center of mass when uniform density. 31

neutral axis An axis in the cross section of a shaft/beam along which there are no longitudinal stresses or strains. 33

oscillatory motion The repeated motion in which an object repeats the same movement. 18, 38

outer most fiber The material fiber with the longest distance to the axis. 33

peripheral speed The distance a given point on the perimeter of a rotating circular object travels, expressed in m/s. 18

pivot point The center point of any rotational system. 41

polymer Polymers are made up of molecules all strung together to form long chains or structures. 11, 12

projected area The rectilinear parallel projection of a surface of any shape onto a plane. 25, 36

self-lubricating Oil-impregnated sintered metal or plastic to provide lubrication on friction. 14, 15

static load Any load that does not change in magnitude or position with time. 14

tensile strength The maximum stress a material can handle before failure due to breaking. 26

von mises It is widely used by designers to check whether their design will withstand a given load condition. It is part of a plasticity theory that applies best to ductile materials, such as metals. 34, 47

yield strength The stress level where a material reaches plastic deformation. 28, 34, 48

Summary

This report is the bachelor thesis in the mechanical engineering study with specialization in Design of Marine Structures, and addresses bearing and gasket failures with the valve TBV96S which is constructed by Valves of Norway. The task is given by Valves of Norway a valve producing company located by the west coast of Norway, outside Bergen.

Related to the gasket and bearing problems, the assignment also is to find the reason for the failures and the modifications needed to avoid these. The components affected by the changes will be stress analyzed with both hand and computer aided calculations. Lastly a mount will be designed for this specific valve.

The TBV96S valve is used for mud and has a problem with leakage between the seat and the ball inside the valve. The reason for the leakage was found to be that the hardness rating of the gasket was to low to handle the corrosion from the hard particles in the mud. This was found through examining the failed valve, and researching the materials used for gaskets. The material that is recommended for further use in the valve is PEEK.

The damaged bearings on the stem and trunnion is found to be overloaded from both the work pressure and from the actuator force. A new material is assigned to the bearings, a woven PTFE reinforced and bonded to metal backing from TENMAT called FEROGLIDE.

A maximum actuator force is then found through several finite element analysis. Further a mount is designed to counteract the friction torques generated by the gasket and bearings. The mount is designed for simplicity and low cost. It will convert the linear motion from the actuator to a rotational motion on the stem. The design is presented using 3D renderings and 2D drawings.

In case of a situations where a higher safety factor or a smaller arm is needed several proposals for mounts that will cancel out the actuator force are suggested.

1 Introduction

1.1 Background for the thesis

The oil and gas industry is a large part of the Norwegian industry. The oil and gas industry started with the discovery of the oilfield "Ekofisk" in 1969. Due to high global development in the oil- and gas industry a large number of platforms and subsea production systems are built.

These production systems contains a large amount of pipelines for transport and control of the well contents. To control the flow inside these pipelines ball valves is often used. A ball valve works by rotating a hollowed out sphere. When the hole is inline with the flow it is open, and closed when perpendicular to the flow. The purpose of the valve is not to adjust the flow and should only be used for shutoff or fully opening the flow. At subsea the valve is often controlled by a remote controlled actuator, where the actuator is either fluid, air or electrically powered and controlled. In some cases where operations of the valve is not done on a regular basis, it can be operated by an ROV (remotely operated vehicle).

Valves at subsea must be robust and maintenance free were possible. This is because of a high cost in ROV maintenance operations and/or replacement of valves. Valve of Norway have a ball valve that failed under usage and the group has been tasked to make the necessary modifications to prevent this in further use.

1.2 About Valves of Norway

Valves of Norway was founded in 1987 by engineers with experience from the North Sea oil sector. It started with maintenance and repairs on valves but they developed to a valve manufacturer. They are located at the rim of the North Sea, Ågotnes outside Bergen, and have around 20 employees.

Valves of Norway is a manufacturer of standard and customized valves in a variety of materials. They produce ball, gate, check, globe, double block and bleed/DBB/combination-/modular valves. They are ISO 9001 and PED (European pressure equipment directive) certified.

They have production facilities and a machine park in their company building. All valves are manufactured and assembled locally. Every part of the valve is made in-house and this gives them control over quality and planned lead times. The company's main goals are to strengthen their position world-wide and keep their reputation as an innovative valve manufacturer.

Valves of Norway is in the process of changing their company name to Norvalves. They will still be refereed to as Valve of Norway since this transition started after the assignment was given.

1.3 The goal for the thesis

The ball valve at hand had defects on both the bearings and gaskets due to faulty assembly and wrong usage. The valve contains three bearings one at the trunnion, and two at the stem. These take up forces from the pressure and the actuator. The gasket is placed around the ball in the valve, preventing leakage. The goal with this thesis is to modify the valve to meet the requirements of the application it is used in.

This will be accomplished by identifying the reasons for the failures and find the necessary changes needed. An overview of the materials is then presented, compered and the most suitable material will then be suggested. The material and dimensional changes will affect several components of the valve. These will be analyzed for stress with both hand and computer aided calculations. Since this is dependent on the actuator force, which is not defined, the maximum allowed actuator force is therefore found.

In addition to the tasks above. A new actuator mount that can manage the forces and moments is to be designed for the same valve. First the limitations is set, and information about the mechanisms on the market today are found, discussed and compared. Further a design will be built on top of this research. This will also be 3D modeled and stress analyzed with both hand and computer aided calculations.

1.4 Methodology

The students got a tour at Valves of Norway's facilities so that they could get an overview of the way valves were manufactured, which equipment the company had and also a better understanding of the manufacturing process. The valve and its parts was examined, which helped define the main cause of the valves failure and to give a better picture of how the valve functioned. Research was done, firstly with a focus on the material choice for the gasket and bearings and later the actuator and mount. Companies were contacted about the gasket material and dimensions to get their point of view.

Some meetings were held between the group and the external sensor Vladimir Mitin, to discuss around the calculations of the forces and friction forces in the thesis. This was formulas that was not documented in a clear way and not found in known academical books or the web.

Finite element analysis was then used in combination with hand calculations to find the most accurate solution. This was done on the stress calculations for the trunnion, stem, bearings and the actuator mount. The solution was then rendered and 2D drawings were made.

1.5 Software

The main programs used to produce this thesis was overleaf, Autodesk inventor and fusion 360. Several support programs were also used including Power Point, Keyshot, Photoshop and Soulver.

Overleaf

Overleaf is a collaborative writing and publishing system that makes the whole process of producing academic papers much quicker for both authors and publishers. Overleaf is an online collaborative LaTeX editor with integrated real-time preview. LaTeX is based on Donald E. Knuth's TeX typesetting language. LaTeX was first developed in 1985 by Leslie Lamport. It is a comprehensive and powerful tool for scientific writing and it is not a word processor. Instead, LaTeX encourages authors focus on the content and not the appearance of their documents. Overleaf was used to write the whole report.

Autodesk

Autodesk is one of the worlds leading companies in 3D design, engineering- and entertainment software and has been on the market since 1982. Autodesk software are used to design, visualize, and simulate ideas before they're built. Autodesk inventor is a 3D parametric history-based computer aided design program. It is used for creating 3D digital prototypes used in design, visualization and simulation of products. In this report it was used for the 2D drawings.

Autodesk Fusion 360 is an all in one tool with CAD, CAM and CAE. It is a multi-platform cloud-based program making it easy for collaboration. The simulation part of the program gives the the ability to predict how the product will handle the environments and situations it is designed for before it is produced. Creating virtual prototypes can reduce both cost and time in the prototyping phase. In this report Fusion 360 is used for 3D modeling and the finite element analysis done on both the valve and mount.

1.6 Limitations

Some factors are given in advance for the valve. The fluid type running through the valve is mud. Working pressure in the valve is 10 MPa, and the temperature is between 0°C and 70°C. The rotational speed and rotation angle of the ball and stem is 15 cycles/min and 90° respectively.

In the report several measurements needed from the valve is taken from the 3D model made for the previous version of the valve. This is provided by Valves of Norway, see appendix E for the 2D assembly drawing. It is assumed that all the information from the sources referred to in the report are reliable and accurate. All abbreviations, symbols and formulas that are not mentioned in the list of variables and subscripts, are believed to be generally known or known to the target audience of this thesis.

Gaskets

The company will keep their seats that are made from the material A276 S23750, a super duplex stainless steel. The materials that will be examined are the one that are most used in gaskets. The gaskets are made at valve of Norway. Valve of Norway prefers if the design is changed from two to only one gasket in each seat. The dimensions of the gasket is given by the company, see section 2.3.

Bearings

It is desired that the changes in bearing dimensions are as small as possible such that the modification of the trunnion, ball and stem is kept at a minimum. The company have a preferred bearing supplier which is TENMAT. These will be used were possible. As stated above, the rotational speed are set to 15 cycles/min. This is a maximum and will in reality most likely be much lower since repeatedly operations will happen on a rare basis. The bearings must be designed to be maintenance free, and wear resistant in corrosive and highly frictional environments.

Calculations

The safety factors used in the calculations are found in the NORSOK standard, ISO 10423 to be 1.5 for pressure containing equipment and 1.2 for the bolts. ISO 10423 is further discussed in section 1.7. The friction factor from trunnion and lower stem bearings are set to 0.1 because they partly run in mud. The loads from the bearings will be calculated as point loads. The actuator force is calculated as a perpendicular force to the flat surface on the stem. An assumption was that the deflection is at such a small scale in comparison to the tolerances that the thrust bearing would not take up any moment. The actuator mount height on the stem is adjustable but is assumed to be placed at the middle of the square section of the stem. Stresses will be calculated for both the open and closed position, it is assumed that these will generate the maximum amount of stress. Each position experiences different forces and torques acting on the components.

Actuator mount

Valve of Norway will not make the actuators, but will produce the mount in their machine park. There is an assumption that the machine park can produce the design that are proposed in the report. The report only considers solutions with one actuator mounted to the valve. Solutions with several actuators exist on the market, but this thesis will focus on a single actuator driven solution. Some assumptions are made for the calculations, these includes what height the mount is on the stem. This is assumed to be 40 mm from the top, which is in the middle of the square section of the stem.

1.7 Regulations

A standard is an agreed way of doing something. That be making of materials, products, managing or delivering a service. It is a document that provides the necessary guidelines, requirements, specifications or characteristics to make sure the quality, safety and reliability is sufficient. All tolerances are found in ISO 2768-f in appendix A if not otherwise specified.

Iso standard

The International Organization for Standardization (ISO) has developed standards since 1947 for the majority of the sectors. Norway collaborates on several important areas in the ISO standards. Most of the ISO standards contains technical specification or guidelines to secure the quality and strength of materials, products, processes and services. Each ISO standard is often concentrated on a small and specific area.

ISO 13628-4

ISO 13628-4: Petroleum and natural gas industries - Design and operation of subsea. Part 4: Subsea wellhead and tree equipment. It contains specifications regarding subsea wellheads, mudline wellheads, drill-through mudline wellheads and both vertical and horizontal subsea trees. It also contains guidelines in design, materials, welding, quality control, marking, storing and shipping. Section 7.10 in ISO 13628-4 with title: Valves, valve blocks and actuators, is the only section that has relevance for this thesis. This mostly refers to the ISO 10423 standard.

ISO 10423

ISO 10423: Petroleum and natural gas industries - Drilling and production equipment - wellhead and christmas tree equipment. It gives recommendations for technical specifications in regards of dimensions, functionality, design, materials, testing, inspection, welding, marking, handling, storing, shipment, purchasing, repair and re manufacture of wellhead and christmas tree equipment for use in the petroleum and natural gas industries. The safety factors given in the standard is 1.5 for all pressurized equipment and 1.2 for the bolts.

1.8 Finite element method

The mathematics of the finite element analysis is complex and will not be explained in detail for this thesis, but the methods will be mention for further reading. In this thesis finite element analysis is used to find the reaction forces in the original and new version of the valve. For the new version both the closed and the open position in addition to the mount will be analyzed.

The finite element method (FEM) also referred to as finite element analysis (FEA), is a numerical technique for finding approximate solutions to boundary value problems for partial differential equations. Finite meaning that it is a limited amount of elements. It is used to build predictive computational models of real-world scenarios. The method was first proposed in 1943 by Richard Courant, but did not gain attention before the 1950s and 1960s [1]. It has since been researched, tested and improved upon and is now an accurate and reliable method to use for calculating complex problems in several fields.

The finite element analysis cuts/divides a structure into several elements, it then reconnects the elements at connection points known as ”nodes”. The nodes work as pins or drops of glue that sticks the elements together. It uses the concept of piece wise polynomial interpolation to simplify the problem. Furthermore the software uses calculations such as matrix manipulations, numerical integration and a set of simultaneous algebraic equations at the nodes to estimate the solution.

The main advantages of finite element analysis over hand calculations is that it can handle very complex geometry, loads and constraints in a wide range of engineering fields. The disadvantage is that it gives a closed-form solution, meaning that parameters can not be changed in post process to examine the affect each parameter has on the system. It also only approximates solutions. In FEM user error is fatal, this is also true for hand calculations, but there they are more easily noticed. FEM also have ”inherent” errors such as simplifying the geometry. Finite element is used for problems such as thermal, fluid flow, electrostatics, elastic and many more. In this case it is used for elastic problems which will be the process in focus.

Process

Before starting the analysis, some factors needs to be defined. These are the loads, constraints, contacts and mesh. The order these are set is not important. In addition the material properties must be defined to minimize the hand calculations.

A load causes stresses, deformations, and displacements in components. It can be a force, pressure, moment, torsion or a remote force. A remote force is a force that is not on a component in the analysis but still affects it. This may be used to reduce computing time if the stress analysis for some parts are unnecessary. The force from gravity can also be included if the mass is of a significant size.

Contacts defines which relationship the bodies or components have to each other. There are several contact types to chose from, bonded, separation with no sliding, sliding with no separation or separation and sliding. Bonded is used where the components are rigid in relationship, for example where the components are welded or bolted together. Separation is when components can as the name implies separate from each other. Sliding is when the components can move freely along a surface but have continually contact with it, like a bearing on a shaft.

Constraints are limitations or restriction of movement. A combination of contacts and constraints will define the degrees of freedom for components. Constraints can be frictionless, pinned, displaced and fixed supports. Frictionless supports allow movement in the plane of the face, but restricts movement normal to it. Pinned is used to prevent displacement of cylindrical surfaces it can be fixed in the radial, axial and/or tangential direction, pinned constraint is as an example used on bearings. Fixed constraints fixes a face or component in place in the x, y, and/or z-direction, this is used when a component is fixed to a non moving support.

A mesh is the name of the wire frame that the lines between the nodes make up. The density of the mesh directly correlates with the accuracy of the result. A more dense mesh will be a better approximation of the real geometry and on that basis give more accurate results. The downside with a larger mesh is the addition in computing time needed to calculate the solution. This is why mesh refinement is used. Mesh refinement is where the mesh starts out coarse, and gets gradually increased to a point were further increase in density have an insignificant increase in precision. In fusion 360 there is also a feature called adaptive mesh refinement, this automates the process. It is set to refine the mesh until the von mises stress result converges for each section of the object, meaning that the result do not differentiate above a certain percentage from the previous result. This feature will give a high density mesh where there is a larger uncertainty and a low density were there is not. This will increase the precision without the large increase in computing time. An example of this is shown in figure 1.1 where the concentration of mesh is greatest at the stress peaks.

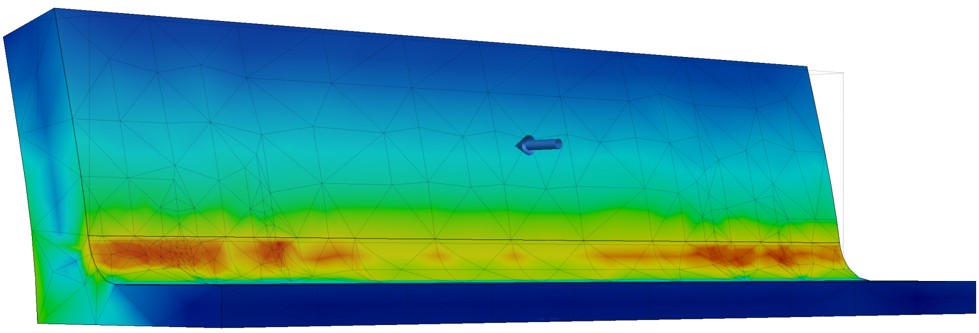


Figure 1.1: Mesh refinement

The result can now be checked for values that are out of the ordinary. Further a variety of display options can be chosen to visualize the impact including animations. Displays include but are not limited to safety factors, normal stress, shear stress, displacement and von mises stress. These results can be used to better understand how the part work under stress, and to see which component or feature that should be modified for better optimizing.

1.9 Ball valve

A ball valve is used to permit flow across the pipeline. It has an inlet and an outlet with a hollowed-out sphere/ball in the middle. When the hole is inline with the flow it is open and closed when the hole is perpendicular to the flow. There is a slot machined in the top of the ball which the stem fits in. The main components of a ball valve is shown in figure 1.2.

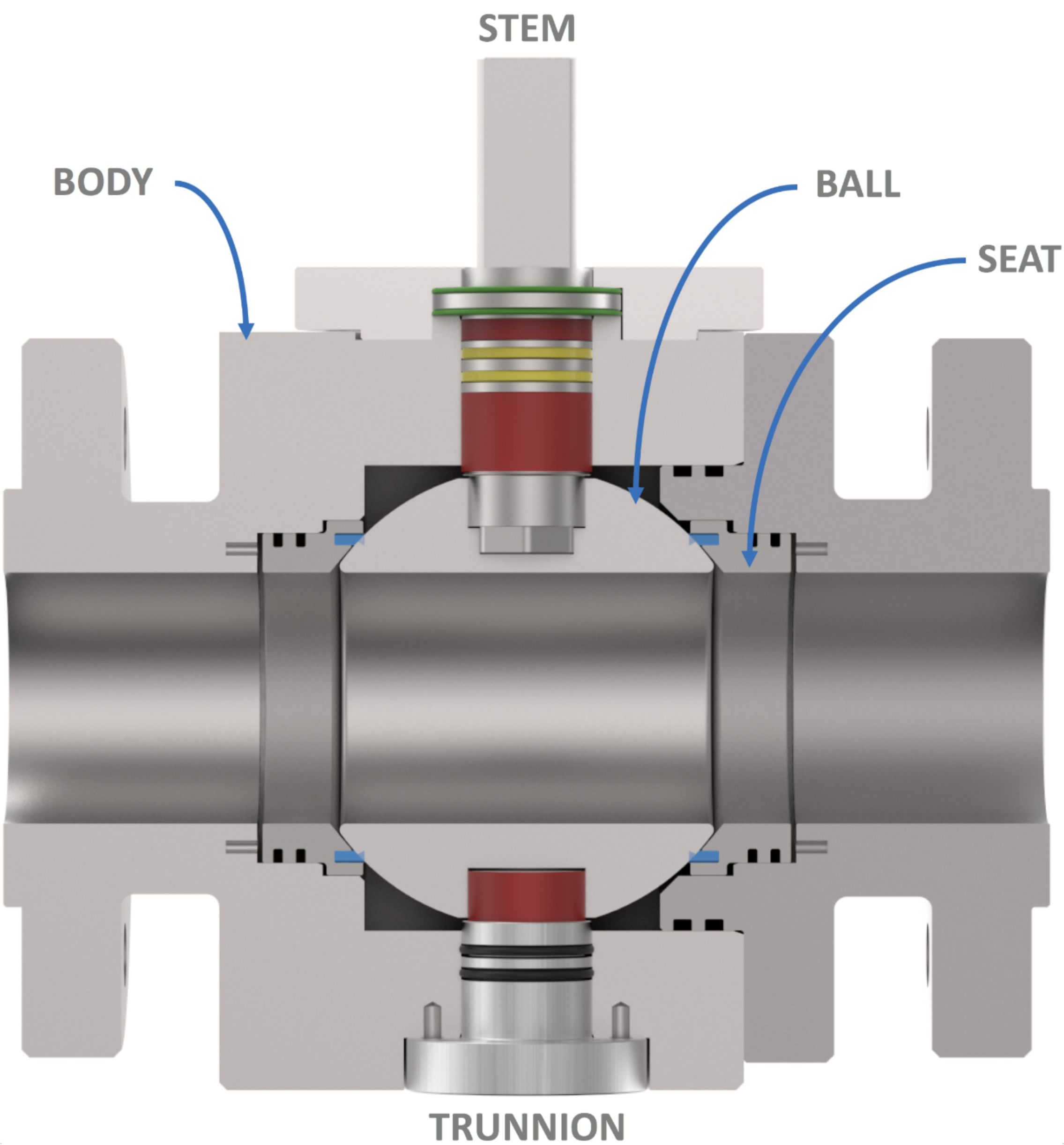


Figure 1.2: The main components of the TBV96S Valve.

The stem is the rotational loaded axle which opens and closes the valve. It connects to the top of the ball and sits in the valve body which consists of two parts, the house that holds all the components and the valve lid. The valve lid is the part that closes off the valve and is bolted to the housing.

The seat is a metal ring which contains an inserted gasket in the front against the ball, and o-rings against the body. The gasket is placed in a groove large enough to firmly hold it in place.

Ball valves can be either full bore or reduced bore. In full bore ball valves the passage is large enough to pass flow without reducing. This means that the inner diameter of the bore is the same as the inner diameter of the pipes the valve is fitted to. Reduced bore has as the name implies a reduced bore hole which also reduce the valve size and cost. These are used when the pipes are not operating at full capacity.

There are two types of ball valves. Float and trunnion. In floating ball valves the ball is supported by the stem and moves against the downstream seat when pressurized. The higher the pressure the tighter the seal between the ball and seat. Floating ball valves has seals in the stem to stop the fluid leaking at the top in addition to a seal in the valve lid pressing against the housing. These have low weight and small cost because of the simplicity. When the valve size increases the seats cant support the forces from the ball and it is necessary with a new solution.

A trunnion ball valve has an additional mechanical anchoring of the ball at the bottom. This makes the stem, ball and trunnion act as a single assembly and the bearings on both the stem and trunnion does not allow for lateral movement. The bearings will therefore take up all the force from the pressure on the ball. The seat will be pressed against the ball with both a force from the work pressure on the back of the seat and several springs working on the seat. This combined force will seal the valve. The trunnion ball valve have in addition to the seals in the floating ball valve, seals in the trunnion to stop the fluid leaking out the bottom. The advantage with the trunnion ball valve is that it can be scaled up in both size and pressure. It also has a lower operating torque which reduces the size of the actuator needed.

TBV96S

The TBV96S valve is a full bore trunnion ball valve. It has a house and valve lid that is made of A479 UNS S31600, a molybdenum-containing austenitic stainless steel. As is the trunnion, trunnion lid and the stem lid. The stem is made of the wrought nickel-base super alloy steel UNS NO 7718, and the ball and seat is made of A276 UNS S32750, a super duplex stainless steel. It is designed for use in a subsea environment, with a maximum depth of 1000 m. The main components is shown in figure 1.2.

2 Gasket

Gaskets are an integral component of any device which requires the confinement of a gas or liquid. They compensate for the irregularities of mating surfaces and creates a seal between two mechanical components. It has to maintain the seal in the intended environment and its conditions. The conditions are the pressure, temperature, fluid type and surface roughness. The gasket is pushed by a force and squeezed such that the gasket surface adapts and fills the imperfections in the mating surface.

To create a working seal three factors must be done correctly. The force must be sufficient to press the gasket as mush as needed. The force must maintain a large enough internal stress to ensure that the gasket will be in contact with the mating surface when pressure is applied. Lastly the material must be chosen such that it will withstand the stress from the pressure, temperature range it is affected by and the corrosive or eroding attack of the fluid.

The fluid in this case is drilling mud. Drilling mud is a highly viscous fluid mixture that is added to the well-bore to carry out cuttings, lubricate and cool down the drill bit. The hydro-static pressure of the drilling mud also helps to prevent the well-bore walls in collapsing, controlling pressure and provide buoyancy. It is water, brine, oil or synthetic based. Water based fluids are used more in less demanding drilling jobs at medium depth. Oil based are used in greater depths, and where the environmental concerns are important the synthetic based is used.

In this valve the failing gasket is the one between the seat and ball. On the basis of the relative small pressure and temperature size in the valve, it is assumed that the reason for the leakage is eroding attack from the fluid. The mud is a corrosive fluid containing hard particles which will erode a softer gasket at a fast phase. Figure 2.1 shows a picture of the failed gasket. This shows that the inner gasket is "eaten" up" which supports the assumption. For this reason the gasket material is changed from the previously material TFM with a hardness of around 59 to a new one.

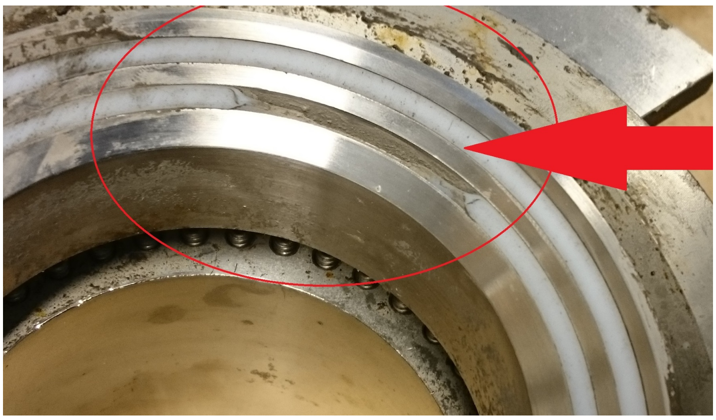


Figure 2.1: The damaged gasket

2.1 Seal type

Metal to metal seals are used where the use of elastomeric and polymer seals are not an option due to high wear, radiation, very high pressure, high vacuum or high temperature in the application. It is especially useful where there is a need for fire-safe equipment. To create a metal sealing often a tungsten carbide coating or electroless nickel plating is used on the metal. The tolerances have improved greatly and the leakage rate for the ANSI standards is down to an impressive 0.01% leakage of full open valve capacity. The major down side with metal to metal seals is its cost.

Soft insert seal are made of softer and more flexible materials called elastomers. The group of elastomers includes materials which can be stretched and bent without exerting great force. Once the deforming force relaxes or no longer acts at all, the parts take their original shape. The term elastomers, which is derived from elastic polymer, is often used interchangeably with the term rubber. Elastomers can be made in varying degrees of hardness, but are sensitive to light, ozone, high temperatures, a large number of fluids and chemicals, and wear [2]. It is easier to obtain a near zero leakage for softer seals, but they will be more vulnerable for corrosive mediums, chemical mediums, temperature and pressures.

Hard insert seal are a middle ground of the two above. This solution uses a thermoplastic ring inserted into the seat. The material used are mostly in the fluoropolymer group. A fluoropolymer is a polymer that contains molecules of carbon and fluorine. They are high-performance plastic materials used in harsh chemical and high-temperature environments, primarily where a critical performance specification must be met.

The valve has a working pressure of 10 MPa and a temperature range from 0°C to 70°C which all seal types can handle. The leakage rate allowed is not specified and is therefore not assumed to be the highest priority. Mud is therefore seen as the main factor. It has particles with a high hardness rating, meaning a high resistance of plastic deformation by indentation, also referred to as the resistance to scratching/wear. The specific hardness is unknown because of the variety of mud compositions but it is assumed that the metal to metal seal is not needed. Hard insert is therefore chosen for its strength yet lower cost.

2.2 Material

In table 2.1 some of the most known fluoropolymers are compared to each other with focus on the hardness shore [3–5]. Be aware that the hardness shore is an approximated value because of small differentials depending on the volume of the materials mixed within the polymer. This depends on the supplier and the number is only there to give an estimate of the polymers hardness. The temperature range for the materials is also documented. They are found at Boedeker Plastics webpage [6] and shows that all the fluoropolymers withstand the required operating temperature range.

Table 2.1: Fluoropolymer materials for gaskets

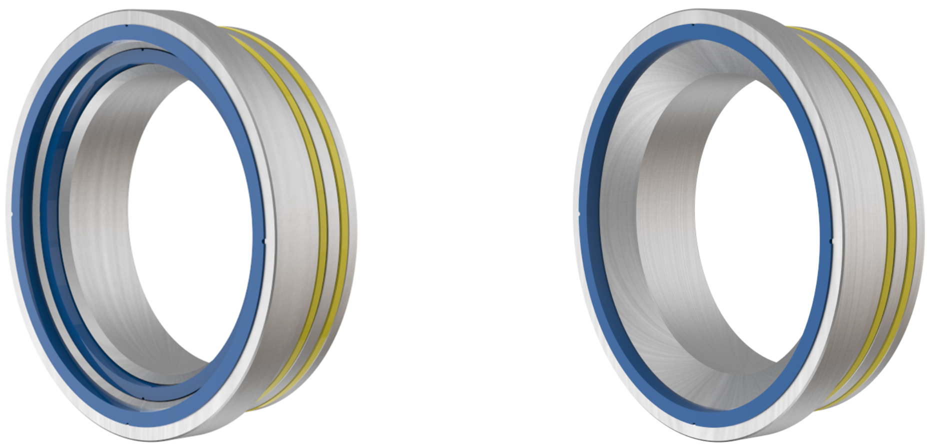
Material	Information	Hardness Shore
PTFE (Polytetra - fluoroethylene) or TFE (Teflon®)	Is a fluorocarbon based polymer and typically is the most chemically-resistant of all plastics while retaining excellent thermal and electrical insulation properties. TFE also has a low coefficient of friction so it is ideal for many low torque applications. Although TFEs mechanical properties are low compared to other engineered plastics, its properties remain useful over a wide temperature range (-73°C to 204°C)	56
TFM	Is a modified second generation TFE polymer that maintains the chemical and heat resistance properties of first generation PTFE. It has a denser polymer structure than standard PTFE with better stress recovery. The temperature range is similar to PTFE (-73°C to 204°C).	59
PCTFE (polychlorotri-fluoroethylene), Kel-F® or Neoflon®	It combines physical and mechanical properties, non flammability, chemical resistance, near-zero moisture absorption and excellent electrical properties not found in any other thermoplastic fluoropolymer that also performs well in a wide temperature range (-240°C to 204°C).	75-85
PEEK (Poly ether ether ketone)	Is a high temperature, high performance engineered thermoplastic offering an unique combination of chemical, mechanical and thermal properties. PEEK is considered a premium seat material with its excellent water/chemical resistance and due to the fact it is unaffected by continuous exposure to hot water/steam. PEEK is non-porous, high strength for high pressure (400 bar) applications and is suitable for high corrosion environments. PEEK typically adds to the torque requirement of the valve given the rigidity of the material. It can be used up to 250°C.	99

2.3 Solution

When considering the gasket material the ones that can give an upgrade from the TFMs hardness rating is PCTFE and PEEK. Although PEEK is a more cost heavy gasket, and in addition it increases the required torque and thus the size of the actuator, it is chosen for its hardness. PEEK has the highest hardness rating, and should therefore handle the wear from the mud.

The increase in hardness from the gasket change gives an increase in the necessary preload from the springs on the gasket. This preload is there to make the contact pressure between the gasket and the ball large enough to prevent leakage. To make the changes on the valve as small as possible, the same springs are used but the start length will be changed, giving them a larger force. This is further explained and calculated in section 4.1. Because of the several assumptions made in the selection of the gasket, the valve should be tested in a testing facility according to the ISO 10423 standard.

In addition to the material selection, valves of Norway requested a more standardization of the design for the gaskets. Today's version of the valve is featured with two gaskets in each seat. This is shown in figure 2.2a where the gasket is shown in blue. After research in gasket manufacturers solutions, it is found that the most widely used solution is seats with a single gasket. On that basis, the new valve will be manufactured with one gasket instead of two. This is shown in figure 2.2b.



(a) The old double gasket solution

(b) The new single gasket solution

Figure 2.2: Illustration of old and new gasket solution

It is preferable to chose a standard measurement of the gasket such that it is easily manufactured or ordered quickly. In collaboration with the external supervisor, the dimension for the new gasket will be 126 mm for the outer diameter, and 116.7 mm for the inner diameter. [1]

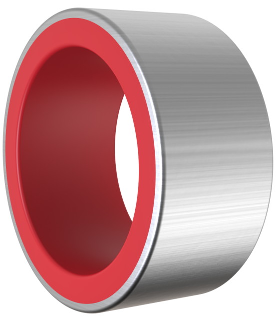
[1] The values were found after consulting Vladimir Mitin, the engineer at Valves of Norway. The values is based on long experience and testing.

3 Bearings

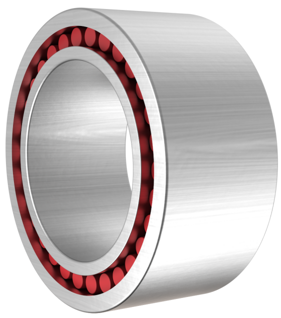
In contacting surfaces with high friction, bearings are implemented to avoid failure by reducing friction. The relevant bearing type for this purpose is roller and plain bearings. Roller bearings transfer forces with rollers in the shape of balls or needles in different forms, placed between two metal cylinders. Plain bearings, also called bushings or sleeve bearings, transfers forces by sliding. The different bearing types are illustrated in figure 3.1a and 3.1b.

Roller bearings are especially suitable for tasks that requires high precision combined with high speeds and high dynamic loads. Roller bearings have an exceptionally low coefficient of friction. If correct lubrication is regularly preformed they may have a longer life span than the plain bearing and are also available with lower clearance. The downsides with roller bearings are the high cost, low vibration damping and that most roller bearings must have lubrication at all times. That said, there are some maintenance-free roller bearings on the market, but they do not inherit the precision, speed and load handling capabilities. However they are often better in corrosive environments.

Plain bearings can further be divided into three types, one for each type of motion that the plain bearing could be subjected to. These are linear, thrust and journal. In linear bearing the bearing slides in the length direction of the shaft it is mounted on. Thrust bearings are placed between to mating surfaces to reduce friction when the surfaces rotates in different directions or speeds. Journal bearing consist of a shaft that rotates inside the bearing placed in a static housing. Plain bearings can be self-lubricating and are vibration dampening. They can handle high static load, are cost and space efficient and have low weight in comparison to roller bearings. Most plain bearings also have high resistant to dirt pollution. The downside for plain bearings are the higher coefficient of friction and the higher clearance which in turns reduces the capable speed and the achievable precision.



(a) Plain Bearing



(b) Roller Bearing

Figure 3.1: The two main types of bearings

Because valve maintenance is both expensive and difficult to execute in subsea environments, the bearings must be maintenance free and have a long life span. The fluid running through the valve is mud and this increases the friction and wear of the bearing. It must also handle low speeds and large forces. The type that meets most of the requirements is the self-lubricating plain bearings.

3.1 Material

When choosing the materials for the bearings there are some factors that must be taken in to account. These are load capacity, friction coefficient, velocity, temperature and required life span. The wide range in material choices ranges from white metals, alloys to plastics. In self-lubricating plain bearings the material choices are listed in table 3.1 [7, 8] :

Table 3.1: Material for bearings

Material	Information	Max. stress [N/mm^2]
Unfilled thermoplastics	It has the lowest cost, is superior in corrosive environments and has a high tolerance to soft material in mating partner.	10
Filled/ reinforced thermoplastics	Injection molded resin with additive fiber as base material. Second lowest cost, compatible with abrasives and adequate toleration of soft counter-face.	Filled: 12 + reinforced: 35 + bounded to metal back: 140
Filled/ reinforced PTFE	PTFE is one of the most versatile plastic materials. It is used as coating, extrusion and molding. The common reinforcements is glass fiber, carbon, bronze and graphite. Handles low temperature exceptionally. Scores good inn handling corrosion, abrasives. It also has low wear and low friction which in turn means a longer life span.	Filled: 7 + bounded to metal back: 140 + reinforced: 420
Filled/ reinforced thermosetting resins	Thermosetting is similar but stronger to thermoplastics. It can handle higher loads, but has some higher wear and cost.	Filled: 30 + reinforced: 50
PTFE impregnated porous metals	PTFE impregnates in the micro-pores in the metals. Can handle higher speeds, high loads. It also has low wear/long life and low cost.	350
Woven PTFE/ glass fibre	Can handle high loads. Has high stiffness and low wear.	Woven + Reinforced + bounded to metal back: 420
Carbons -graphite	Handles high temperatures and corrosive environments. Has low wear and is Chemically inert	1.4 – 2
Metal -graphite mixtures	Good in high temperature and high speeds. Scores high in dimensional stability and is compatible with fluid lubricants.	3.4
Solid film lubricants	Coating of fine particles of lubricating pigments, binder and additives that cures in to a solid film. Has low friction and scores high in handling high and low temperatures, high speeds and high loads. Good stiffness and stability. Has high tolerance to soft counter-faces.	
Ceramics, cermets, hard metals	Tackles high temperatures, high loads fluid lubrication and corrosion. It is also dimensional stable.	

3.2 Reaction forces

The most widely used method for calculating the materials ability to withstand the frictional energy generated is the PV factor. This is the load capacity times the peripheral speed. To high PV value will give unstable temperatures that leads to rapid failure. To get a reasonable value, a safety factor of 2 is added.

The given work pressure $p = 10$ MPa $= 10$ N/mm^2 pushes on the ball with a diameter of $D_b = 176$ mm. The stem and ball will be rotated at a maximum of $c = 15$ cycles/min and have an oscillating angle of $\theta = 90° = 1/2 \cdot \pi$. The gaskets has an outer diameter of $D_g = 126$ mm and an inner diameter of $d_g = 116.7$ mm.

There is both a force directly on the ball, and a force on the seat from the working pressure. This will in turn create reaction forces on the three bearings. With three unknown forces and two usable equations, the system is unsolvable. Computer aided analysis is therefor used to find these forces. In this situation fusion 360 was used, see appendix M for the report. The direct and indirect forces working on the bearings are calculated in equation 3.2 and 3.4. The total force on the ball from the work pressure is then calculated in equation 3.5.

The work pressure will be inside the valve house, with the stem and trunnion, stopping at the trunnion seals and the stem seal/glide rings. This will cancel out most of the force from the pressure on the seat. This means that the force from the work pressure on the seat will only come from the area of the gasket. In addition, the force on the ball from the work pressure will only affect the area inside the seat gaskets.

$$A_{pg} = \frac{\pi \cdot \left(D_g^2 - d_g^2\right)}{4} = 1773 \text{ mm}^2 \tag{3.1}$$

$$F_{pg} = A_{pg} \cdot p = 17727 \text{ N} \tag{3.2}$$

$$A_{pb} = \frac{\pi d_g^2}{4} = 10696 \text{ mm}^2 \tag{3.3}$$

$$F_{pb} = A_{pb} \cdot P = 106960 \text{ N} \tag{3.4}$$

$$F_{tot} = F_{sp} + F_p = 124689 \text{ N} \tag{3.5}$$

The results from the computer aided analysis shows that the reaction forces was for the upper stem $F_{us} = 3.6$ kN in the negative direction, the lower stem $F_{ls} = 48$ kN and for the trunnion $F_t = 73$ kN.

With hand calculations the ratios can be compered. The net force and net moment adds up to zero for static problems. L_1 is the distance from the trunnion to the ball center, where the force F_b acts. L_4 is the distance from F_b to F_{ls} and L_5 is from F_{ls} to F_{us}. This is shown in figure 3.2. From this the equations below are found.

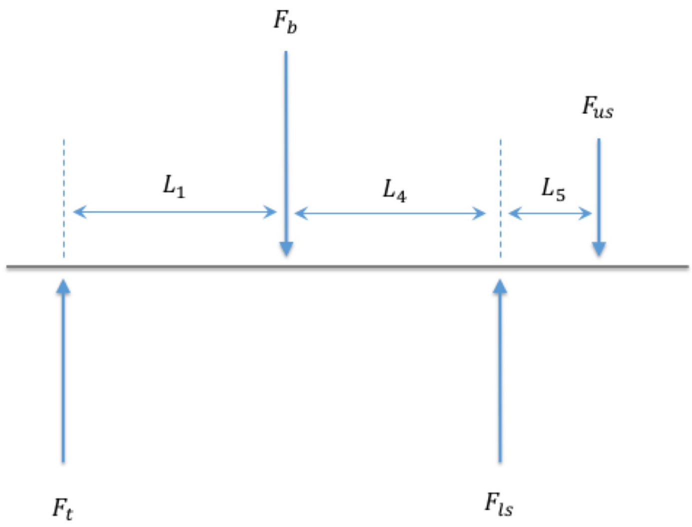


Figure 3.2: Free body diagram

$$\sum M_t = F_b \cdot L_1 + (L_1 + L_4 + L_5) \cdot F_{us} - (L_1 + L_4) \cdot F_{ls} = 0 \tag{3.6}$$

$$F_{us} = \frac{F_{ls} \cdot (L_1 + L_4) - F_b \cdot L_1}{L_1 + L_4 + L_5} = -4 \text{ kN} \tag{3.7}$$

$$\sum F_x = F_t - F_b + F_{ls} = 0 \tag{3.8}$$

$$F_{us} = F_t - F_b + F_{ls} = -3.6 \text{ kN} \tag{3.9}$$

The hand calculations and the FEM result gave a difference of around 500 N in equation 3.7. This is a small difference and only support the conclusion that the finite element analysis gave the correct values. These values will be used in further calculations.

3.3 PV factor

To find the PV factor, the rotational speed and working load need to be found. These are dependent on the bearing dimensions which is commonly defined by the length-to diameter ratio. The length-to diameter ratio is often set in the range from 0.5 to 1.5, and if the diameter is known the ratio often set to 1.

In this case the dimensions from the previous version of the valve will be used for an estimation. Values for each bearing is shown in table 3.2. The peripheral speed is calculated with the formulas in appendix D. Since the valve operates with an oscillatory motion, equation numbered two in the appendix is used.

$$P = \frac{F}{L \cdot d} \qquad (3.10)$$

$$V = \frac{dc\theta}{10^3} \qquad (3.11)$$

Table 3.2: Data for calculating PV factor

	d [mm]	L [mm]	F [kN]	P [N/mm^2]	V [m/s]	PV·2
Upper stem	46	7.8	3.6	10	0.009	0.18
Lower Stem	49	30	48	33	0.0096	0.63
Trunnion	44	19	73	87	0.0086	1.50

Appendix B shows that the materials suitable for the lower stem- and trunnion bearings is the PTFE with filler, bonded to steel backing with a PV value up to 1.75, or the woven PTFE reinforced and bonded to metal backing with a PV value up to 1.60. Both can handle the required loading. Other materials could be chosen for the upper stem bearing, but with forces from the actuator this would also require the same material as the others.

3.4 Selection from supplier

Valves of Norway are now buying bearings from the company TENMAT. They want to continue this. For use in valves, TENMAT recommends their FEROGLIDE material for oil and gas applications that demands high strength and low friction. It is a woven PTFE fabric with strengthening fibers applied to a metal backing. The maximum loading in static and dynamic loading situations are listed in table 3.3 and 3.4, these includes comments about when these are valid.

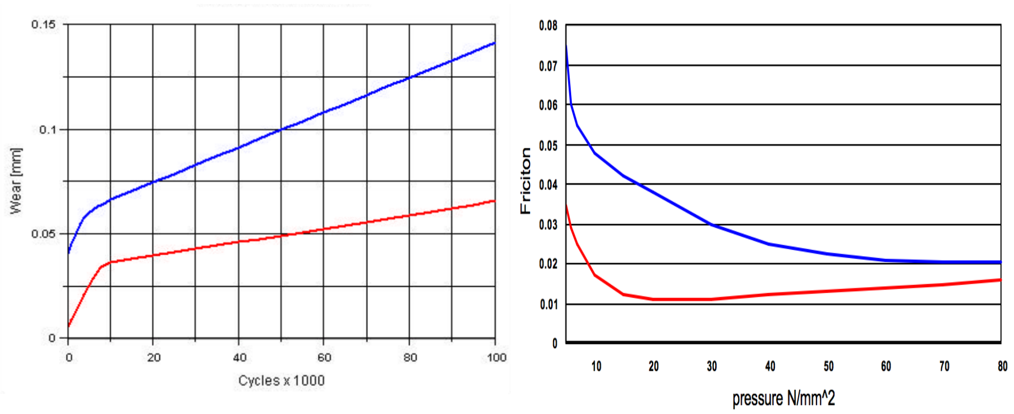
Table 3.3: FEROGLIDE static load capacity

Maximum static load [N/mm^2]	Backing metal
210	Mild steel
240	Bronze
420	Inconel

Table 3.4: FEROGLIDE dynamic load capacity

Maximum dynamic load [N/mm^2]	Comments
14 – 28	Long life
140	Recommended Maximum Load
176	High Strength backing metals

Static is used where there is little to no movement. Dynamic loading is when there is oscillating motion or linear movement below 10 m/min. In this case it is an oscillating motion and the table for dynamic loading is used. The recommended maximum load is chosen to give the bearings long life span, but still be capable of handling the forces necessary.



(a) Wear (b) Friction

Figure 3.3: FEROGLIDE wear and friction graphs

Figure 3.3a and 3.3b shows two lines on both graphs. One in red and one in blue. The FEROGLIDE report does not describe the difference of the two. For that reasons it is assumed that the two lines indicate the minimum and maximum values. To be on the safe side, the values for the blue lines are therefor used in further calculations.

Figure 3.3a shows the wear on a journal bearing with a load of 140 N/mm^2. The bearing was fixed and the shaft was oscillating with an amplitude of $\pm 45°$. The frequency was 10 cycles/min [9]. This is close to the values for this valve and is therefor a good representation of what to expect when used. For the usage the valve is designed for the wear will be low and hold for the life span of the valve.

The dimension choices for the FEROGLIDE journal coiled bearings are found in appendix C and the dimensions closest to the previous version of the valve are chosen and listed in table 3.5 with comments about the necessary changes of the valve. All dimensions are in millimeters.

Table 3.5: FEROGLIDE bearing dimensions

	d/D	L	Comment
Trunnion	40/44	20	The ball hole length must be machined 1 mm in the height (The new length is 1 mm longer then the old), and 1 mm must be added to ball hole diameter. In addition the trunnion diameter must be machined down 4 mm beneath the bearing.
Lower stem	45/50	30	The stem diameter beneath the bearing must be machined down 4 mm.
Upper stem	45/50	7.8	The stem diameter must be machined 2.52 mm beneath the bearing, and the length must be cut or specially ordered.

4 Calculations

The changes made to the gasket and bearings will affect several components of the valve which will have to be recalculated for the new version. This includes the spring force, friction torques and the stress on the stem, trunnion and bearings, calculated in that order. For all of them the safety factor is 1.5 from the standard, see section 1.7. The effect on the ball and seat are small and the calculations of stress on these components are neglected in this thesis.

Some assumptions are made, this includes what height the mount is on the stem. This is assumed to be 40 mm from the top, which is in the middle of the square section. This height is adjustable and if the mount is placed higher the result in this thesis should be viewed as an underestimate, and maximum allowed force should be reduced some.

Assuming that the highest stress occurs in either the fully opened or closed position, the actuator force is calculated as a perpendicular force to the flat surface on the stem. The stress on the stem and trunnion will be a combination of stresses from shear, torsion and moments. Calculation of the stress on the bearings will be simplified to only radial force and the area will be the projected area, this was agreed upon with the external supervisor.

The actuator can push on all four sides of the stem. The four directions is the positive and negative x- and y-directions, when the pressure is working in the positive x-direction also shown in figure 4.1. The worst case scenario is the one that is further checked. To find the worst case scenario, several FEM analysis are done with the force from different directions while changing the actuator forces. Through a large number of analysis, it was found that when the actuator force worked in the opposite direction of the work pressure it gave the highest reaction forces on the bearings. This is therefore the direction that will be calculated, tested and designed for.

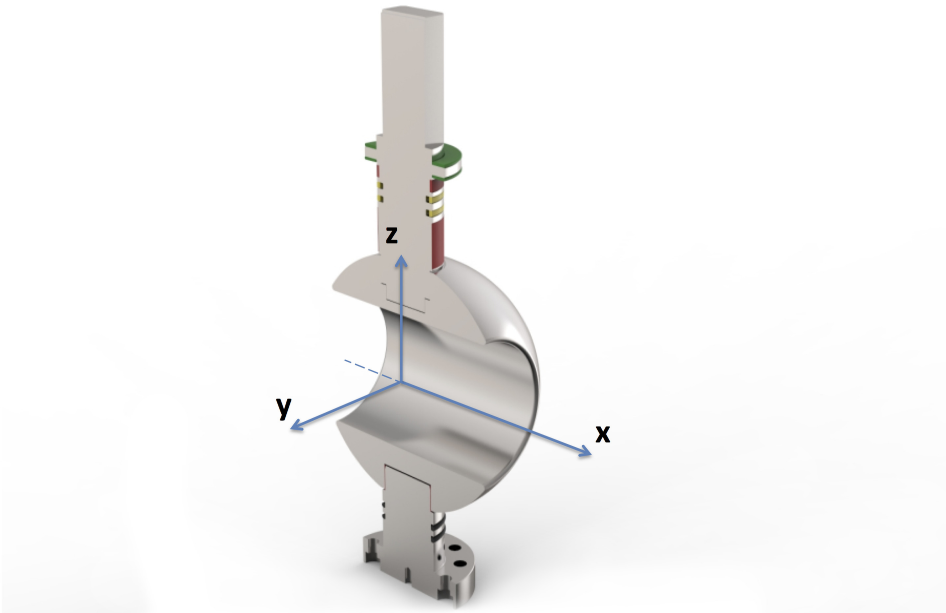


Figure 4.1: Illustration of the directions when the ball is in open position

Since both the stress on the bearings, trunnion and stem depend on the magnitude of the actuator force, the maximum force from the actuator must be found to have as a limit. This is done to have a value for the minimal arm length when designing the mount.

A parametric Soulver file will be created where all equations discussed in the next subsections are implemented and the reaction forces and moments from the bearings are imported from the finite element analysis. Soulver is a calculator software. It can remember variables, which can later be used in formulas. This means that values can easily be changed and the files can be saved unlike in a physical calculator.

A number of FEM analysis are then run with different actuator forces, checking each one in soulver to find the force which will bring one of the components to its allowed stress limit. The FEM setup is shown in figure 4.2. The complexity and amount of these calculations is at a scale that makes it unnecessary to have in the body of this thesis and is therefor placed in appendix K . Only the methods used and the results are shown in the body.

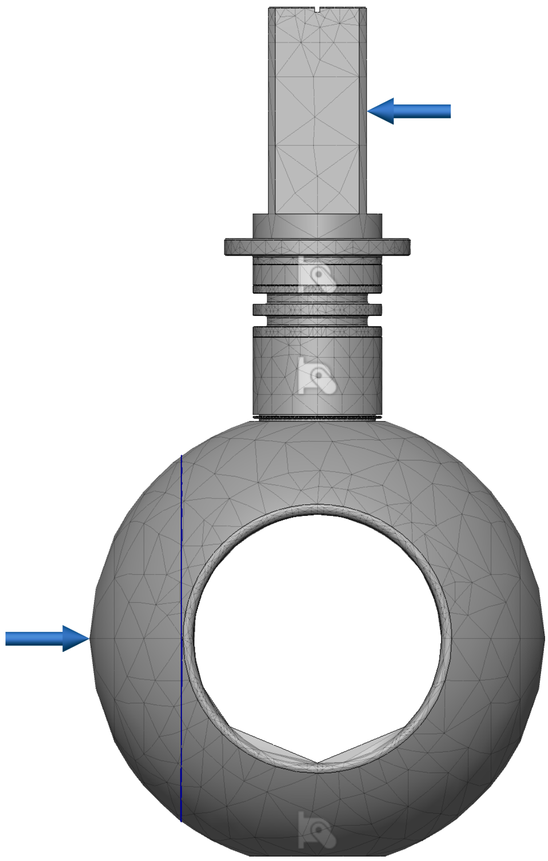


Figure 4.2: FEM setup

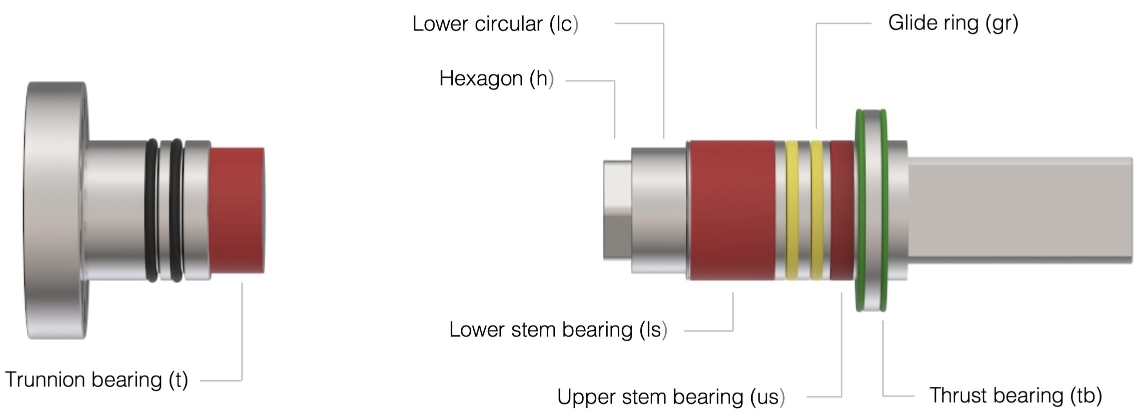


Figure 4.3: Notations used in calculations

The figure 4.3 shows the cross sections that the stress is calculated for, excluding the thrust bearing. Also their notations is written in parenthesis, which is their subscript in the calculations. These sections are chosen because of their change in geometry and some by the local moment peaks. Figure 4.4 shows further how the stem is divided up.

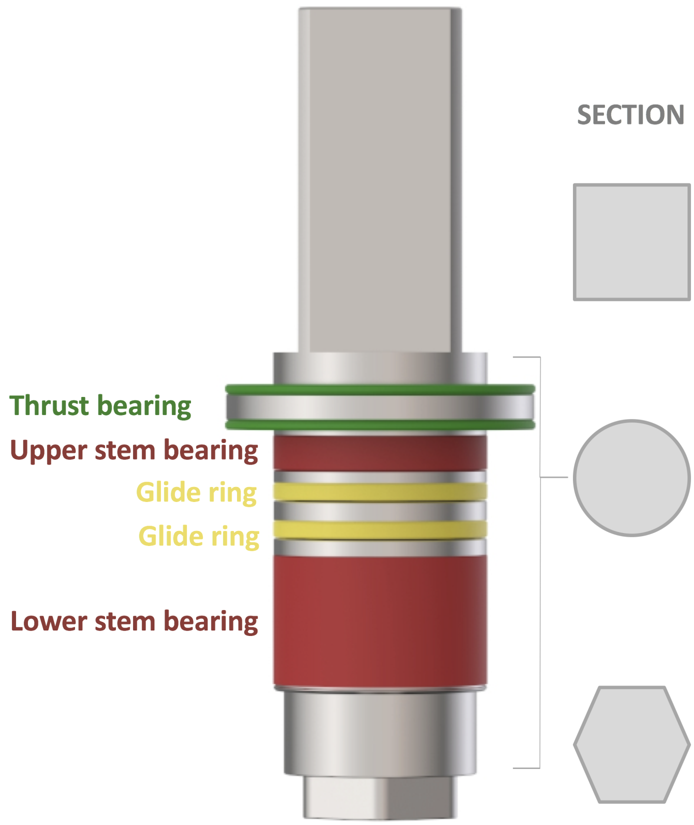


Figure 4.4: Illustration of the components used on the stem

4.1 Friction torques from components

There are friction torques from several components. On the stem there is the upper and lower stem bearing shown in red and the thrust bearing shown in green in figure 4.4. Additionally there is the gasket and the trunnion bearing. These friction torques will in turn create the torsion.

Friction torque from bearings

The friction coefficients for FEROGLIDE are found in the graph from TENMAT in figure 3.3b on page 20. For pressures above 40 MPa the coefficients will be rounded up to 0.03 because of the uncertainty with the friction graphs. This value is used for the upper stem bearing. The lower stem bearing and the trunnion bearing will run partly in mud and this increases the friction coefficient greatly and is therefore estimated to 0.1.[2] If the seals were to be placed below the lower bearing the friction coefficient increase would be avoided for this bearing. On the other hand the distance from the force on the ball would increase the force and stress on the trunnion bearing, which it would not withstand with the material it is assigned.

The friction torque from the bearings are calculated with an equation from SKF [10], where the arm is the radius of the bearing. The new friction coefficients and dimensions for each bearing is listed in table 4.1. The frictional torque is dependent on the magnitude of the actuator force and is therefore calculated for each FEA, where equation 4.1 is used.

$$M = \frac{F}{2} \cdot \mu \cdot D \tag{4.1}$$

Table 4.1: Friction torque on the bearings

	μ	L [mm]	D [mm]
Upper stem	0.03	7.8	50
Lower stem	0.1	30	50
Trunnion	0.1	20	44

[2]From meeting with Vladimir Mitin where the friction coefficients were discussed. 23/02/2016

Friction torque form glide rings

The friction torque from the glide rings depend on the projected area of the glide ring on the stem hole, and the arm which is the radius from the stem center to the glide ring edge. The glide rings are made of the material TFM [3], with a friction coefficient of 0.06. This is increased to 0.1 to take the effect of the mud in to account. The friction torque is calculated in equation 4.2, where the height of the glide ring is $w = 3.9$ mm and the diameter of the stem hole is $d_{sh} = 50$ mm.

$$M_{gr} = \mu \cdot p \cdot w \cdot d_{sh} \cdot \frac{d_s h}{2} = 4875 \text{ Nmm} \tag{4.2}$$

Friction torque from the thrust bearing

The thrust bearing shown in green in figure 4.4 takes up all the vertical force from the work pressure. The pressure works on the area of the stem hole. The valve is designed for a depth of 1000 m, which will also give it a pressure of $L \cdot \rho \cdot g = 1000 \text{ m} \cdot 1000 \text{ kg/m}^3 \cdot 9.81 \text{ m/s}^2 \approx$ 10 MPa on the top of the stem directed downwards. This will create a reaction force of equal size and they cancel each other out. To get a conservative result it is therefor calculated for a vacuum. The thrust bearing is made from PETP and the friction coefficient is $\mu = 0.18$ [11]. The outer diameter and inner diameter is $D_{tb} = 72.5$ mm and $d_{tb} = 50.5$ mm as in the previous version of the valve.

$$M_{tb} = \mu \cdot p \cdot \pi \cdot d_{sh}^2 \cdot \frac{1}{4} \cdot \frac{1}{4} \cdot (D_{tb} + d_{tb}) = 60377.48 \text{ Nmm} \tag{4.3}$$

Friction torque from the gasket

The gasket gives a force from both the 40 springs and the resultant force from the work pressure on the seat. The force from the work pressure was calculated in equation 3.2 on page 16. These will generate the friction torques from equation 4.6 and 4.7. These equations are used by the engineers at Valves of Norway, found in [12]. The equation has a correction factor to compensate for the inconsistent arm or the distance from the gasket to the vertical center line of the ball. This correction factor is dependent on the slope or incline of the seat, which in this case is $\beta = 50°$.

The spring force is in accordance with Vladimir Mitin determined by the suggested preload of the gasket. The equation for preload is based on experience and testing. It is found to be around 5% of the maximum allowed loading which is calculated as 60% of the tensile strength $\sigma_t = 110$ N/mm^2 [5]. [3]

The same springs will be used in the valve as in the previous version. To get the correct spring force the start length will be changed. The spring coefficient is set to $k = 55$ N/mm, found in the data sheet for the springs which is located on page xxvi. The end length is found in the 3D model to be $l_1 = 13.9$ mm.

$$F_{sg} = 0.05 \cdot 0.6 \cdot \sigma_{tensile} \cdot \frac{\pi}{4} \cdot \left(D_g^2 - d_g^2\right) = 5850.01 \text{ N} \tag{4.4}$$

$$l_0 = \frac{F}{n_{sg} \cdot k} + l_1 = 16.56 \text{ mm} \tag{4.5}$$

The gasket has an outer diameter of $D_g = 126$ mm and an inner diameter of $d_g = 116.7$ mm. The friction coefficient is approximately 0.4, a high friction coefficient in comparison to the bearings and seals. This along with the large force from the work pressure gives the gasket the highest friction torque of all the components.

The friction torque from the gaskets will depend much on the position of the valve. If it is closed there will only be pressure on one side of the ball. In this way only one of the gaskets will generate friction torque. When the valve is opened the work pressure will work on both sides and the friction doubles. The result from this on the stress analysis is discussed further in the next section. The force from the springs will work on both sides for both positions.

$$M_{pg} = F_{pg} \cdot \mu \cdot \frac{1}{2} \cdot D_b \cdot \frac{(1 + \sin(50^\circ)) \cdot 0.5}{\sin(50^\circ) + 0.05 \cdot \cos(50^\circ)} = 690325.60 \text{ Nmm} \tag{4.6}$$

$$M_{sg} = 2 \cdot F_{sg} \cdot \mu \cdot \frac{1}{2} \cdot D_b \cdot \frac{(1 + \sin(50^\circ)) \cdot 0.5}{\sin(50^\circ) + 0.05 \cdot \cos(50^\circ)} = 227807.45 \text{ Nmm} \tag{4.7}$$

[3] From email correspondence with Vladimir Mitin where the formula is based on testing and long experience in the valve business 01/03/2016, the e-mail is in appendix J.

4.2 Stress Calculations

The stress will be different for each section of the trunnion and stem. It will be a combination of torsion, shear and moment stress. The stress will also be different for the two positions; open and closed. When the valve is open, there will be a pressure working on the gasket on both sides of the ball. In this position there will be no reaction forces from the bearings, meaning no friction torques from the bearings, no shear force and no moment.

If the valve is closed, there will only be friction torque from one of the gasket, but the friction torque, shear force and the moment from the bearings will have an effect here. Because of the high friction torque of the gasket, both situations have to be checked to see which gives the highest total stress.

The different sections are the trunnions circular section, the stems hexagonal section at the bottom of the stem. Further the stems lower circular section in addition to its circular sections below the two bearings and the seal/glide rings. These sections are chosen because of the change in geometry and/or the size of the moment, shear and torsion.

The values shown are the final result with the maximum actuator force. The moment of inertia and the first moment of area for the hexagon section is dependent on where the neutral line will go. In this case it will go through the corners when closed. For the hexagon the dimensions used are the width of $w = 36$ mm and side lengths of $a = 21$ mm. For the circular section the diameters are listed in table 4.2.

Table 4.2: Diameters of circular sections

	Trunnion	Low circular	Lower stem	Seal	Upper stem
Diameter $[mm]$	44	45	45	39	45

Torsion

The torsion will increase for each addition of frictional torque, starting with the trunnion bearing and adding up to the total at the thrust bearing. The distribution is shown in the torsional diagram in figure 4.5. The highest jump is from the gasket, which have a high frictional torque because of its high friction coefficient, high force, and a long distance to the center of the valve. Since this is above the trunnion, the trunnion itself can be made of a material with a lower yield strength than the rest. At the top of the stem there will be a total frictional torque that the actuator has to counter with a moment in the opposite direction.

The friction torques depend heavily on the position of the valve. If it is open there will be a force of zero on the trunnion bearing and the lower- and upper stem bearing, but the pressure affect both gaskets. This significantly increases the torsion. In this position the torsion will be constant at any actuator force. When the valve is closed on the other hand, the friction torque from the bearings will vary with the actuator force as described previously.

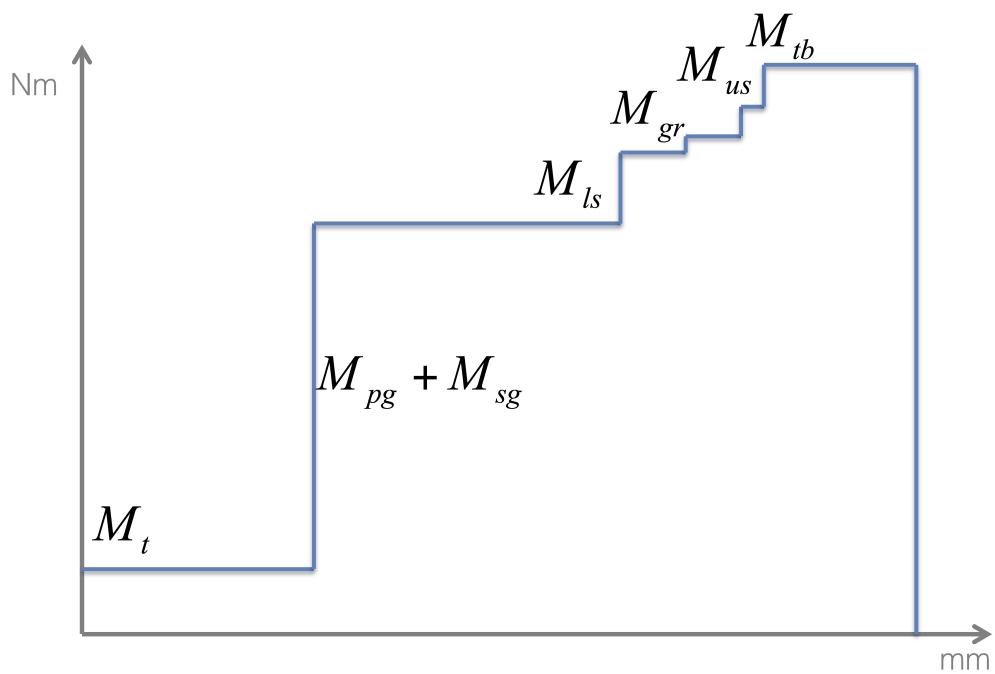


Figure 4.5: Torsion diagram

To calculate the stress from the torsion on circular sections which is a uniform cross-section, equation 4.9 is used. Where T is the torsion, r is the radius and J is the torsion constant which is equal to the polar moment of inertia, which is used to estimate the objects ability to resist torsion.

$$J = \frac{1}{2} \cdot \pi \cdot r^4 \tag{4.8}$$

$$\tau_t = \frac{T \cdot r}{J} = \frac{16 \cdot T}{\pi \cdot d^3} \tag{4.9}$$

The equation above only applies for uniform cross-sections. Non uniform cross-sections such as the hexagon behaves non-symmetrically when torque is exerted on them. In addition to this the stress distribution is often non-linear. This means for hexagonal sections that the equation is estimated to equation 4.11 [13], where the width (w) is measured from the flat sides of the hexagon. The results are listed in table 4.3 for both open and the closed position.

$$J = 0.0649 \cdot w^4 \tag{4.10}$$

$$\tau_t = 5.297 \cdot \frac{T}{w^3} \tag{4.11}$$

Table 4.3: Torsional stress with closed and open valve

$[Nm]$	Trunnion	Hexagon	Low circular	Lower stem	Seal	Upper stem
Closed	164	1310	1310	1410	1415	1436
Open	0	1836	1836	1836	1841	1841

Shear

Shear stress is caused by forces acting in a direction parallel to a surface or in other words, a force acting along a plane that passes through a body. The forces in this case are the ones from the work pressure, the actuator force and from the reaction forces in the bearings.

In figure 4.6 the shear diagram is illustrated. This is not to scale and only have the purpose of illustrating how the shear forces is over the trunnion and stem. V_A is from the trunnion force. The force from the work pressure will then bring the shear force below zero to V_B. Further it will change direction for the lower and upper stem bearing. Finally ending at zero. The calculation method for the shear forces are shown in equation 4.12.

$$V_A = F_t \tag{4.12}$$

$$V_B = V_A - (F_{ps} + F_{pb}) \tag{4.13}$$

$$V_C = V_B + F_{ls} \tag{4.14}$$

$$V_D = V_C - F_{us} \tag{4.15}$$

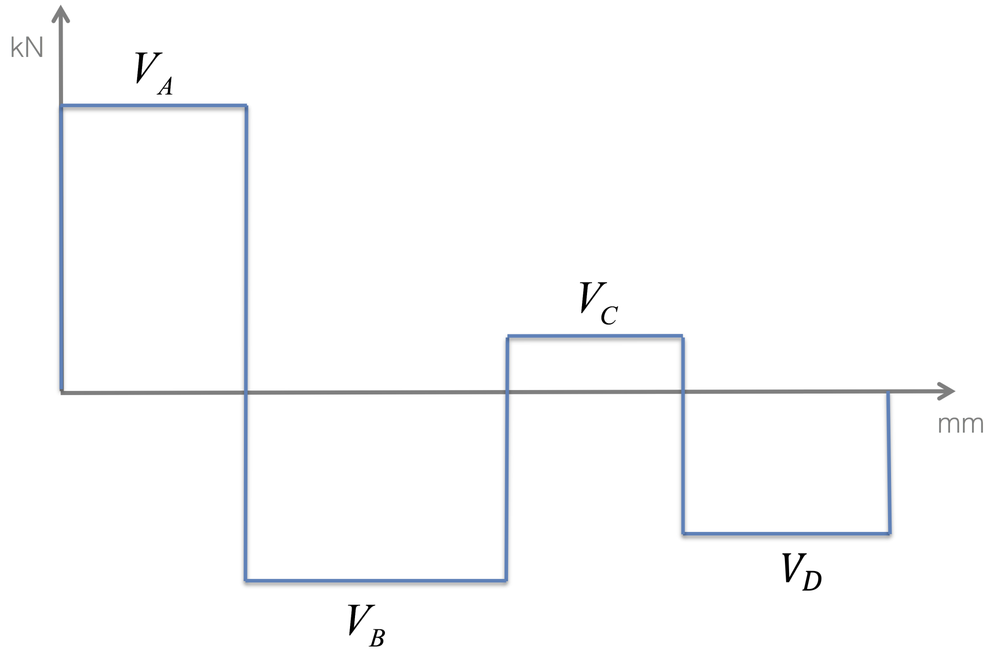


Figure 4.6: Shear force diagram

In many cases, especially for long sections, the shear stress is neglected because of its small size in comparison to the stress from moment. For small and thick beams the shear stress has a much larger importance. The shear force that comes from bending is called transverse shear and is at its highest at the neutral axis and at zero with the walls. The geometric factors of the transverse shear is the width of the object, the moment of inertia I, and the first moment of area Q. First moment of area is the area of either the above or below cross-section of the center line, times the distance from the center line to this areas centroid. Equation 4.18 shows the method of calculating shear stress for circular sections.

$$Q = \sum a_i \cdot y_i = \frac{\pi \cdot r^2}{2} \cdot \frac{4r}{3 \cdot \pi} = \frac{1}{12} \cdot d^3 \tag{4.16}$$

$$I = \frac{\pi}{64} \cdot d^4 \tag{4.17}$$

$$\tau_s = \frac{V \cdot Q}{I \cdot d} = \frac{16}{3} \cdot \frac{V}{\pi \cdot d^2} \tag{4.18}$$

To solve for Q for the hexagonal section, the hexagon is divided in two. A top and bottom half. These are equal and which of them the area is found for is the same. To find the area the half is further divided into a square and two right angled triangles, one on each side. The triangles and square will have a height of $w/2$, and the length of the hypotenuse of the triangles and the width of the square is $a = 20.67$ mm, which is the length of the hexagon sides. From this the stress is found with the equations below.

$$Q = \sum a_i \cdot y_i = \overbrace{\frac{w}{2} \cdot a}^{\text{area of square}} \cdot \overbrace{\frac{w}{4}}^{\text{distance}} + 2\overbrace{\left(\frac{1}{2} \cdot \frac{w}{2} \cdot \sqrt{a^2 - \left(\frac{w}{2}\right)^2}\right)}^{\text{area of triangles}} \cdot \overbrace{\frac{1}{6} \cdot w}^{\text{distance}} \tag{4.19}$$

$$Q = \frac{1}{8} \cdot a \cdot w^2 + \frac{1}{12} \cdot w^2 \cdot \sqrt{a^2 - \left(\frac{w}{2}\right)^2} \tag{4.20}$$

$$I = \frac{5 \cdot \sqrt{3} \cdot a_h^4}{16} \tag{4.21}$$

$$\tau_s = \frac{V \cdot Q}{I \cdot w} \tag{4.22}$$

The trunnion works more like a pin than a beam and is therefor calculated for average shear stress. $\tau_s = V/A$. And for the section with the lower stem bearing and the upper stem bearing the shear force will be zero. The results are shown in table 4.4.

Table 4.4: Shear stress

	Trunnion	Hexagon	Low circular	Seal
τ_s [N/mm^2]	65	21	36	2

Moment

The moment is defined as the force multiplied with the lever arm which is the perpendicular distance between the the center of moments and the force. In this case the shear force above is used and the lever arms are listed in table 4.5.

Table 4.5: Lengths

Notation	Length [mm]	Description
L_1	75	From trunnion to ball center
L_2	56	From ball center to start of hexagon
L_3	66	From ball center to start of lower circular
L_4	100	From ball to mid lower stem bearing
L_5	30	From mid lower stem bearing to mid second seal
L_6	39	From lower to upper stem bearing
L_7	63	From upper stem bearing to middle of square stem

The finite element analysis calculates that the bearings not only take up forces but also some amount of moment. These moments are added as jumps in the moment diagram. The moment diagram in figure 4.7 is not to scale and only there for illustration purposes.

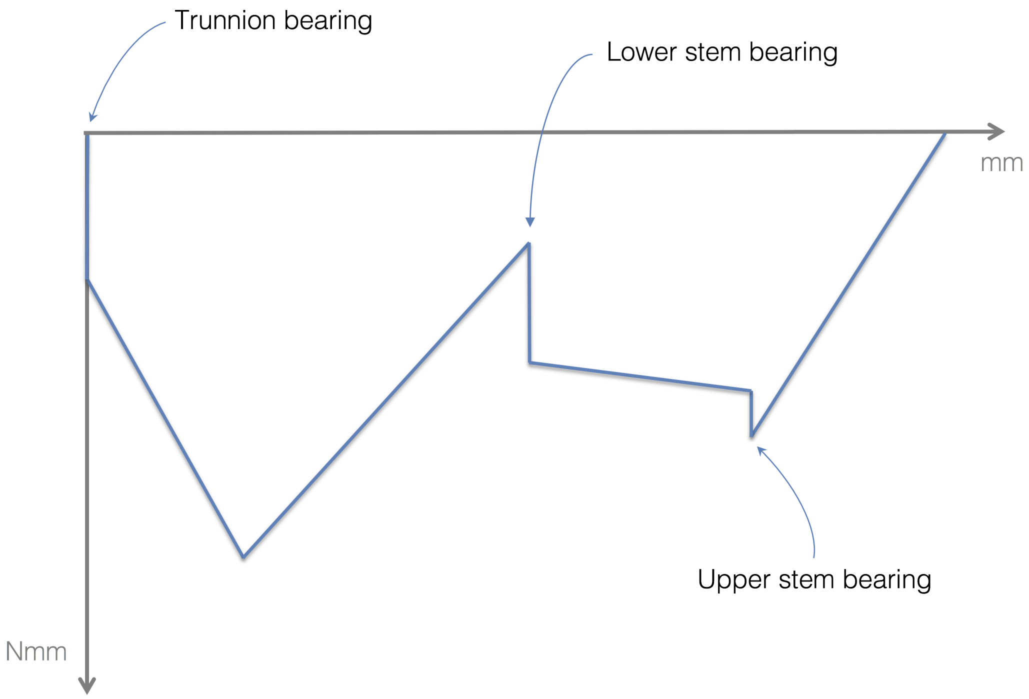


Figure 4.7: Moment diagram

The moment of inertia (I) and the distance to the outer most fiber (y) is calculated in different ways for the circular and hexagon cross-sections. For the circular sections the outer most fiber is a distance of the radius from the neutral axis, and equations for moment of inertia is shown below.

$$y = \frac{d}{2} \tag{4.23}$$

$$I_x = \frac{1}{64} \cdot d^4 \tag{4.24}$$

For the hexagon section, the outer most fiber is a distance of the width divided by two.

$$y = \frac{w}{2} \tag{4.25}$$

$$I = \frac{5 \cdot \sqrt{3} \cdot a_h^4}{16} \tag{4.26}$$

With the moment, moment of inertia and the distance from the neutral axis to the outer most fiber, the stress from moment can be found with equation 4.27. The results are listed in table 4.6.

$$\sigma = \frac{M}{I} \cdot y \tag{4.27}$$

Table 4.6: Moment stress

	Trunnion	Hexagon	Low circular	lower stem	seal	upper stem
τ [N/mm^2]	96	134	303	140	226	205

Total stress

The total stress is a combination of the torsion, shear and moment. To estimate the equivalent stress, the von mises criteria is used. It is denoted as σ_v and this must be lower then the total allowed stress for the material. The von mises stress result is listed in table 4.7.

The material used for the stem is a nickel based super alloy called UNS NO7718. It has a yield strength of $\sigma_y = 827\ \text{N/mm}^2$ [14]. With a safety factor of 1.5 the allowed stress is $\sigma_a = \sigma_y/1.5 = 551.33\ \text{N/mm}^2$.

The trunnion is made of a weaker material since it experiences a smaller amount of stress. It is made of the stainless steel A479 UNS S31600 with a yield strength of $\sigma_y = 204\ \text{N/mm}^2$ [15], giving it an allowed stress of $\sigma_a = 136\ \text{N/mm}^2$.

$$\sigma_v = \sqrt{\sigma^2 + 3\left(\tau_s + \tau_t\ \right)^2} \tag{4.28}$$

Table 4.7: Von mises stress

	Trunnion	Hexagon	Low circular	lower stem	seal	upper stem
Closed	135	327	357	195	309	248
Open	0	367	178	178	278	178

The results are shown as utilization of the material and listed in table 4.8. If this is close to hundred percent, the utilization is good and no changes need to be done. If it is above hundred percent, the material can not be used or the actuator force should be reduced. If it is well below, possible changes will be suggested but not implemented in this thesis.

$$Utilization = \frac{\sigma_v}{\sigma_a} \cdot 100\ \% \tag{4.29}$$

Table 4.8: Utilization with closed and open valve

	Trunnion	Hexagon	Low circular	lower stem	seal	upper stem
Closed	99	59	65	35	56	45
Open	0	67	32	32	50	32

4.3 Finite element analysis

After many trials and errors, tweaking the constraints and loads the final result was found. All bearings were constrained with pin constraints. For the lower and upper bearing only the radial direction was locked. To take up the small vertical force the trunnion bearing was locked in both the radial and the axial direction. This is in reality taken up by the thrust bearing, but if the thrust bearings was included in the analysis they would in addition take up a large amount of the moment. This may be the case if the deflection is large enough but not likely, so these were excluded from the analysis.

The highest actuator force was found to be around 30 kN were the failing section was the trunnion in addition to the bearing on the trunnion. By adding the torsion to the analysis, the stress for each part was found and listed in table 4.9. There were singularities, but these are ignored. The analysis also converged to a convergence rate of $2 - 4\%$ with the von mises stress results. The utilizations are listed in table 4.10.

A side note is that the peak stress for the hexagon and the lower circular part were at the top of each section instead of at the bottom as in the hand calculations.

Table 4.9: Stress from FEM

	Trunnion	Hexagon	Low circular	lower stem	seal	upper stem
Closed	120	360	350	200	325	220
Open	20	400	220	150	400	230

Table 4.10: Utilization from FEM

	Trunnion	Hexagon	Low circular	lower stem	seal	upper stem
Closed	90	65	65	35	60	40
Open	15	75	40	30	70	40

4.4 Stress on bearings

The calculation for the bearing stress is simplified to be plain compression stress on the projected area of the bearing. The projected area is the width times the diameter. The safety factor is set to 1.5 from the ISO 10423 standard. With a maximum dynamic load of $\sigma_{max} = 140$ N/mm^2, the total allowed stress is $\sigma_a = \sigma_{max}/\gamma = 93.33$ N/mm^2 for the FEROGLIDE bearings. The force acting on them is the reaction forces, these are found with finite element analysis. The stress and utility results are listed in table 4.11 and 4.12

$$A = b \cdot h \tag{4.30}$$

$$\sigma = \frac{F}{A} \tag{4.31}$$

$$Utilization = \frac{\sigma}{\sigma_{allowed}} \cdot 100\% \tag{4.32}$$

Table 4.11: Stress on bearings

	Trunnion	lower stem	upper stem
Stress [N/mm^2]	93	30	81

Table 4.12: Utilization on bearings

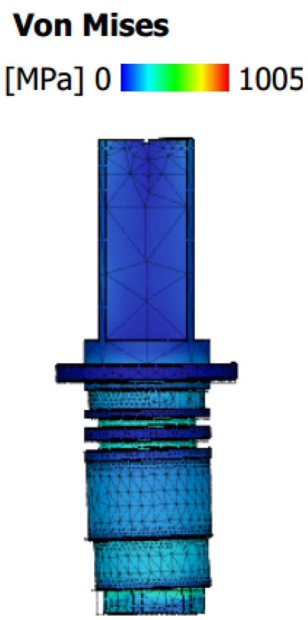
	Trunnion	lower stem	upper stem
Utilization [%]	99	32	87

4.5 Conclusion stress calculations

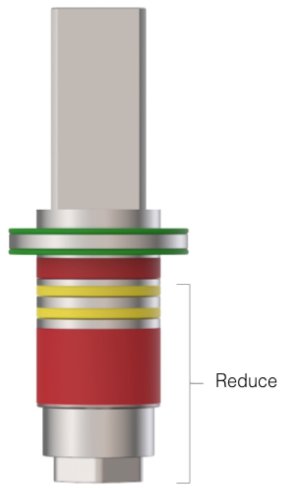
There was a small gap between the FEM analysis and the hand calculations. The FEM analysis showed a higher stress on especially the seal at the open position, which gave it an utility of a little over 70%. The difference is of a small scale and most likely from the complexity of the calculation. Therefore the results mostly support each other and the conclusion will be that they are correct. In figure 4.8a the von mises stress is shown, where the least affected areas are shown in blue and the more affected areas are shown in green. The yellow and red areas from the scale at the top of the figure is not visible in the model since they indicate the points with singularities and are to small to notice.

As the calculations shows, most sections only have a utilization of around $40 - 60\%$. This means that the trunnion, trunnion bearing and the upper stem bearing could be scaled up to accommodate more stress, or the other sections could be scaled down. The last solution will reduce the material usage and size, and in turn the material cost on the valve. The sections that should be reduced is shown in figure 4.8b.

The results gave an actuator force of 30 kN which gives a minimum arm length of $L = T_{tot}/F_{act} \approx 50$ mm. Since there is no safety factor for the friction torques, other than the added factor for the friction coefficient on the bearings running in mud, the friction torques may be a conservative value. If a safety factor of 1.5 is added the minimum arm length will be 75 mm. Combined with a stem width of 40 mm, this is still a small arm length and will not create any problem when designing the mount.



(a) Von Mises when closed



(b) Possible modifications

Figure 4.8: Result stress analysis

5 Valve Actuator

For this valve the actuator shall meet the requirements of the ISO 10423, ch.10.16.3 [16]. An actuator is a motor that controls and moves mechanisms or systems by taking a power source in the form of hydraulic fluid, pneumatic pressure or electric current to perform work. This can be in the form of linear, rotary or oscillatory motion. They have a diverse area of usage in many fields, including engineering.

Actuators are used when human operations are not desirable/possible, and when operations are done either quickly or frequently. An advantage is also that the actuator and in turn the valve can be controlled remotely. In this situation the actuator has a linear motion and the motion is transformed to a rotary motion by the mount.

5.1 Types of actuators

There are three types of actuators that are mainly used, each type giving different mount solutions. The actuator type that is most likely to be used with this valve is found, and the mount is then designed with this in mind. The information about each type is found in [17, 18].

Electric actuator

Electric actuator converts electrical energy into torque. An electric motor mechanically connected turns a lead screw. It is very simple to connect and wire, also very easy to use. Electrical actuators offers high precision-control positioning, they provide complete control of motion and can include encoders to control velocity, position, torque, and applied force. The initial unit cost of an electrical actuator is higher than pneumatic and hydraulic actuators, but it has very low operating cost. The reliability is great because of its repeatable, reproducible performance during the entire product life and very little maintenance is required. It does not have as long life expectancy as the hydraulic actuator, and it is not suited for all environments.

Pneumatic actuator

Pneumatic actuator consist of a piston inside a hollow cylinder. Pressurized air from an external compressor or manual pump moves the piston inside the cylinder. Pneumatic actuators generate precise linear motion. The cost of pneumatic actuators is low compared to other actuators and pneumatic actuators are also lightweight, require minimal maintenance and have durable components. This makes pneumatic actuators a cost-effective method to produce linear motion. The pneumatic actuator is also safe to use in hazardous and flammable areas. The main disadvantage with this type of actuator is its medium. Air is a compressible medium which makes this a less efficient actuator.

Hydraulic actuator

Hydraulic actuators operates similarly to pneumatic actuators, but uses an in-compressible fluid from a pump rather than pressurized air to move the cylinder. It requires plumbing, filtering, pumps etc. and requires electronic/fluid interfaces and valve designs to be operated. The speed is difficult to control accurately, but it gives virtually unlimited force and is the most powerful choice. The actuator can produce forces 25 times greater than pneumatic actuator of equal size.

A hydraulic actuator can hold the force and torque constant without the pump supplying more fluid or pressure, this due to the in-compressible fluid. The pumps and motors can be placed a considerable distance away with minimal loss of power. Hydraulics is contamination sensitive and seals are prone to leak, but it has a longer life expectancy than electrical when maintained regularly. Components often cost less than the electrical, but installation and maintenance are high.

Figure 5.1 illustrates the principle of hydraulic actuators, where the fluid is pumped up from a tank and delivered to one of the ends of the actuator pushing on the actuator piston and shaft. The return is either done with springs or by changing the flow direction.

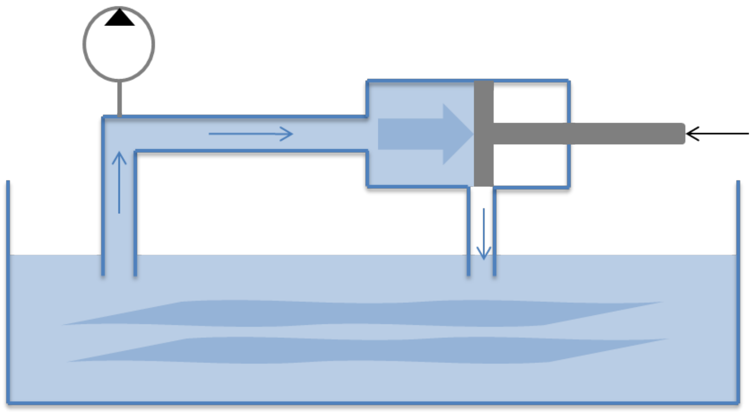


Figure 5.1: Illustration of how the fluid actuator work.

Choosing the right type of actuator

In a subsea environment there is most likely already installed a hydraulic power unit and therefore this cost is not taken in to consideration. The electric actuator requires little maintenance, but have shorter life expectancy. The pneumatic actuator are less efficient and not suitable in subsea environment because of its compressible medium. In this task precision is not important and for this and the reasons above it is assumed that a hydraulic actuator is used. The mount will therefore be designed for this.

5.2 Commonly used mechanisms

The commonly used mounts are scotch yoke, slider-crank, lever and the rack and pinion.

The scotch yoke is a mechanism that converts linear motion to rotational motion by having a slot in a sliding yoke. A pin on a off-center point on the rotational part is then placed in the slot. When the yoke slides forward and backward the pin inside slides up and down, resulting in a rotational motion, see figure 5.2. This is an efficient production of rotational motion as it spends more time at the high points of the rotation then a piston and has few parts. The setup is commonly used in control valve actuators in high-pressure oil and gas pipeline [19].

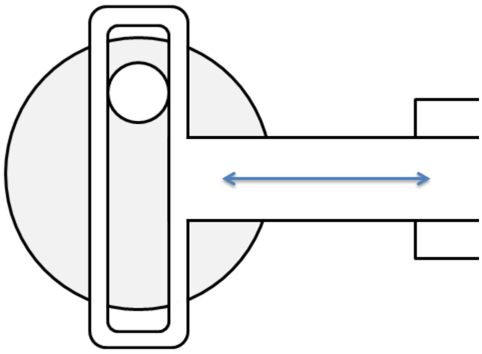


Figure 5.2: Illustration of the scotch yoke.

The slider-crank mechanism transfers the linear motion from the actuator to a crank, also called a lever, see figure 5.3. The mechanism contains two joints to maintain the necessary freedom of motion. The lever is there to create a larger moment on the rotational part. If the lever is longer, the moment gets larger or the force can be reduced. The downside of scaling up is that the necessary travel length of the slider would increase. This mechanism contains many parts and rotating joints that all need bearings.

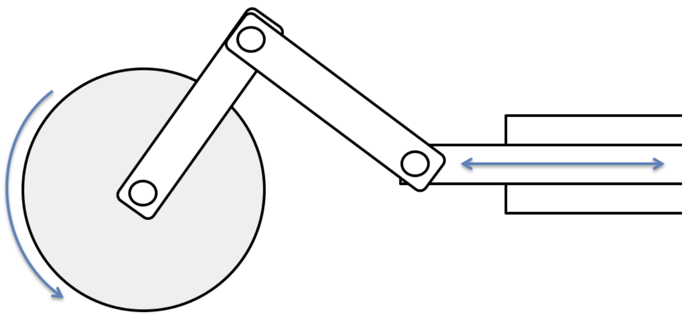


Figure 5.3: Illustration of the sliding crank.

Lever mechanism is much like the slider-crank mechanism, but instead of two joints, it uses a slot to maintain the necessary degrees of freedom, see figure 5.4. This is the opposite method that the scotch yoke mechanism uses, where the slot is in the yoke. The advantage is that it has few parts and uses a small amount of space. There are three types of lever: First order, second order and third order. The first order levers works by placing the pivot point in between the input force and output force. Since the forces are on different sides of the pivot point, they will act in the opposite directions. In the second order lever, both forces are placed on the same side of the pivot point. Were the input force is furthest away. In this type of lever both forces will act in the same direction. In the third-order lever the input and output forces are still on the same side, but now the output force is on the outer end. This will also give the forces the same direction. The advantage here is the larger travel distance of the output, but in turn the input force must be larger to counter the moment.

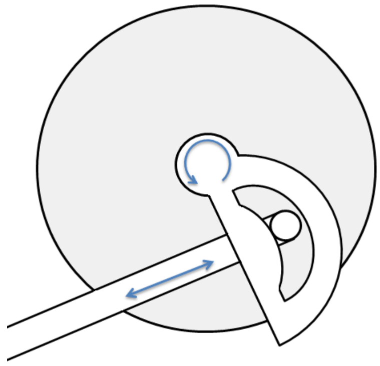


Figure 5.4: Illustration of the lever solution.

The rack and pinion mechanism use gears to convert the motion. The pinion is the gear on the rotating part, this is being rotated by a linear "gear" bar that is called the rack. The rack travels linearly on the edge of the pinion with the the teeth engaged. This pushes the pinion in a rotational motion, see figure 5.5. The solutions contains very few parts, and is therefor a simple and robust mechanism. It does not take up a large amount of space but the gears may be sensitive to debris.

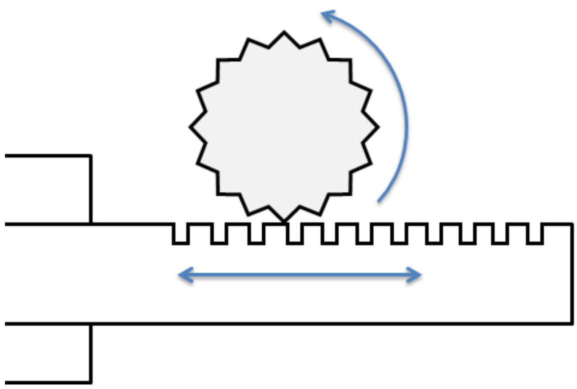


Figure 5.5: Illustration of the rack and pinion mechanism.

5.3 Design

When designing a mount a principle base is chosen from the most commonly used mechanisms and then innovated on for this specific situation. The mechanism choice is predominantly based on the simplicity, in addition to the material and space usage.

The slider-crank mechanism is not preferred here at all due to its large motion range, and that it contains several parts, including several bearings which only adds to the costs and complexity in this project. The rack and pinion does not take up a lot of space, but due to the gears it could be more sensitive to debris, and therefore should be maintained regularly to check for this, which is not possible in this case. Both the slider-crank and the rack and pinion mechanisms have arms that must be an extension to the actuator arm so the actuator may need to be attached further from the stem. For these reasons the slider-crank and rack and pinion are discarded as a base mechanism.

The scotch yoke is small in size and easy to produce. The actuator can be placed closer to the stem in both directions, making it a compact solution which can be mounted at the center line of the valve if needed. This reduces the travel length of the actuator shaft. The lever also has a small space usage and are easy to produce with little complexity and a small material usage. Based on these advantages, further development will be based on the scotch yoke and lever.

Pointers to take into consideration when designing the mount is that the stem end is a square with a 39×39 mm^2 area and the mount mechanism is being mounted to this. The height on the stem should be adjustable to accommodate for different actuator models and sizes. The actuator should also be able to be mounted to the valve, meaning that the distance from the stem center to the actuator shaft should not be longer then around 200 mm, a length estimated from studying the 3D model and the 2D assembly shown in appendix E. The necessary turn angle is 90°, and the solution does not need to accommodate a larger one.

Furthermore the material chosen is a stainless steel A479 UNS S31600 with a maximum allowed stress of 206 N/mm^2, same as the trunnion. The slot in the mount should also be rounded in such a way that the resultant force from the actuator always is near tangential to the stem. For this reason the actuator must deliver a higher force if it is at an angle to the tangential of the stem to make the resultant force large enough to give the necessary moment. This is further discussed in the calculations.

Trough brain storming some concepts were drawn up and presented in figure 5.6 with notes about their positive and negative features. Some contains features from both the scotch yoke and the lever, and some is based solely on the lever mechanism with different connection methods. The mechanism numbered one uses a pin connected at both ends trough a slot in the mount but the connection method could be changed to all of the three in the top of the figure. The mechanism numbered two uses a linear bearing and a pin in a square slot, and the mechanisms numbered three uses a slot in the shaft of the actuator to push a pin fixed to the mount.

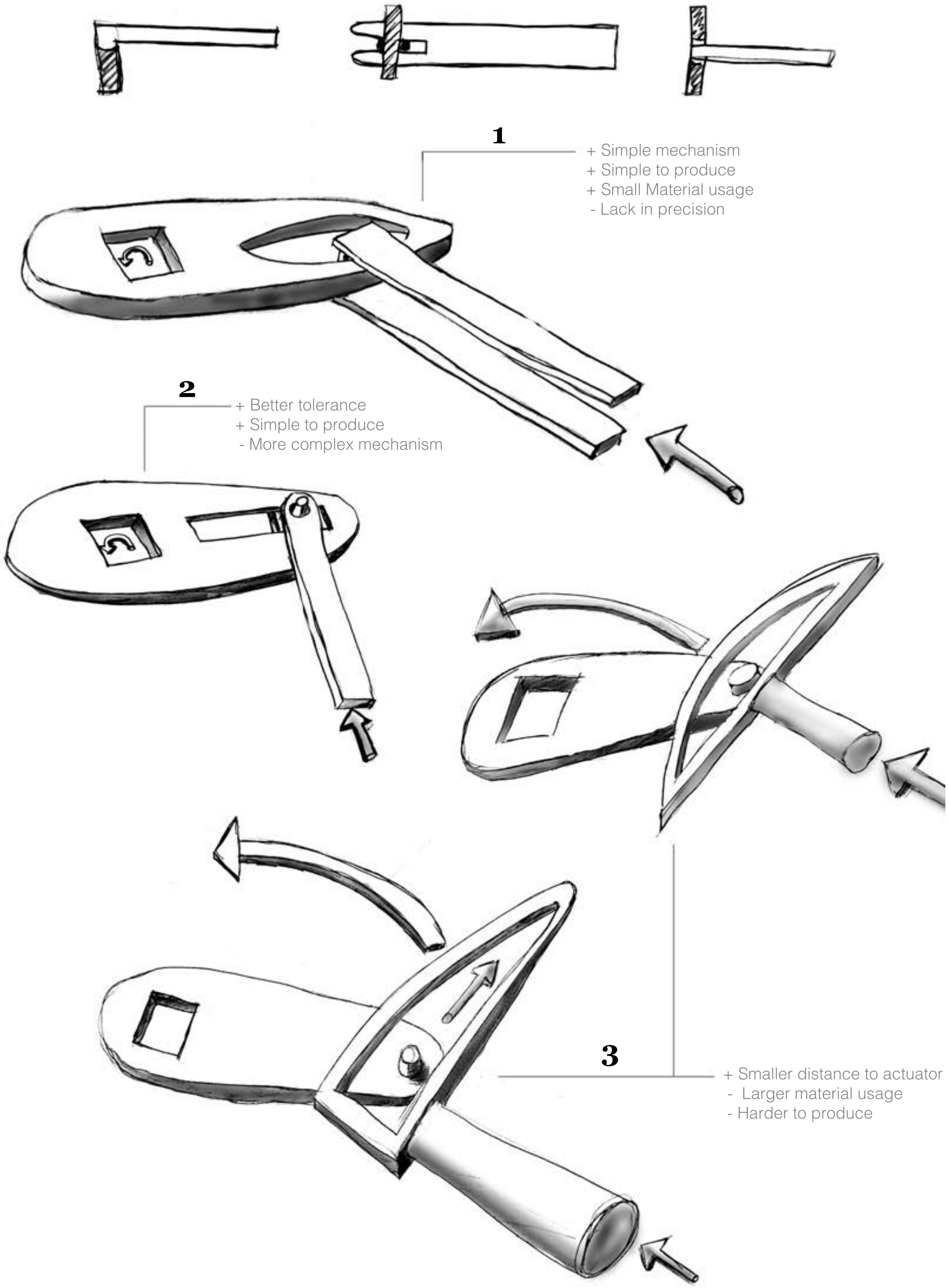


Figure 5.6: Sketches of concepts

The features that is most important is the simplicity of the mechanism and production in addition to a small material usage. This reduces the cost and at the same time increases the rigidity, which is beneficial as it is placed in a harsh environment with limited possibilities of maintenance. The solution shown as number one in figure 5.6 is the solution chosen for its simplicity. The main disadvantage with this solution is its lack of precision. This is not a major concern since the clearance when opening and closing a valve should be sufficient and will not suffer for slack in the mount slot.

A slide is cut in the back of the mount with a M6 stainless steel socket head cap screw through the gap, clamping the mount to the stem at the desired height. The tap- and clearance drill dimensions is found in appendix F. The total friction torques is at its largest when the valve is opened, and is therefore the position calculated for.

Finite element analysis is used to tweak the dimensions by strengthening the weak sections, and reducing the strong section. To accommodate the changing intersection angle of the actuator and the tangent to the stem, the slot is not made straight. Instead it is made with an arc. In the first version this is a quarter of a circle, but in the second the slot is only a portion of that, such that the it gives the end a more correct angle to the stem. The resultant force has a more parallel direction to the tangent of the stem with these curvatures. The portion of the force that is not parallel, does not contribute with moment to the stem. Both arcs are estimated by testing the motion with a 3D modeled actuator and contact sets to simulate their behavior.

The fist version, shown in figure 5.7 was designed for simplicity and for its low material and space usage. This makes the production and material costs as low as possible. Since the moment is at its highest at the stem center and decreases towards the force, version one is designed with a larger amount of material around the stem, giving this section a larger moment of inertia. Since the shaft can not be placed to close to the stem center, this versions has to start and end at an angle. In the FEM analysis, an angle of 45° to the valves x-direction is used (length axis). The actuator therefore has to deliver a larger force to give the stem the necessary moment from the resultant force. This means that the shear force will have a larger importance in the stress analysis, and more importantly that the actuator has to be larger and in turn more costly. For these reasons the first version was discarded over the second.

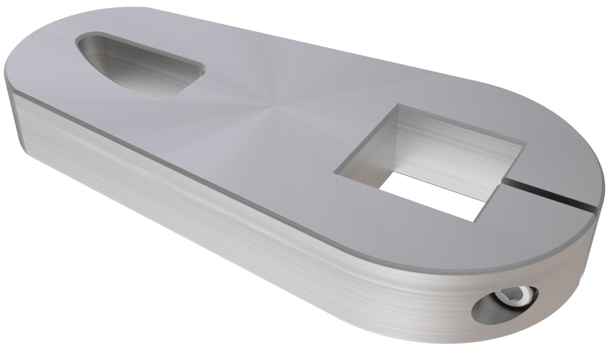


Figure 5.7: Render of mount - version one

The second version shown in figure 5.8 was made to improve and innovate on the problems of the first version. The largest moment is as previously stated when the valve is fully opened, it decreases as the valve closes and is at its lowest when the valve is closed. This means that the actuator force should come at a 90° angle at the start, which makes a equal but opposite directed resultant force parallel to the tangent of the stem. To achieve this without making the shaft operate closer to the stem, a bend is made to the mount such that the end of the mount is not aligned to the stem. This gives the mount the ability to start perpendicular and end parallel to the valves x-direction. The slot was also narrowed, giving it a small increase in precision. The only downside this version has to the first is its larger material and space usage, but this increase is of a small scale, and will therefore be neglected.

Figure 5.8: Render of mount - final version

Stress on mount

The total torsion is in the closed position approximately $1.5 \cdot 10^6$ Nmm and in the open position $1.9 \cdot 10^6$ Nmm. Since the torsion is at its highest in the open position, this value is used in further calculations.

The highest moment will occur with the center line of the stem. The moment of inertia is calculated in equation 5.1 through 5.6, where $r = 19$ mm is the width of the stem divided by two. $a = 20$ mm is the mount thickness and $b = 26$ mm is the width of each part in the section. This is shown in figure 5.9. The maximum allowed stress $\sigma_a = 136$ N/mm^2.

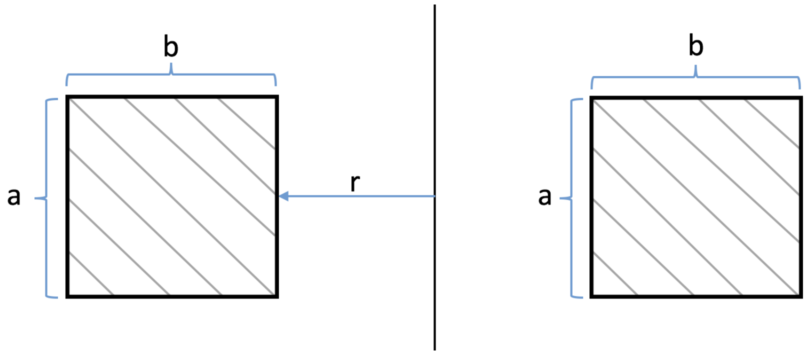


Figure 5.9: Illustration of a and b

$$I_x = \sum \left(I_i + A_i \cdot d_i^2\right) \tag{5.1}$$

$$I_1 = I_2 = I = \frac{1}{12} \cdot a_1 \cdot b_1^3 \tag{5.2}$$

$$A_1 = A_2 = A = a \cdot b \tag{5.3}$$

$$|y_1| = |y_2| = y = r + \frac{b}{2} \tag{5.4}$$

$$\bar{y} = 0 \tag{5.5}$$

$$I_x = 2 \cdot I + 2 \cdot A(\bar{y} - y)^2 = \frac{5}{12} \cdot a \cdot b^3 + 722 \cdot a \cdot b + 38 \cdot a \cdot b^2 \approx 1 \cdot 10^6 \text{ mm}^4 \tag{5.6}$$

$$\sigma = \frac{T_{tot}}{I_x} \cdot (r + b) \approx 86 \text{ N/mm}^2 \tag{5.7}$$

$$\frac{\sigma \cdot 100\%}{\sigma_a} \approx 63\% \tag{5.8}$$

The mount is also tested in FEM analysis and the von mises stress result is shown in figure 5.10. The peak stress was found to be in the slot corner. There were also stress peaks in the corners of the square hole,which is known as stress concentration, an increase in stress due to unevenly distributed forces where there is a sudden change in geometric shape.

With the length from the actuators start position to the center of the stem being approximately 130 mm the force necessary is $F = T_{tot}/L = 12$ kN. In the FEM analysis on the other hand, the necessary force to generate the right moment was found to be 14.5 kN. This may have a correlation with the offset from the center line, or from the slight curvature of the slot. The force in the FEA is the minimum force the actuator has to deliver.

When ignoring stress singularities, the von mises stress is around $70 - 100$ MPa, which gives a utility of around 80%, see appendix N. The mount could be better optimized, but with the singularities a more conservative approach is taken. The conclusion is that this mount will handle the forces applied.

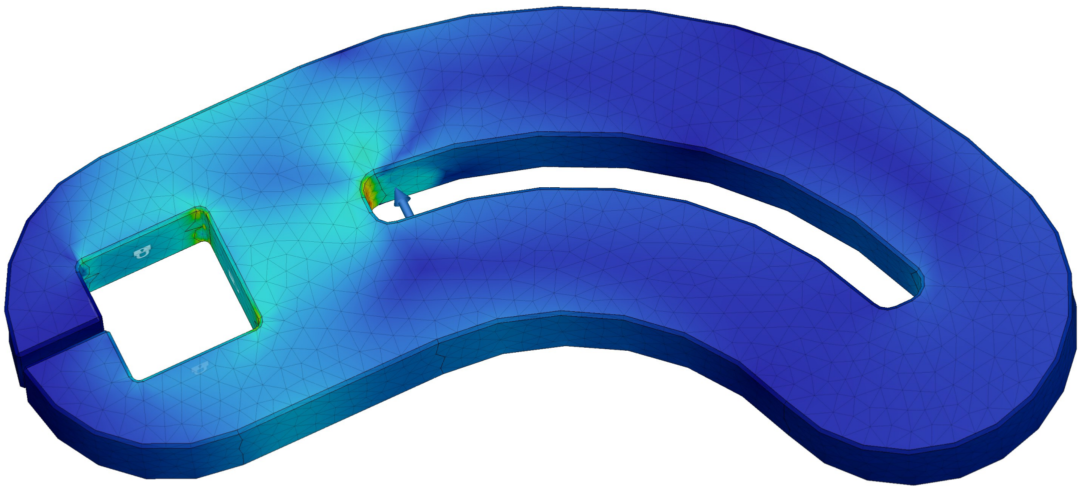


Figure 5.10: Von mises FEM result for mount

Bolt stress

The bolt that is chosen for the mount is an M6 stainless steel socket head cap screw made of Metric Type 316 Stainless Steel—DIN 912-A4, see 2D drawings on page xxii. The material has a yield strength of $\sigma_y = 205$ MPa, see appendix G. This bolt is mostly loaded with tension, and the tensile area is based on the the average of the minor and pitch diameter. [20] The pitch is set to $P = 1$ mm giving the bolt a tensile stress area shown in equation 5.11.

$$d_p = d - (0.649519 \cdot P) = 5.35 \text{ mm} \tag{5.9}$$

$$d_r = d - (1.226869 \cdot P) = 4.77 \text{ mm} \tag{5.10}$$

$$A_t = \frac{\pi}{16} \cdot (d_p + d_r)^2 = 20.1 \text{ mm}^2 \tag{5.11}$$

In most cases a preload is used to give the bolt a clamping force. The friction created between the threads, head and the main component makes sure that the bolt do not come lose under vibration. High preload tension also increases the strength of a joint. The preload is different for reusable and permanent bolts. In this case the bolt is assumed to be permanently installed.

The preload is calculated with the proof strength σ_p. For this material the proof strength is unknown and is therefor estimated. This is done with equation 5.12, and further used to find the preload for the permanently installed bolt in equation 5.13.[20]

$$\sigma_p = 0.85 \cdot \sigma_y = 174.25 \text{ MPa} \tag{5.12}$$

$$F_i = 0.9 \cdot A_t \cdot \sigma_p = 3200 \text{ N} \tag{5.13}$$

The torque required to tighten the bolt is dependent on the constant K which depends on the bolt material and size. Using the table of friction coefficients for stainless steels in appendix H [21]. Assuming that the bolt and mount has equivalent properties to a A2 material grade, and a high resilient of connection the friction coefficients is found to be $\mu = 0.26$ in the thread and $\mu = 0.35$ under the head. Although these values are specifically for hex head screws, they are assumed to be close to the values for socket head cap screw. The corresponding K value is found in appendix I [22] to be $K \approx 0.383$. From this the torque needed is calculated in equation 5.14.

$$T = K \cdot F_i \cdot d \approx 7400 \text{ Nmm} \tag{5.14}$$

This clamping force gives a friction force between the mount and the stem. This will be dependent on the friction coefficient between the two parts. In this case it is steel against steel, giving it an approximated friction factor of $\mu = 0.5$. The force needed to move the mount can then be found using equation 5.15. Both forces are illustrated in figure 5.11.

$$F = F_i \cdot \mu = 1600 \text{ N} \tag{5.15}$$

The equivalent mass is $m = F/g \approx 160$ kg, were g is the gravitational acceleration. This is the weight that can be placed on the mount without displacement. This should be sufficient to prevent any problems from its own weight and any debris hitting the mount in subsea environments.

The bolt will only be affected by tension and the only force acting on it is the preload. This means that the stress is simply force over tensile area, shown in equation 5.16. As described in section 1.7, the safety factor is 1.2 for bolts. From this the allowed stress and utilization is calculated in equation 5.17 and 5.18.

$$\sigma = \frac{F_i}{A_{tsa}} = 160 \text{ MPa} \tag{5.16}$$

$$\sigma_a = \frac{\sigma_y}{\gamma} = 171 \text{ MPa} \tag{5.17}$$

$$\frac{\sigma}{\sigma_a} \cdot 100\% = 93\% \tag{5.18}$$

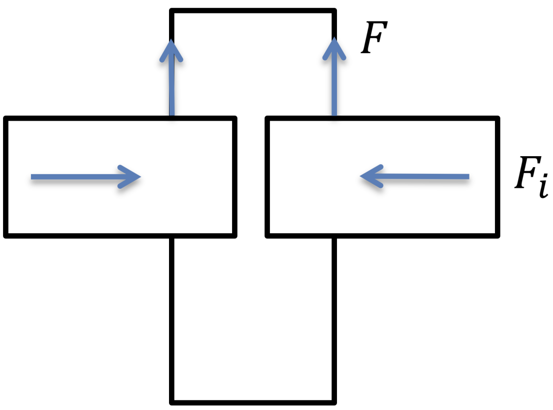


Figure 5.11: Preload and friction force

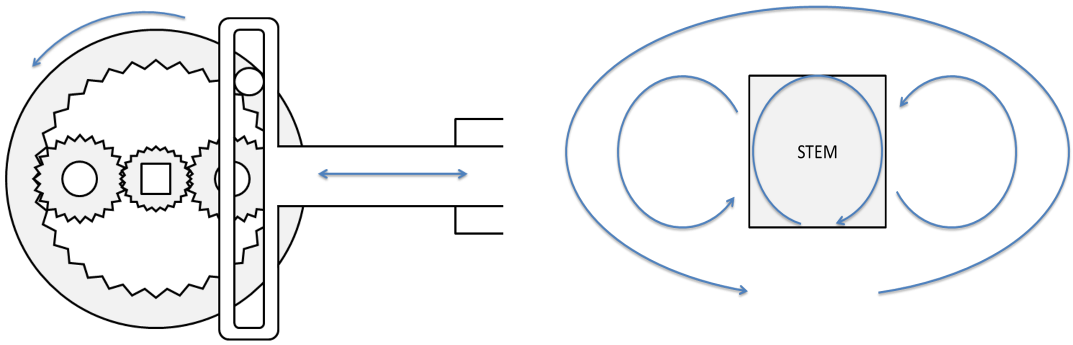
5.4 Alternative design

If a situation requires a smaller arm and/or a higher safety factor, the solutions below will cancel out the resultant force on the stem. The main disadvantages with these solutions is the complexity and cost.

The goal will be to design a mount for only one actuator without exerting a resulting force on the stem to reduce the stress. The mount should take up as little space as possible, and be mounted to the valve itself. It has to work in the designated environment for the expected lifespan of the valve. Three solutions are presented with their advantages and disadvantages.

The circular gear mechanism is based on the slider crank principle. The actuator with its linear movement makes the crank rotate an outer gear. Inside the outer gear there are two gears on each side running on the inner edge shown in figure 5.12a. These are connected to the valve house and will derive the center gear with two equal forces, working in opposite directions. These will cancel out and the center gear will only be affected by the moment generated by the forces. The forces is shown in figure 5.12b.

This solution is self-aligning which makes it rigged. The actuator can be mounted close to the gears which makes this a compact solution. The moment is scalable without increasing the total size. This is done by increasing the size of the center gear while decreasing the size of the two gears on the side. In addition it is simple to produce, or order from a third party. The disadvantages with this solution is its need for several bearings including a thrust bearing which can make it a more costly solution. It also is depending on the tolerances sensitive to debris.

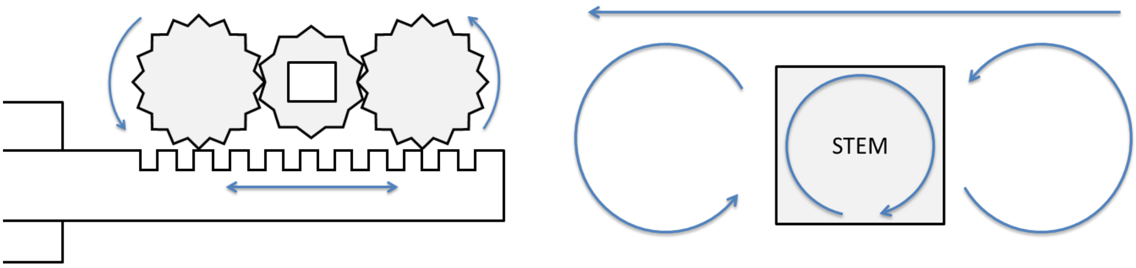


(a) Circular gear mechanism　　(b) Forces acting

Figure 5.12: Illustration of the circular gear mechanism

The linear gear mechanism is based on the "rack and pinion" principle. The outer gear is in this solution a "rack" which is on the end of the actuator. The two side gears must be larger then the center gear, or the rack and side gears must operate above it. Both gears receive a force from the "rack" which in turn works on opposite sides and directions on the center gear, illustrated by figure 5.13.

This solution is one of the more simpler solutions with few parts. The downside is that it still has to have two bearings, and it loses the rigidity of the circular gear mechanism. Because hydraulic actuators have round shafts, the "rack" can not be contracted inside the actuator, resulting in a longer distance from the stem to the actuator. This may become a problem since the actuator should be mounted to the valve.



(a) Linear gear mechanism (b) Forces acting

Figure 5.13: Illustration of the linear gear mechanism

The "wrench" mechanism is based on the idea of an independent wrench. It will be mounted to a support structure on the valve. The "wrench" will be fitted in a bearing fixed to the support connected to the valve body with the valve lid bolts. This will take up the resulting forces from the actuator, and the "wrench" will only deliver the moment through to the stem. The concept is shown in figure 5.14. The advantages with this mechanism will be that it takes up a small space and it contains few and simple parts. A downside is that it requires a large bearing, and if enough slack is present the force can create a moment on the stem.

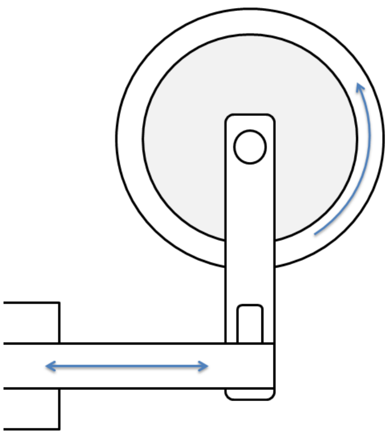


Figure 5.14: Illustration of the "wrench" mechanism

6 Discussion

The TBV96S trunnion ball valve is in this case used for controlling the flow of mud and had problems in both the bearings and gaskets. Through examination of the failed valve, the most likely cause was found to be faulty assembly and usage. The group was therefore tasked to make the changes needed such that the valve could be used in the designated assignment.

Mud is a substance with hard particles, this eroded the valves gaskets creating leakage between the seat and ball. The material used was found to be of a lower hardness rating and was for that reason changed. The new material assigned for this valve was PEEK, a high performance thermoplastic. PEEK was chosen for its higher hardness rating and should prevent the extensive wear from the fluid. Other fluoropolymers and elastomers was to soft for the intended usage. The cost is higher for PEEK, and if through physical testing it is found that the material do not need the highest hardness rating, the slightly softer PCTFE should be used. Since the mud is made on site with different formulas for each case and phase, the hardness and wear it produces can differ considerable. Additionally the dimension selection is mostly based on the experience and knowledge from the engineers at Valves of Norway. For these reasons the valve and gasket should be tested according to the ISO 10423 standard discussed in section 1.7. For situations where a low leakage rate is key, it is important to note that softer materials commonly gives a better seal.

The other challenge in this valve was the bearings, they were broken and this was the main reason that the valve were returned to the company. The bearing failure was probably caused by a the faulty assembly in addition to a large actuator force. Since the actuator force was unknown, the problem was solved through a material change. Plain bearing was chosen over roller bearing especially for its maintenance free options which is demanded in subsea environments. Some innovation is done with maintenance free roller bearings, but these loose many of the positive aspects due to the material that is used.

The chosen material was the woven PTFE reinforced and bonded to a metal backing from TENMAT called FEROGLIDE. TENMAT was the preferred supplier for Valves of Norway, and others where therefore not researched. PTFE was chosen because it had the necessary allowed PV factor, meaning it could handle the load and velocity on the bearings, in addition to its temperature range and wear rate. The wear rate and friction coefficients from TENMAT is from an unknown sample size and with an insufficient description. The values are therefore rounded up to accommodate for the uncertainties. The dimensions was chosen to make the changes on the stem and trunnion as subtle as possible. Only small changes was done on the diameters of the stem, trunnion and ball hole.

The PV factor is dependent on the forces on the bearings. Identifying the magnitude of the force on each bearing was challenging due to the complexity. There were more unknowns than equations, and finite element analysis was therefore used to find the reaction forces from the bearings. The result was then compared with the hand calculated ratios, the difference was small and the results was taken as correct values. The analysis and PV factor was only calculated for the previous bearing dimensions which could give an error, but because of the minor changes this is likely irrelevant.

The changes made on the bearings and gasket affects both the trunnion, stem and springs. In advance the actuator force was undefined and the maximum allowed force was therefore found. The worst case scenario was chosen for the direction of the actuator force (against flow), if other directions are used the actuator force could be of a larger magnitude and the stress distribution could be different.

Friction torques and stresses on the different sections was calculated with hand and FEA calculations. Unconventional formulas has been used for many of the friction torque calculations, this in agreement with the engineer at Valves of Norway. Although the formulas used are believed to be of a sufficient accuracy, a suggestion for further research is to find more well documented and reasoned formulas. Additionally the friction coefficients from the bearings is assumed to be 0.1 because they run in mud. This is possibly a high value, and if testing is done it may be reduced significantly.

The result showed that only the trunnion, trunnion bearing and upper stem bearing was close to their material limit and these should either be increased or the other sections should be reduced to make the design more optimized. Increasing the parts will give the valve the ability to handle a larger actuator force and reduce the arm length if this is desirable. The valve as is will need a minimum arm length of 75 mm with a safety factor of 1.5 on the friction torques. When designing the mouth this value did not create any problems. On that basis a suggestion might be to reduce all components to optimize the material usage of the valve.

Several solutions for the mount were proposed, including mounts that will reduce the resultant force on the stem to zero and thereby increasing the safety factor drastically. The concept that was further designed on was the lever principle for a hydraulic actuator, assuming access to an hydraulic systems. In later years electric actuators have been manufactured for subsea use. These actuators deliver a different set of solutions for mounts, and if this is used in the application, a new mount should be designed.

A bend was added to the mount to give the ability to start perpendicular to the stem, such that the force tangent to the stem was highest at the starting point were the friction torques was largest. The part was 3D modeled along with an actuator, and these were used to simulate their behavior. This type of simulations is cost efficient, and can reduce the time used for prototyping to a great extent.

When stress analyzing the mount, the height was set to 40 mm from the top as an estimate. The height is actually adjustable and if placed higher the result should be re assessed. The mount was made an amount to strong, and further optimizing could be done, but because of singularities it then should be physically tested before use.

7 Conclusion

This report is the bachelor thesis in the mechanical engineering study with specialization in marine structural engineering, and addresses bearing and gasket failures with the valve TBV96S which is constructed by Valves of Norway. The task is given by Valves of Norway a valve producing company located by the west coast of Norway, outside Bergen.

A new material is chosen for both the gaskets and bearings in the valve. This change should be sufficient in avoiding a repeated failure of the valve. The valve is stress analyzed and all components are established to be capable of handling the work pressure of 10 MPa in addition to an actuator force of up to 30 kN. All calculations follows the recommendations from ISO 10423. In case of a situations where a higher safety factor or a smaller arm is needed several proposals for mounts that will cancel out the actuator force are suggested.

This thesis should give a clear and reasoned description of the selection process for the gasket and bearing which can be used in all types of applications, aiding engineers to take a good material choice for further projects. Although other projects may have different priorities, all aspects should have been mentioned. The proposed mount solutions is also believed to be an innovative solutions giving the highest force were needed, but still be simple in both the mechanism and production. The thesis also addresses how to calculate torsion for both uniform and un-uniform geometry, in addition to average and shear stress and lastly moment stress.

A large amount of time should be given to the selection and calculation process, which will greatly reduce the cost in physical prototypes and returns. Digital tools should also be taken to use, they can improve the optimization of a product even before a physical model is made as well as calculating problems with considerable complexity. This thesis shows how digital tools can be used as an extension of the hand calculations to achieve more accurate results and solutions.

Even though with the high complexity of the task and the lack of well documented formulas, the results shows great precision and the group considers the task to be achieved.

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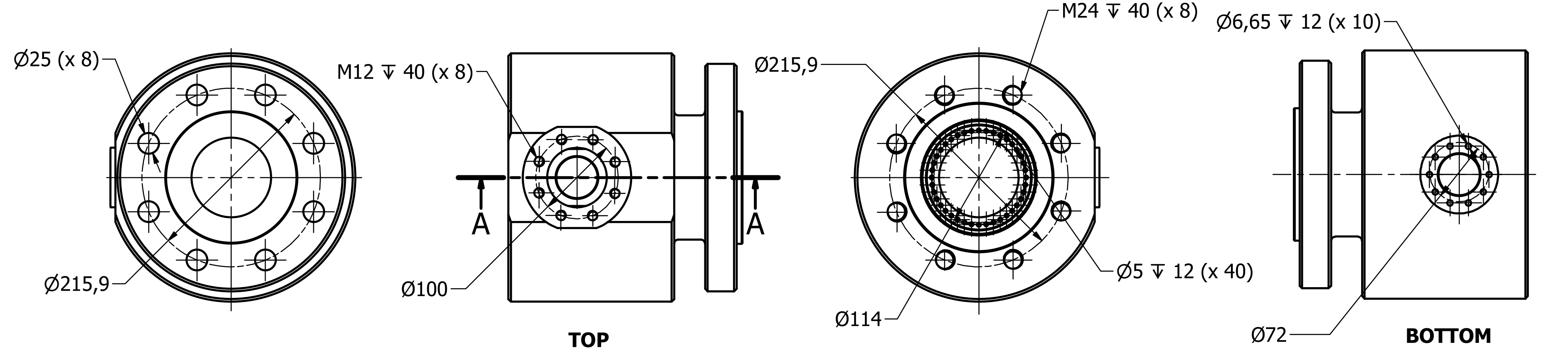

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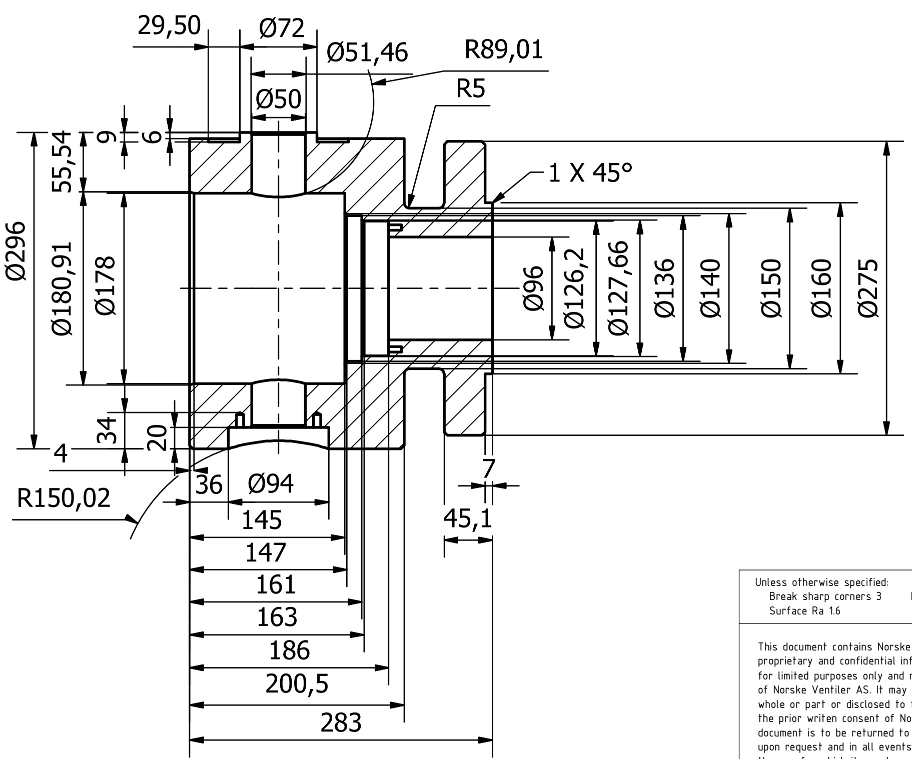

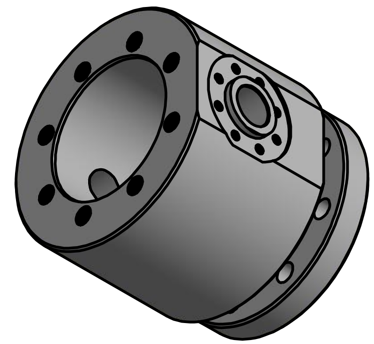

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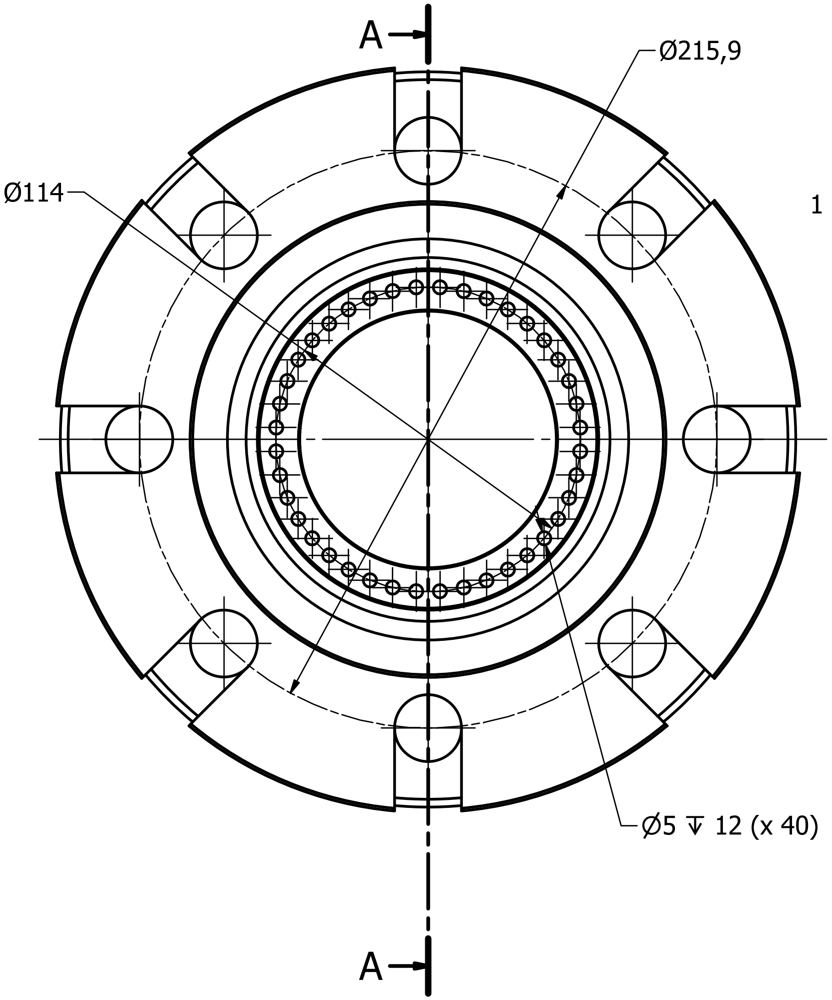

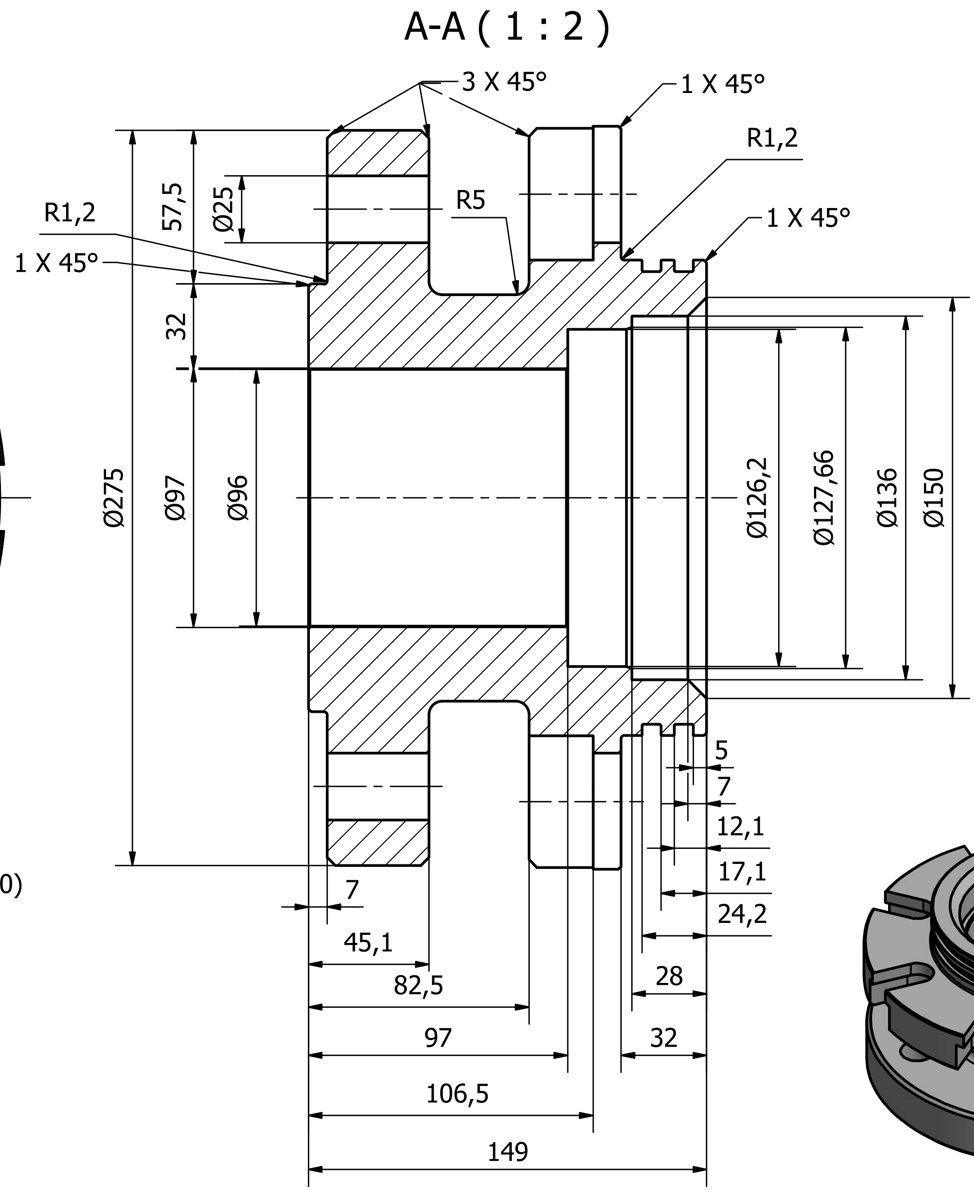

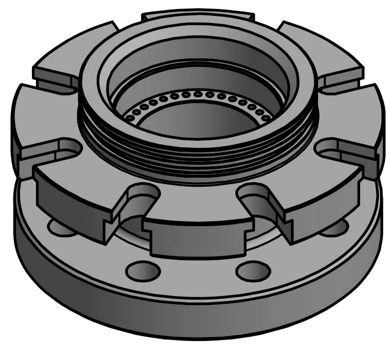

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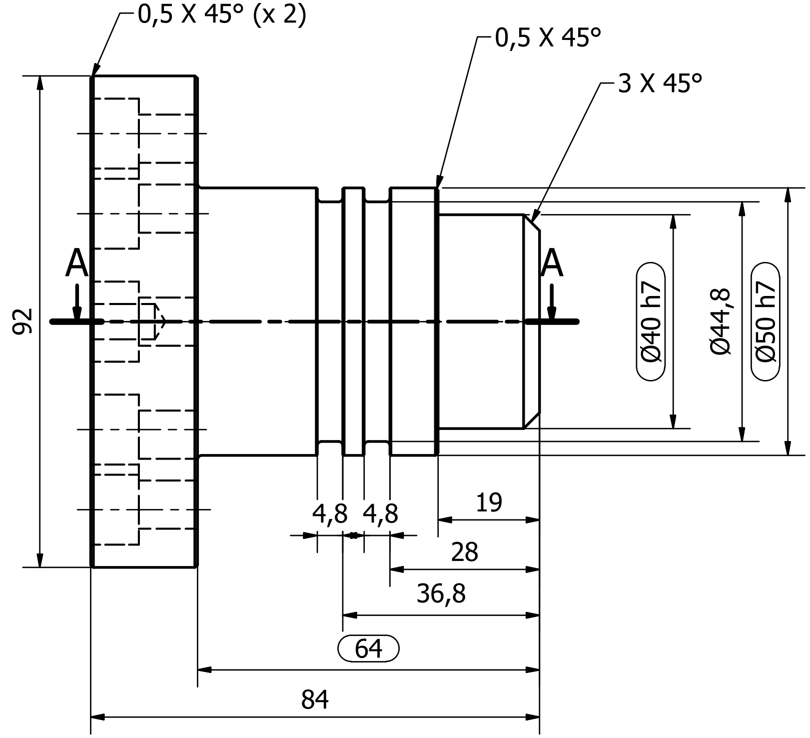

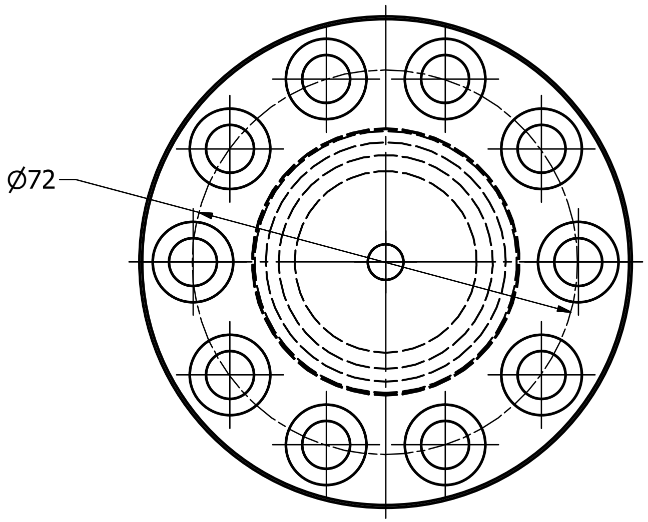

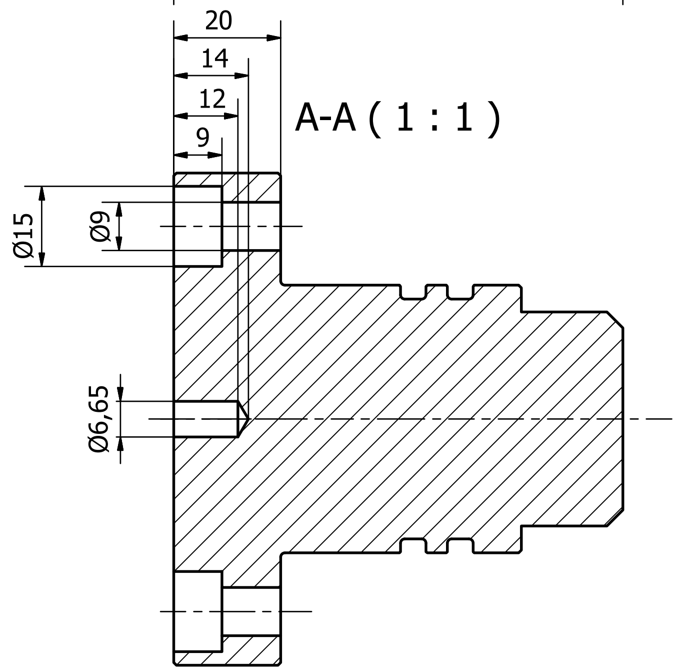

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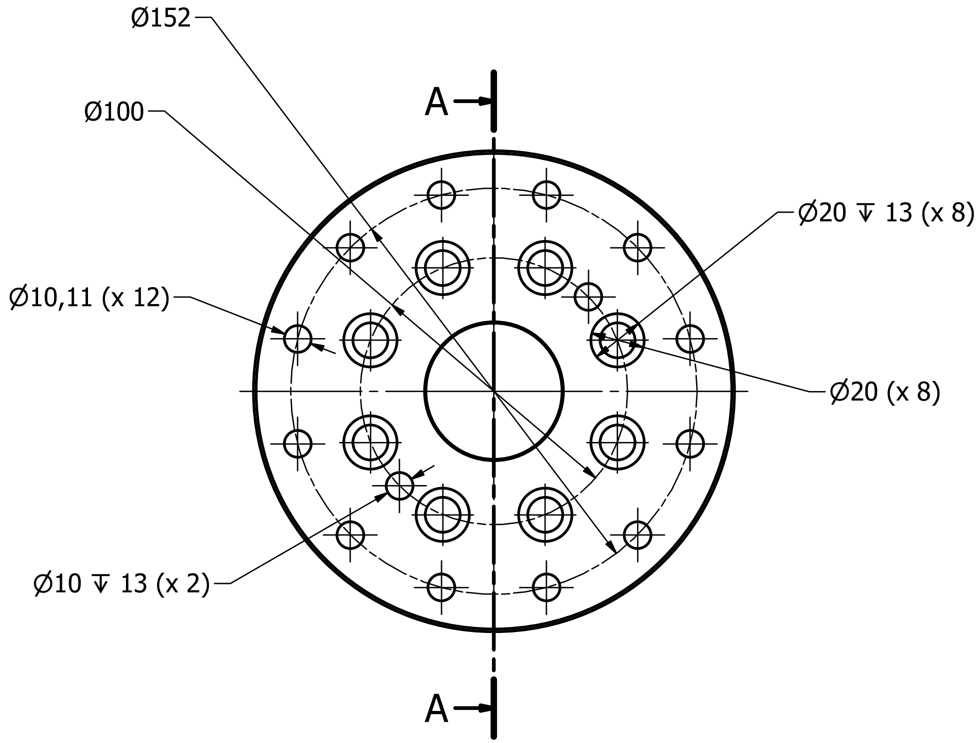

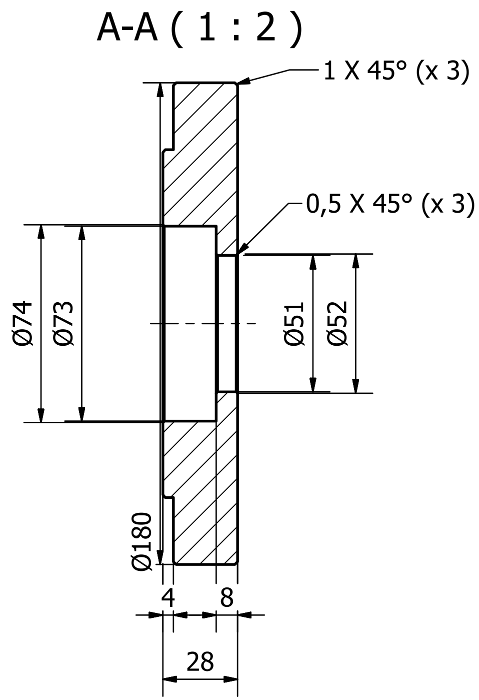

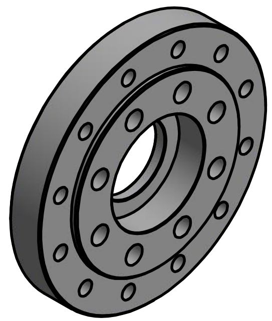

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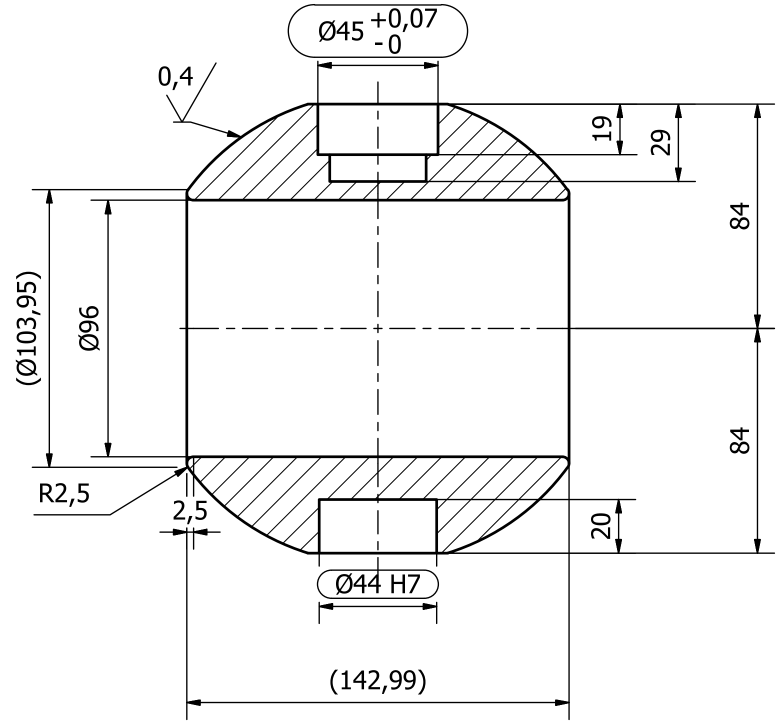
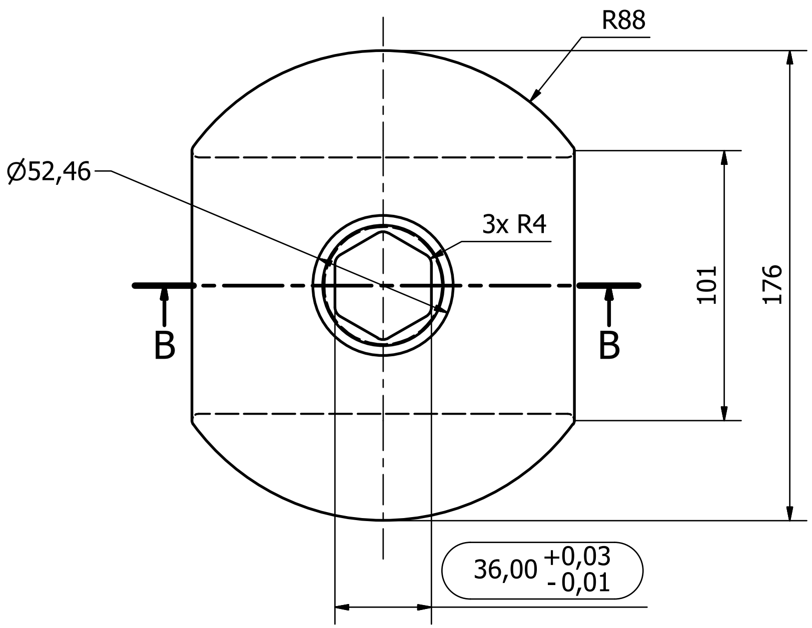
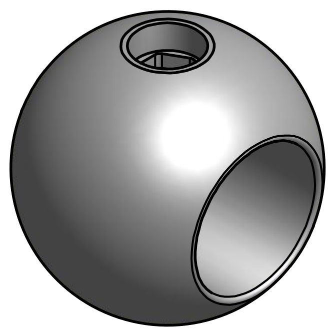
NORSKE VENTILER AS

TITEL	Stem Lid	
DOCUMENT NO.	NV 200000	REV. 1
Scale: 1:2		Size: A3

B-B (1 : 2)







Unless otherwise specified:
Break sharp corners 0.2 | Fillet radii 0.8
Surface Ra 1.6 | Tolerances ISO 2768-f

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Material	A276 S32750		
Weight	12,5kg		
Originator	Helene	Date	15.04.2016
Modified by	Helene	Date	
Checked by		Date	
Approved by		Date	
Design state		Release Date	

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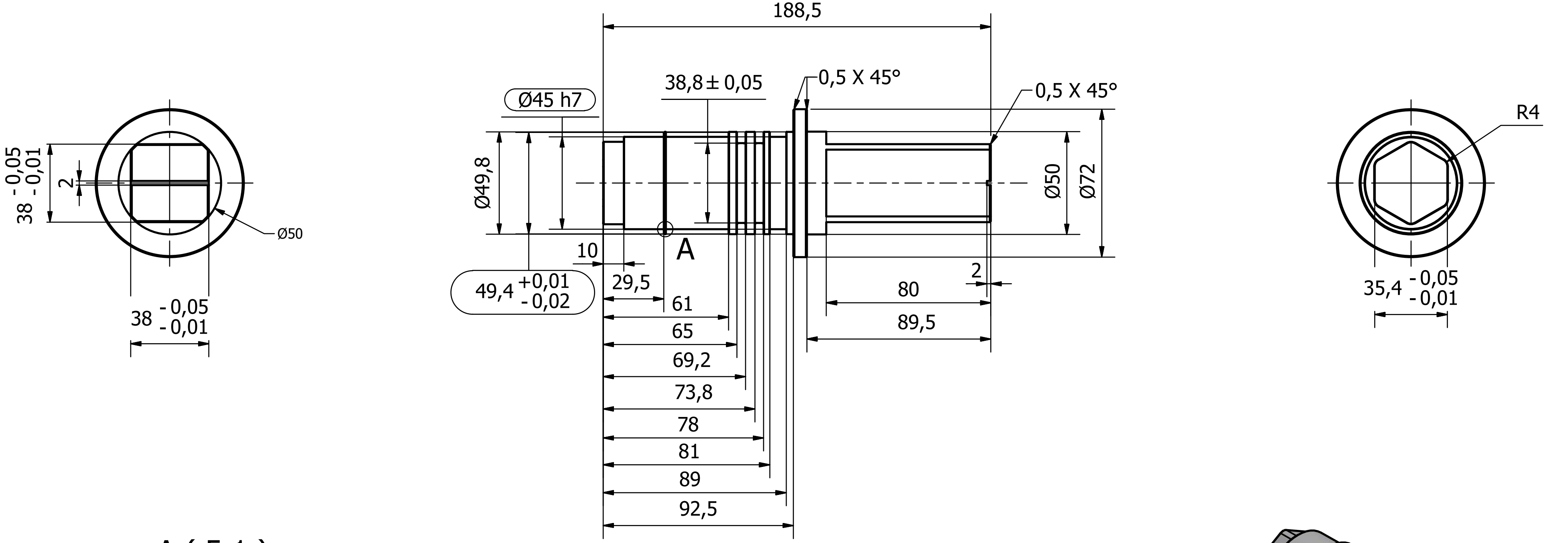
VALVES OF NORWAY
NORSKE VENTILER AS

TITEL Ball

DOCUMENT NO. 20367 | REV. 1

Scale: 1:2 | Size: A3 | Sheet 1 of 1



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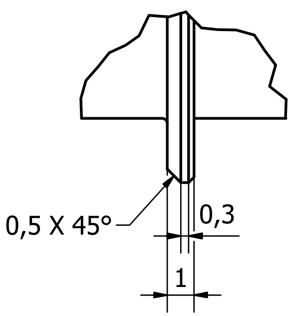

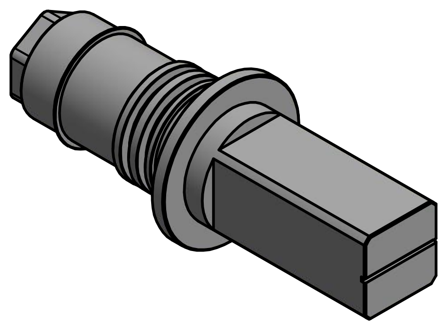

Unless otherwise specified: Break sharp corners 0.2 — Fillet radii 0.8 Surface Ra 1.6 — Tolerances ISO 2768-f	Material: A479 UNS S31600 Weight: 2,5kg	VALVES OF NORWAY NORSKE VENTILER AS
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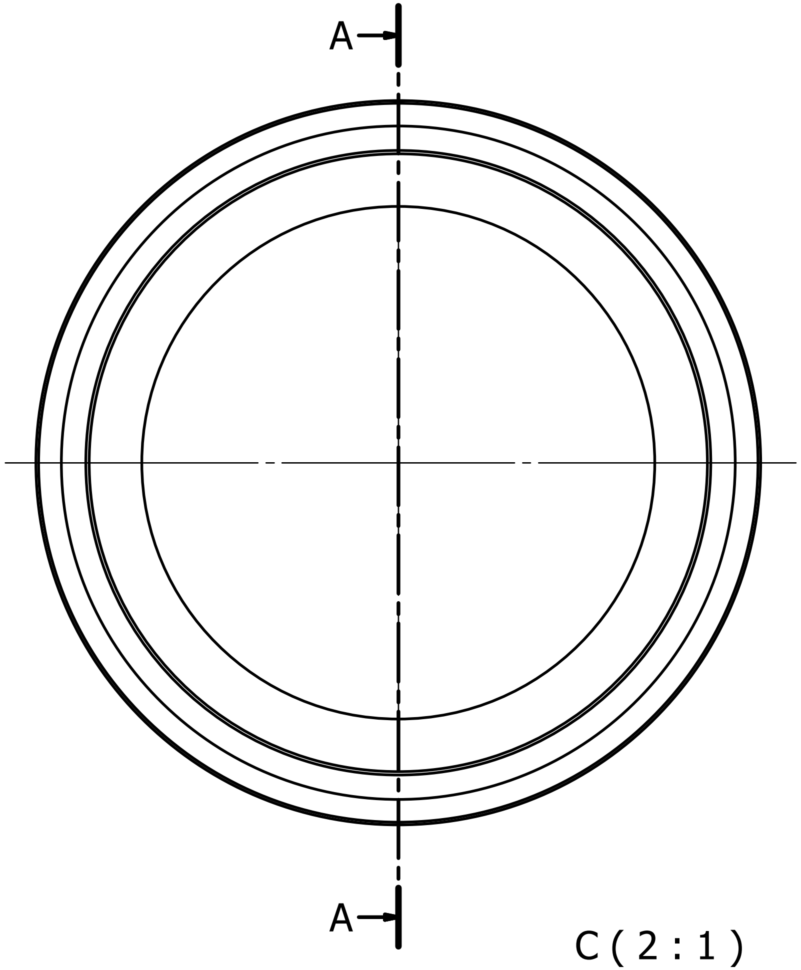
	www.norskeventiler.no	Scale: 1:2 — Size: A3 — Sheet 1 of 1



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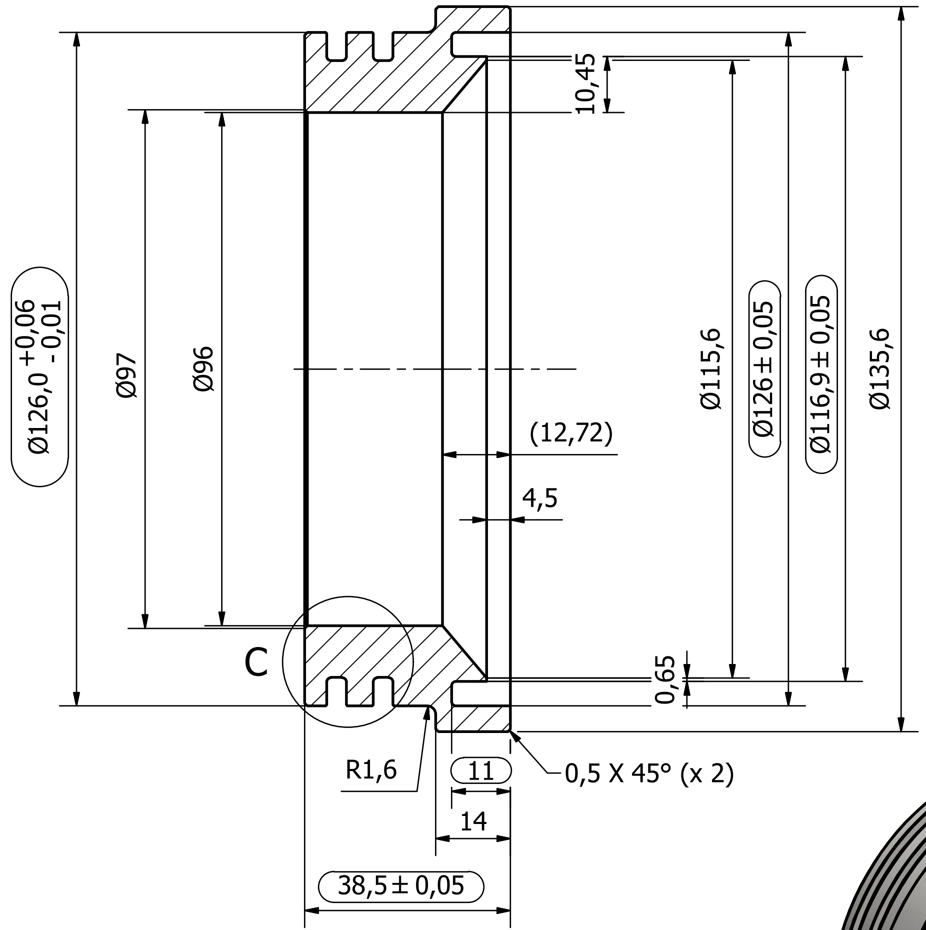
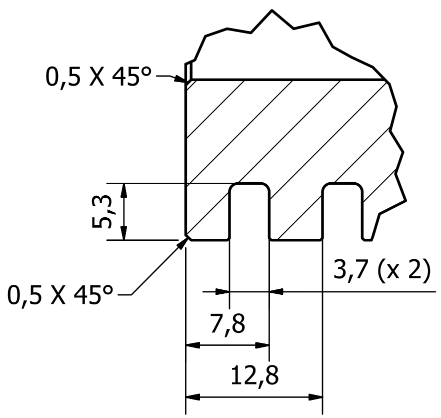


C (2 : 1)



Unless otherwise specified: Break sharp corners 0.2; Surface Ra 1.6; Fillet radii 0.8; Tolerances ISO 2768-f	Material: A276 S32750	VALVES OF NORWAY NORSKE VENTILER AS	
	Weight: 1,3kg		
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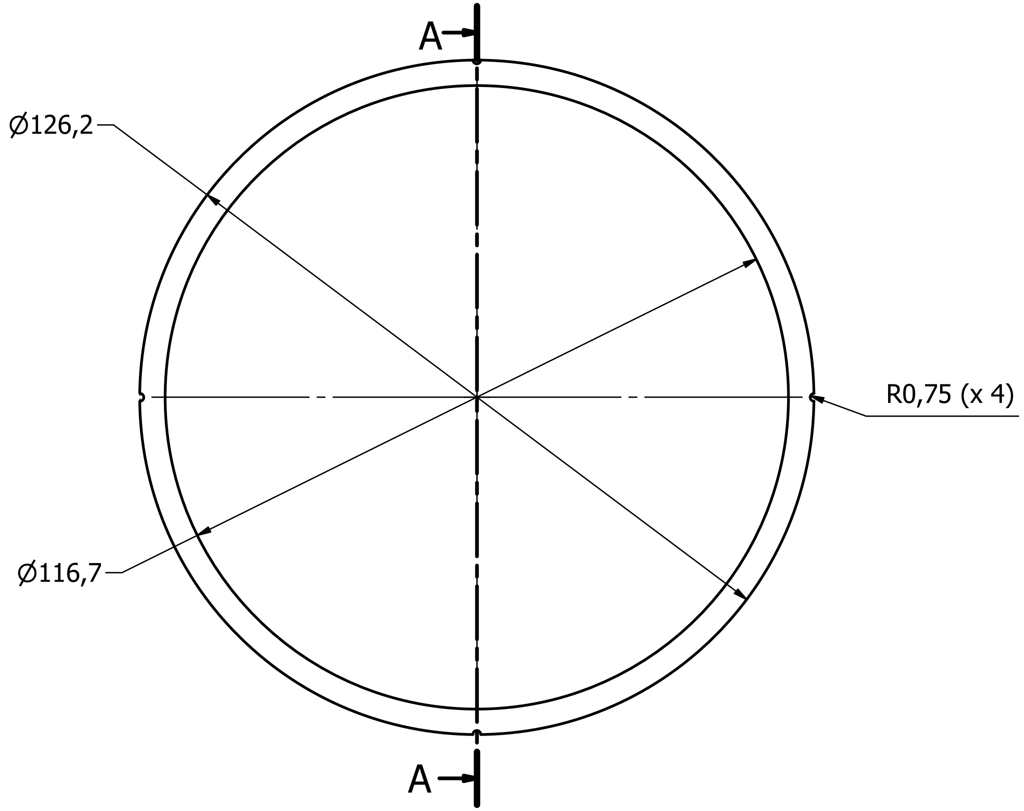
	Modified by: Helene F. S. — Date: 29.04.2016		
	Checked by: Einar K. — Date: 29.04.2016	DOCUMENT NO. NV200163	REV. 2
	Approved by: Håkon M. — Date: 30.04.2016		
	Design state — Release Date		
	www.norskeventiler.no	Scale: 1:1 — Size: A3	Sheet 1 of 1



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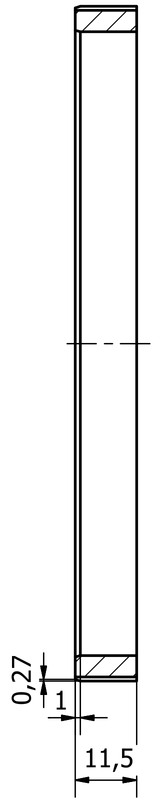

Unless otherwise specified: Break sharp corners 0.2 — Fillet radii 0.8 Surface Ra 1.6 — Tolerances ISO 2768-f	Material: PEEK Weight: N/A		VALVES OF NORWAY NORSKE VENTILER AS
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	Modified by: Helene F. S.	Date: 29.04.2016	DOCUMENT NO.: NV200164
	Checked by: Einar K.	Date: 29.04.2016	REV.: 2
	Approved by: Håkon M.	Date: 30.04.2016	
	Design state	Release Date	
	www.norskeventiler.no		Scale: 1:1 — Size: A3 — Sheet 1 of 1

Bearing Upper Stem

Type: Plain linear self-lubricating

FEROGLIDE

PA1 JOURNAL BEARINGS COILED

Standard Backing Metals:
1) Zinc coated mild steel
2) Stainless Steel (AISI 316)
3) Inconel 625 id

Sizes and Tolerances to DIN1494

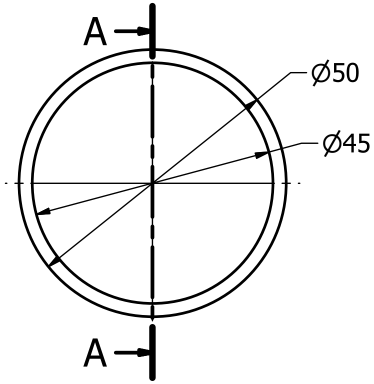
Inner diameter tolerance H10 when installed in housing.

Material: Woven PTFE fabric with strengthening fibers applied to a metal backing.

Part nr.: On request

Supplier: TENMAT Ltd.

For more information see
http://www.tenmat.com/Content/Feroglide



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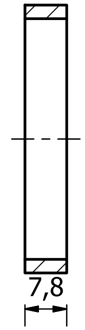

The bearing is bought by TENMAT and therefore this is a dummy 2D drawing.	Material: PTFE		VALVES OF NORWAY NORSKE VENTILER AS
	Weight: N/A		
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	Modified by: Helene F. S.	Date: 29.04.2016	
	Checked by: Einar K.	Date: 29.04.2016	DOCUMENT NO. -
	Approved by: Håkon M.	Date: 30.04.2016	REV. 1
	Design state	Release Date	
	www.norskeventiler.no		Scale: 1:1 · Size: A3 · Sheet 1 of 1

Bearing Lower Stem

Type: Plain linear self-lubricating

FEROGLIDE

PA1 JOURNAL BEARINGS COILED

Standard Backing Metals:
1) Zinc coated mild steel
2) Stainless Steel (AISI 316)
3) Inconel 625 id

Sizes and Tolerances to DIN1494

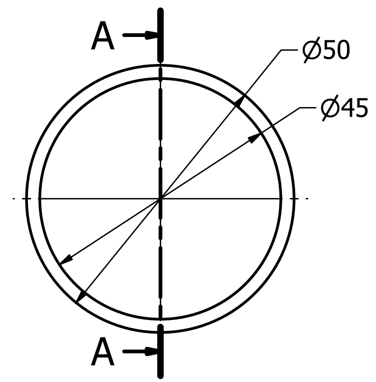
Inner diameter tolerance H10 when installed in housing.

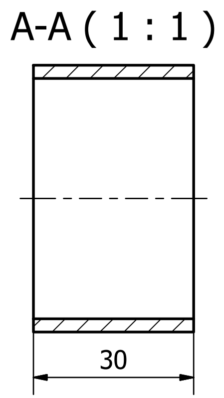
Material: Woven PTFE fabric with strengthening fibers applied to a metal backing.

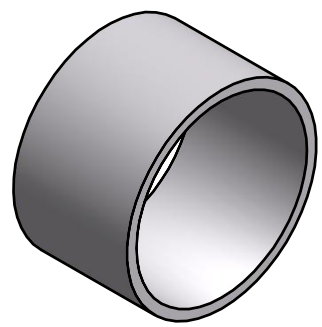
Part nr.: P-A1-45.30

Supplier: TENMAT Ltd.

For more information see
http://www.tenmat.com/Content/Feroglide







The bearing is bought by TENMAT and therefore this is a dummy 2D drawing.	Material: PTFE		VALVES OF NORWAY NORSKE VENTILER AS	
	Weight: N/A			
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	Modified by: Helene F. S.	Date: 29.04.2016		
	Checked by: Einar K.	Date: 29.04.2016	DOCUMENT NO. –	REV. 1
	Approved by: Håkon M.	Date: 30.04.2016		
	Design state	Release Date		
	www.norskeventiler.no		Scale: 1:1 / Size: A3	Sheet 1 of 1

Bearing Trunnion Lid

Type: Plain linear self-lubricating

FEROGLIDE

PA1 JOURNAL BEARINGS COILED

Standard Backing Metals:
1) Zinc coated mild steel
2) Stainless Steel (AISI 316)
3) Inconel 625 id

Sizes and Tolerances to DIN1494

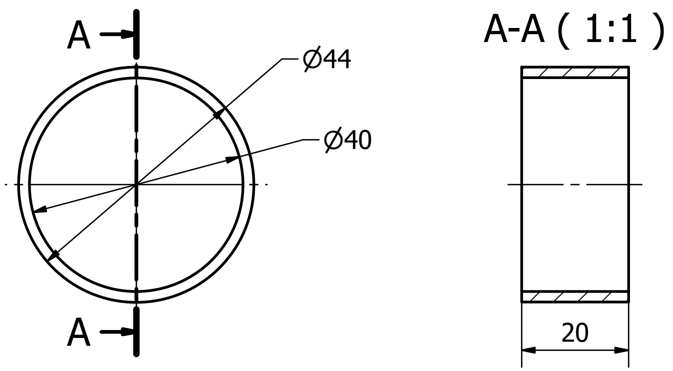
Inner diameter tolerance H10 when installed in housing.

Material: Woven PTFE fabric with strengthening fibers applied to a metal backing.

Part nr.: P-A1-45.20

Supplier: TENMAT Ltd.

For more information see
http://www.tenmat.com/Content/Feroglide



The bearing is bought by TENMAT and therefore this is a dummy 2D drawing.

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Material	PTFE	
Weight	N/A	
Originator	Helene F. S.	Date 23.04.2016
Modified by	Helene F. S.	Date 29.04.2016
Checked by	Einar K.	Date 29.04.2016
Approved by	Håkon M	Date 30.04.2016
Design state		Release Date

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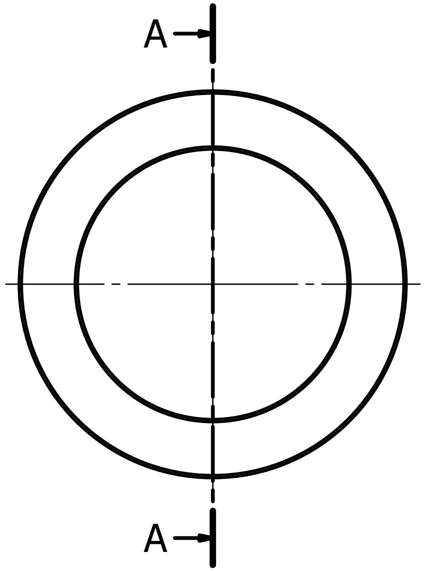
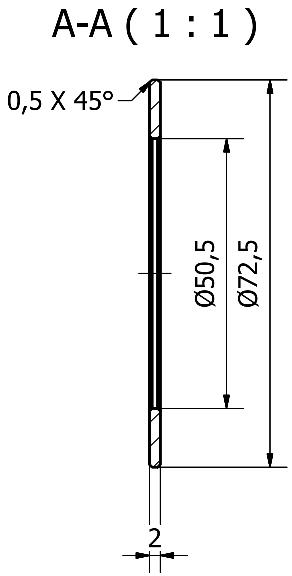
VALVES OF NORWAY
NORSKE VENTILER AS

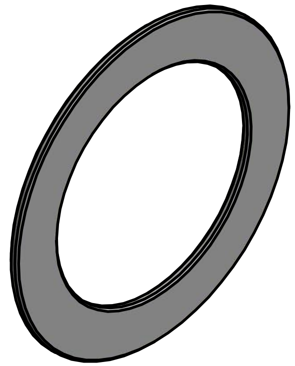
TITEL **Bearing Trunnion**

DOCUMENT NO. –	REV. 1

Scale: 1:1		Size: A3	Sheet 1 of 1







This is a dummy 2D drawing.

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Material	PETP	
Weight	N/A	
Originator	Helene F. S.	Date 02.05.2016
Modified by	Helene	Date 02.05.2016
Checked by	Håkon M.	Date 02.05.2016
Approved by	Helene F. S.	Date 02.05.2016
Design state		Release Date

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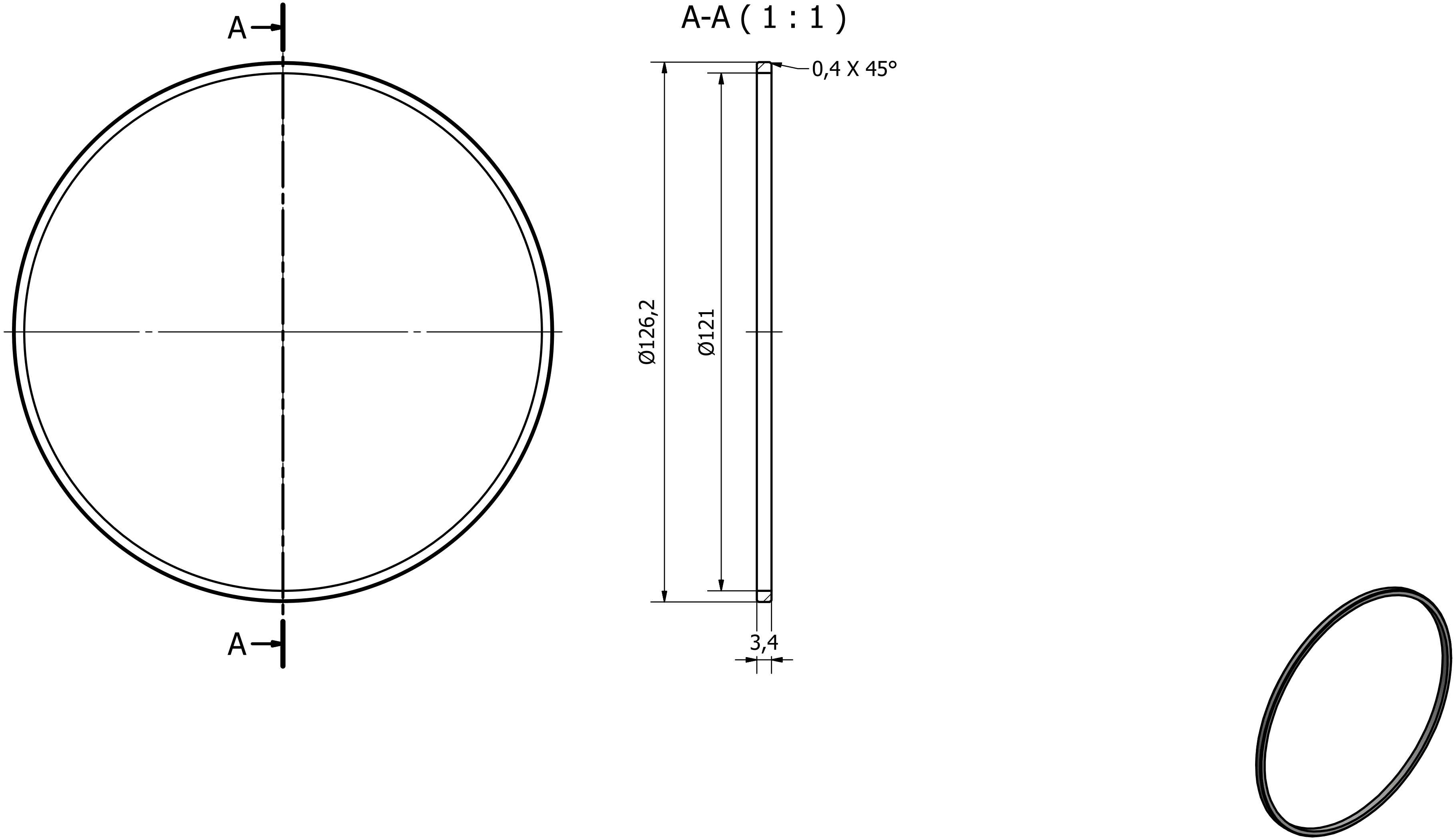
VALVES OF NORWAY
NORSKE VENTILER AS

TITEL: Thrust Bearing

DOCUMENT NO. 20562 | REV. 1

Scale: 1:1 | Size: A3 | Sheet 1 of 1



Unless otherwise specified:
Break sharp corners 0.2 — Fillet radii 0.8
Surface Ra 1.6 — Tolerances ISO 2768-f

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		Date
Material	TFM	
Weight	N/A	
Originator	Helene F. S.	01.05.2016
Modified by	Helene	01.05.2016
Checked by	Håkon M.	01.05.2016
Approved by	Helene F. S.	01.05.2016
Design state		Release Date

VALVES OF NORWAY
NORSKE VENTILER AS

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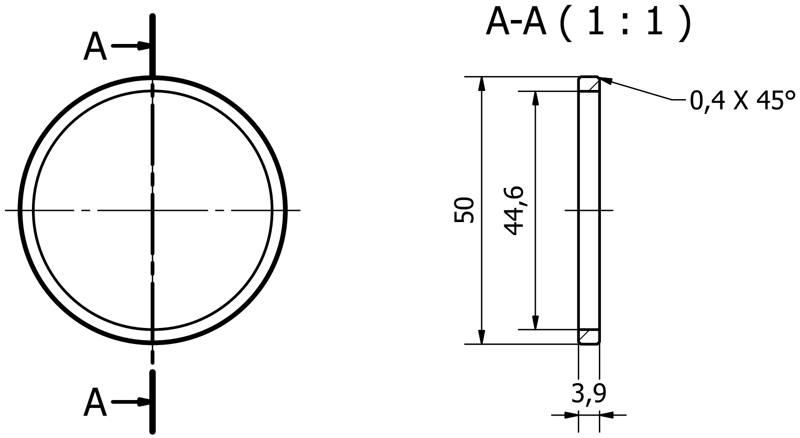
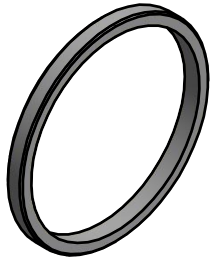
TITEL **Seat Glide Ring**

DOCUMENT NO. 20358 — REV. 1

Scale: 1:1 — Size: A3 — Sheet 1 of 1





Unless otherwise specified:
Break sharp corners 0.2 | Fillet radii 0.8
Surface Ra 1.6 | Tolerances ISO 2768-f

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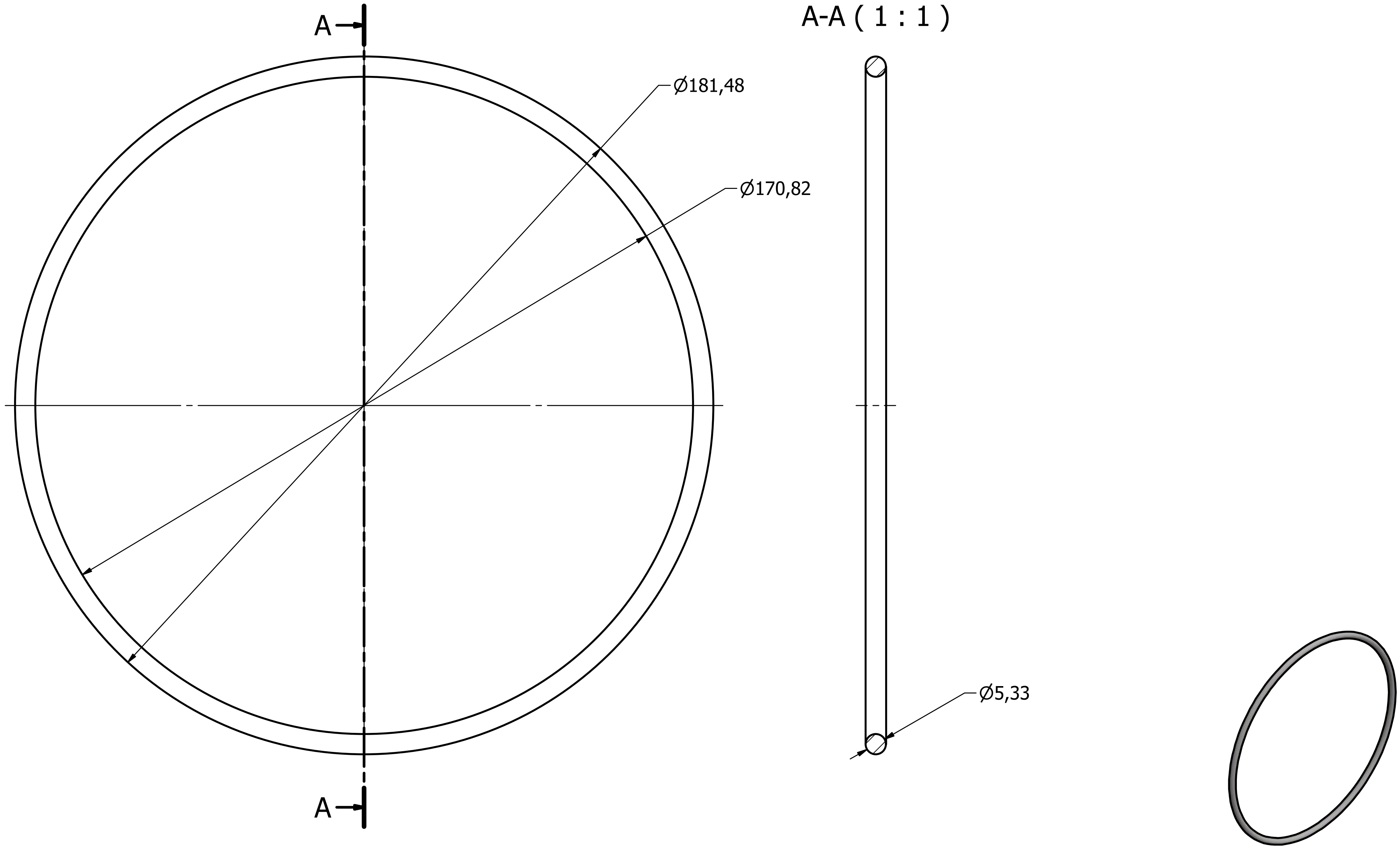
Material	TFM	
Weight	N/A	
Originator	Helene F. S.	Date 01.05.2016
Modified by	Helene	Date 01.05.2016
Checked by	Håkon M.	Date 01.05.2016
Approved by	Helene F. S.	Date 01.05.2016
Design state		Release Date
www.norskeventiler.no		

VALVES OF NORWAY
NORSKE VENTILER AS

TITEL	Stem Glide Ring		
DOCUMENT NO.	20561	REV.	1
Scale: 1:1		Size: A3	Sheet 1 of 1



Unless otherwise specified:
Break sharp corners 0.2 | Fillet radii 0.8
Surface Ra 1.6 | Tolerances ISO 2768-f

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Material	NBR 70	
Weight	N/A	
Originator	Helene F. S.	Date 01.05.2016
Modified by	Helene	Date 01.05.2016
Checked by	Håkon M.	Date 01.05.2016
Approved by	Helene F. S.	Date
Design state		Release Date

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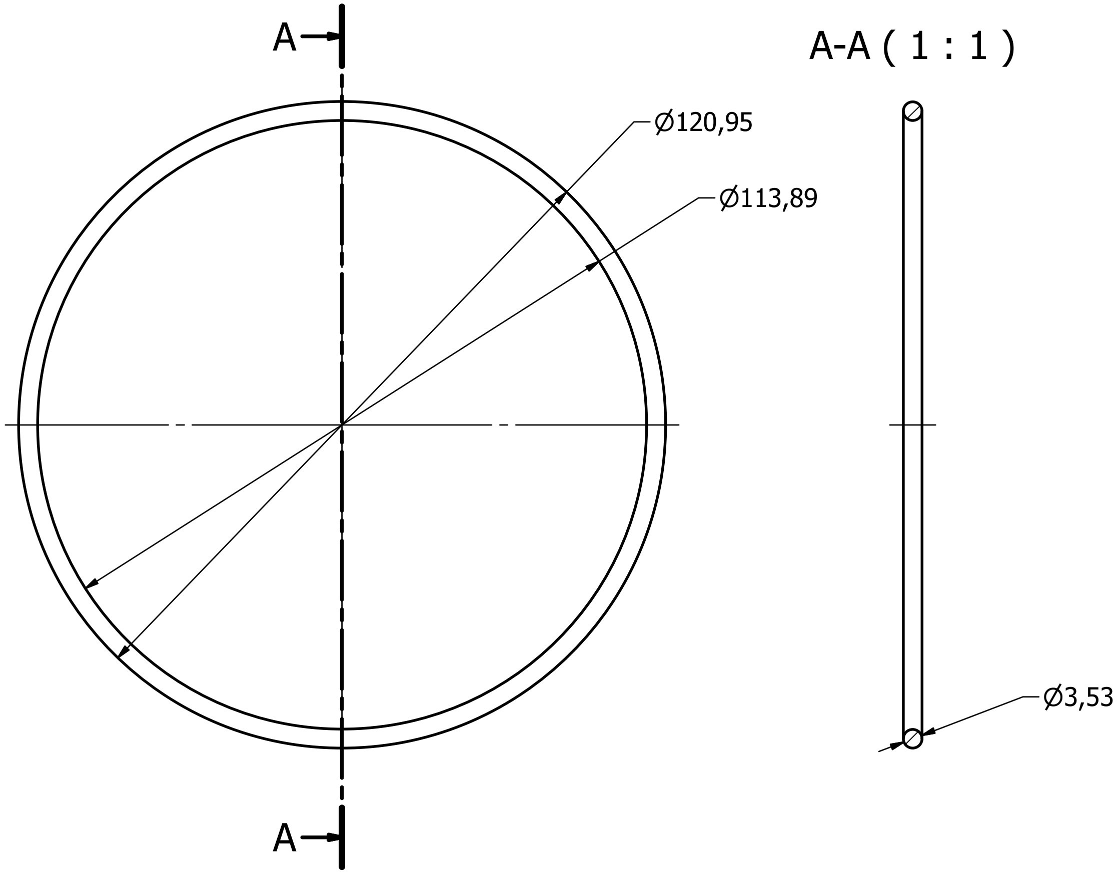
VALVES OF NORWAY
NORSKE VENTILER AS

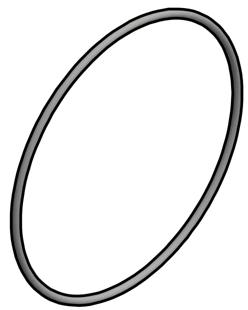
TITEL: O-Ring 170,82 x 5,33

DOCUMENT NO. -

REV. 1

Scale: 1:1 | Size: A3 | Sheet 1 of 1





Unless otherwise specified: Break sharp corners 0.2 — Fillet radii 0.8 Surface Ra 1.6 — Tolerances ISO 2768-f	Material: NBR 70 Weight: N/A

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	Name	Date
Originator	Helene F. S.	01.05.2016
Modified by	Helene	01.05.2016
Checked by	Håkon M.	01.05.2016
Approved by	Helene F. S.	01.05.2016
Design state		Release Date

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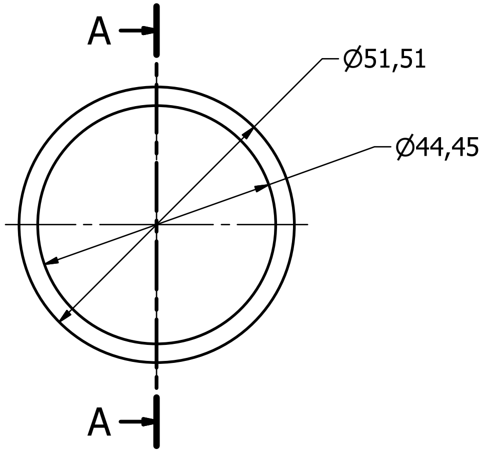
VALVES OF NORWAY
NORSKE VENTILER AS

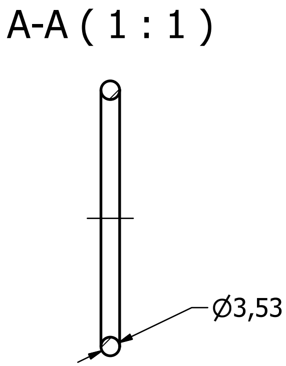
TITEL: O-Ring 113,89 x 3,53

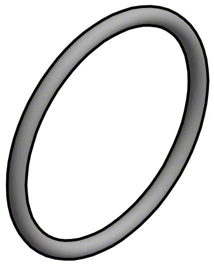
DOCUMENT NO.: -

REV.: 1

Scale: 1:1 | Size: A3 | Sheet 1 of 1







Unless otherwise specified:
Break sharp corners 0.2 — Fillet radii 0.8
Surface Ra 1.6 — Tolerances ISO 2768-f

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Material	NBR 70	
Weight	N/A	
Originator	Helene F. S.	Date 01.05.2016
Modified by	Helene	Date 01.05.2016
Checked by	Håkon M.	Date 01.05.2016
Approved by	Helene F. S.	Date 01.05.2016
Design state		Release Date

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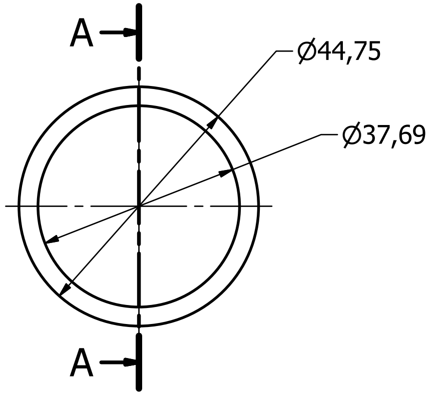
VALVES OF NORWAY
NORSKE VENTILER AS

TITEL
O-Ring 44,45 x 3,53

DOCUMENT NO.	REV.
-	1

Scale:	Size:	
1:1	A3	Sheet 1 of 1



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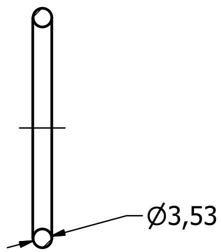

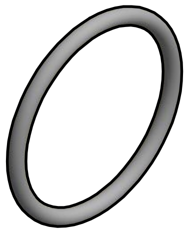

Unless otherwise specified: Break sharp corners 0.2; Surface Ra 1.6	Fillet radii 0.8; Tolerances ISO 2768-f
Material	NBR 70
Weight	N/A
Originator	Helene F. S.
Date	01.05.2016
Modified by	Helene
Date	01.05.2016
Checked by	Håkon M.
Date	01.05.2016
Approved by	Helene F. S.
Date	01.05.2016
Design state	
Release Date	

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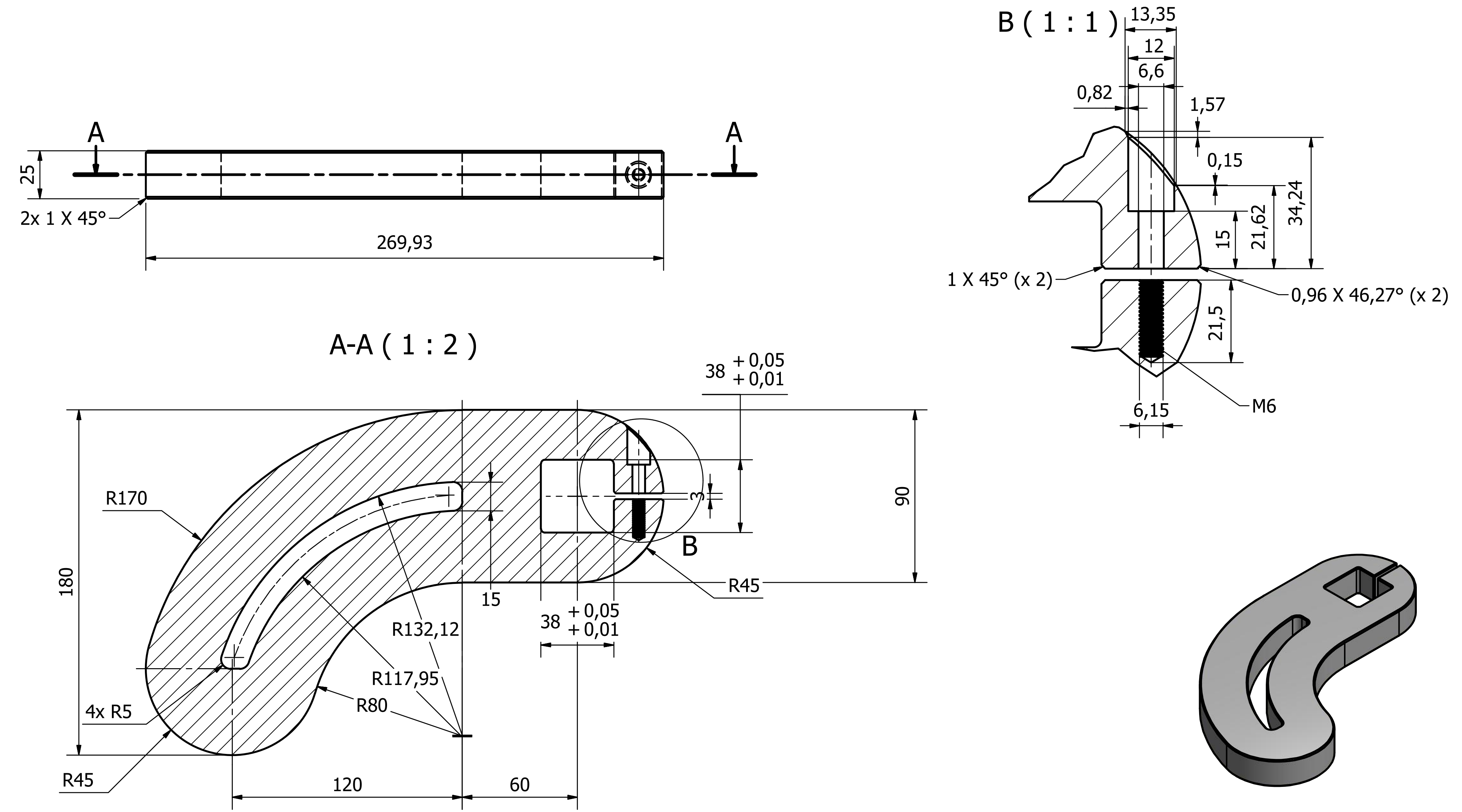
VALVES OF NORWAY
NORSKE VENTILER AS

TITEL
O-Ring 37,69 x 3,53

DOCUMENT NO.	REV.
–	

www.norskeventiler.no	Scale: 1:1	Size: A3	Sheet 1 of 1



Unless otherwise specified: Break sharp corners 0.2; Surface Ra 1.6; Fillet radii 0.8; Tolerances ISO 2768-f	Material: A479 UNS S31600	VALVES OF NORWAY — NORSKE VENTILER AS
	Weight: 2,3 kg	
	Originator: Håkon M. — Date: 19.04.2016	TITEL: Mount
	Modified by: Helene F. S. — Date: 29.04.2016	
	Checked by: Einar K. — Date: 29.04.2016	DOCUMENT NO.: 001 — REV.: 1
	Approved by: Håkon M. — Date: 30.04.2016	
	Design state — Release Date	
	www.norskeventiler.no	Scale: 1:2 — Size: A3 — Sheet 1 of 1

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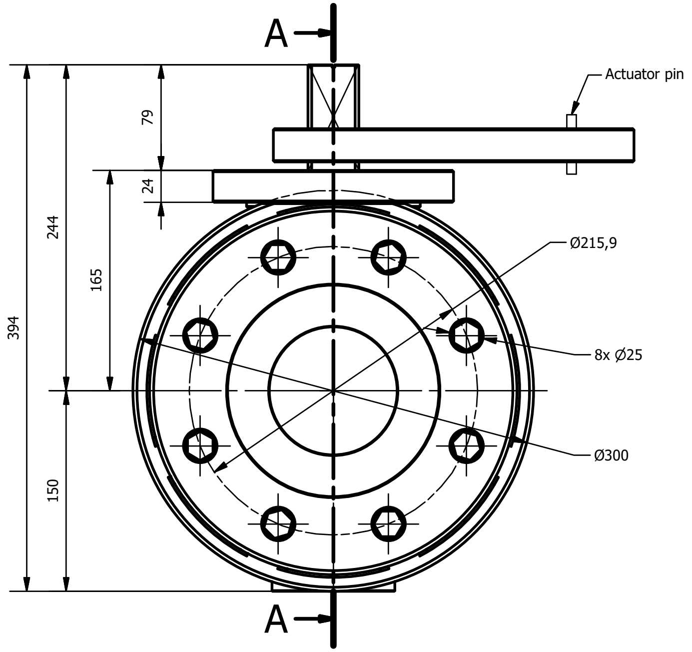
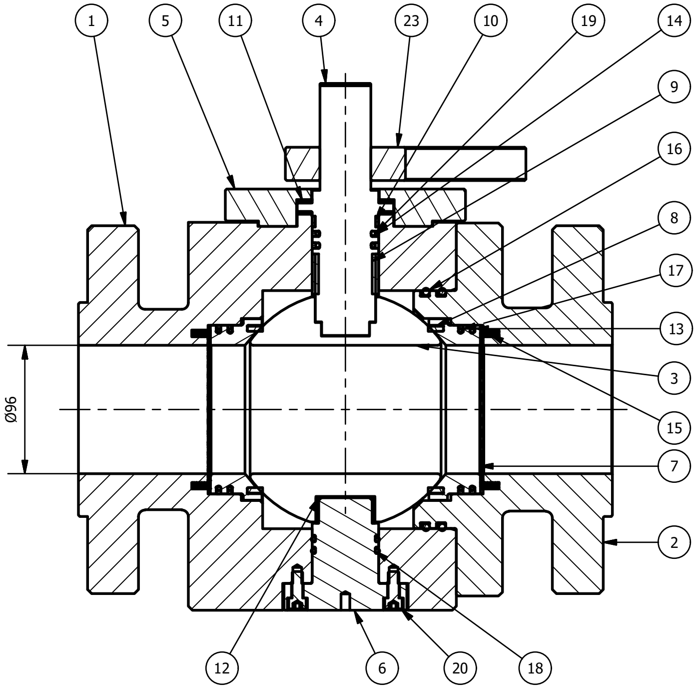
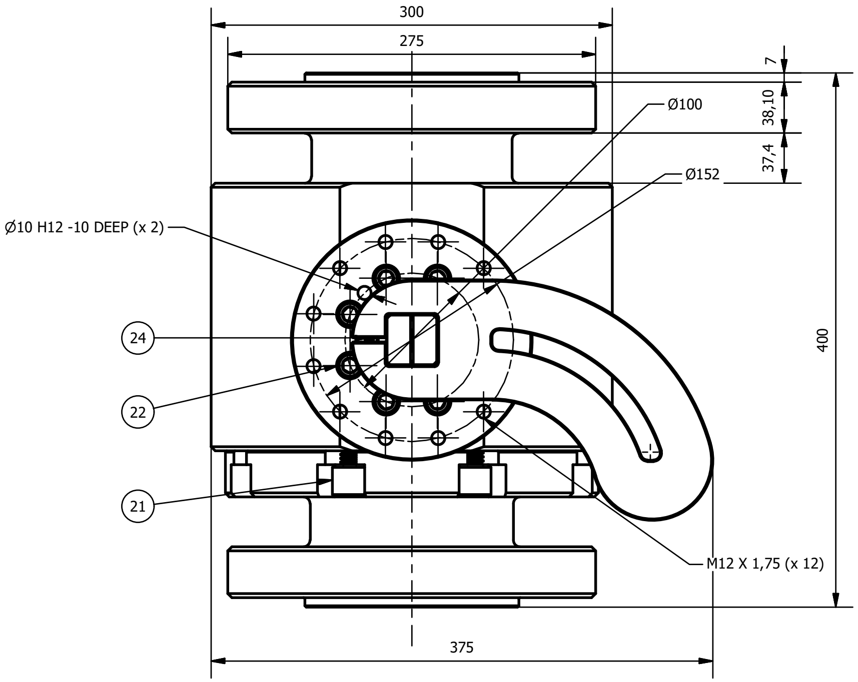
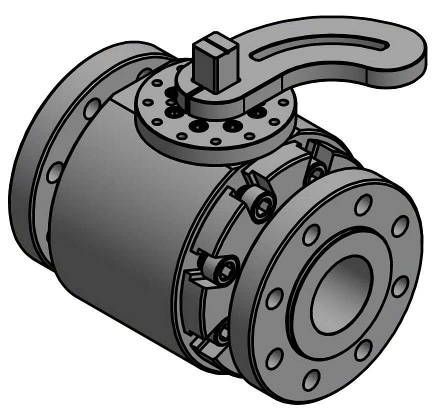


A-A (1 : 4)





PARTS LIST				
ITEM	QTY	PART NUM	TITLE	MATERIAL
1	1	NV 200009	Valve House	A479 UNS S31600
2	1	NV 200010	Valve Lid	A479 UNS S31600
3	1	20367	Ball	A276 S32750
4	1	NV 200012	Stem	UNS NO7718
5	1	NV 200000	Stem Lid	A479 UNS S31600
6	1	NV 200001	Trunnion Lid	A479 UNS S31600
7	2	NV 200163	Seat	A276 S32750
8	2	NV 200164	Seat gasket	PEEK
9	1		Bearing Lower Stem	FEROGLIDE woven PTFE
10	1		Bearing Upper Stem	FEROGLIDE woven PTFE
11	2	20562	Thrust Bearing	PETP
12	1		Bearing Trunnion	FEROGLIDE woven PTFE
13	4	20358	Seat Glide Ring	TFM
14	2	20561	Stem Glide Ring	TFM
15	80	103732B	Spring	UNS NO6625
16	2		O-Ring 170,82 x 5,33	NBR 70
17	4		O-Ring 113,89 x 3,53	NBR 70
18	2		O-Ring 44,45 x 3,53	NBR 70
19	2		O-Ring 37,69 x 3,53	NBR 70
20	10		ISO 4762 - M8 x 20	A4-80
21	8		ISO 4762 - M24 x 40	A4-80
22	8		ISO 4762 - M12 x 40	A4-80
23	1	001	Mount	A479 UNS S31600
24	1		ISO 4762 - M6 x 35	A4-80



Design Code: ISO 13628
Working Pressure: 10 MPa
Operating Pressure: 20 MPa
Body Test Pressure: 15 MPa
Design Temperature: 0 to 70 ˚C
Maximum Depth: 1000 m
Required Operating Torque: 4000 Nm
Interface: 4" ASME B16,5 CL600 RF

Unless otherwise specified:
Break sharp corners 0.2 | Fillet radii 0.8
Surface Ra 1.6 | Tolerances ISO 2768-f

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Material		
Weight	124 kg	
Originator	Helene F. S.	Date 19.04.2016
Modified by	Helene F. S.	Date 29.04.2016
Checked by	Einar K.	Date 29.04.2016
Approved by	Håkon M.	Date 30.04.2016
Design state		Release Date

VALVES OF NORWAY
NORSKE VENTILER AS

TITEL: TVB96S

DOCUMENT NO.

REV. 4

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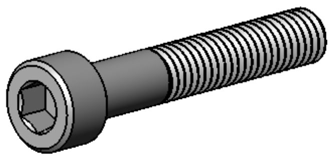
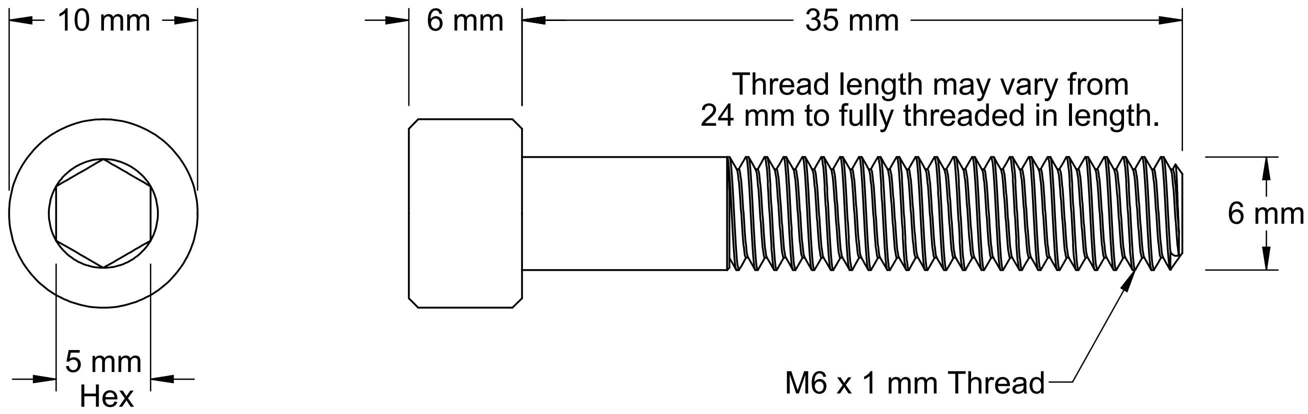
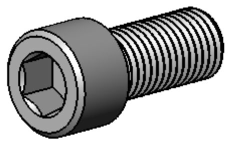
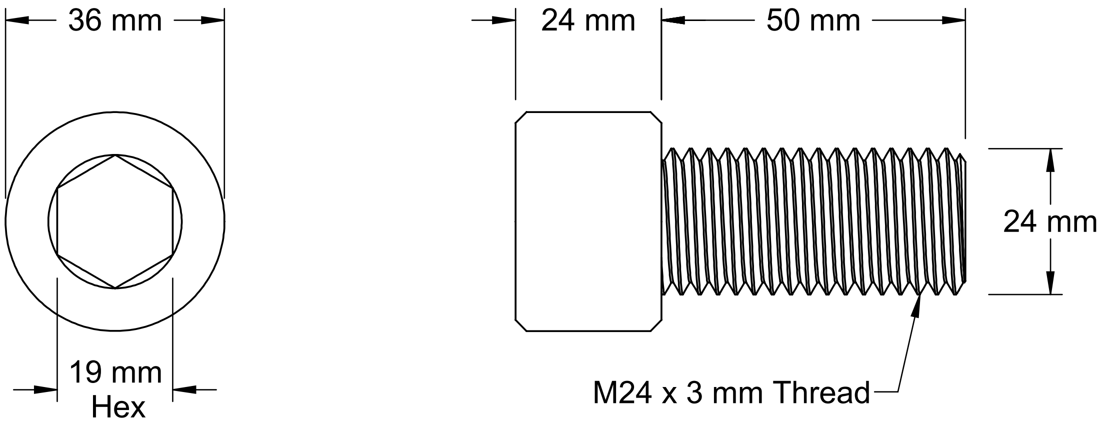


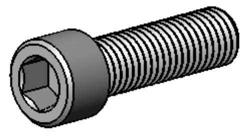


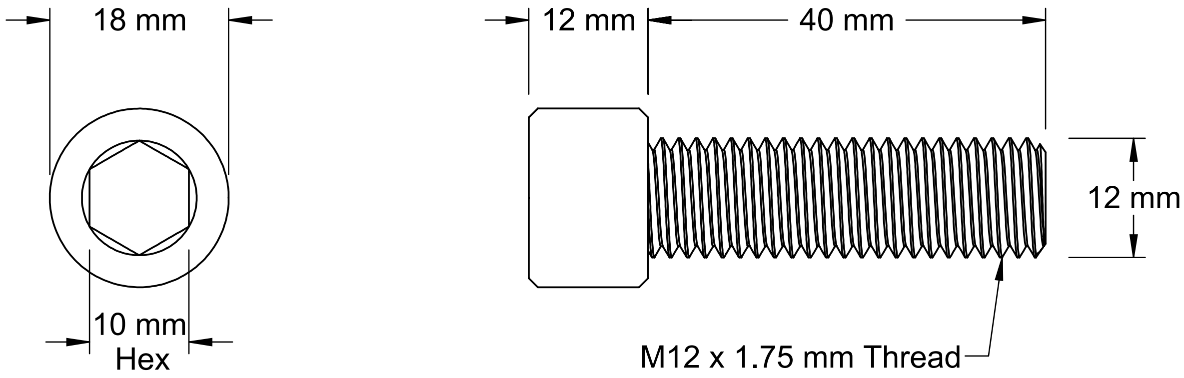
McMASTER-CARR CAD	PART NUMBER 92290A334
http://www.mcmaster.com © 2014 McMaster-Carr Supply Company Information in this drawing is provided for reference only.	Stainless Steel Socket Head Cap Screw



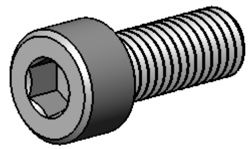


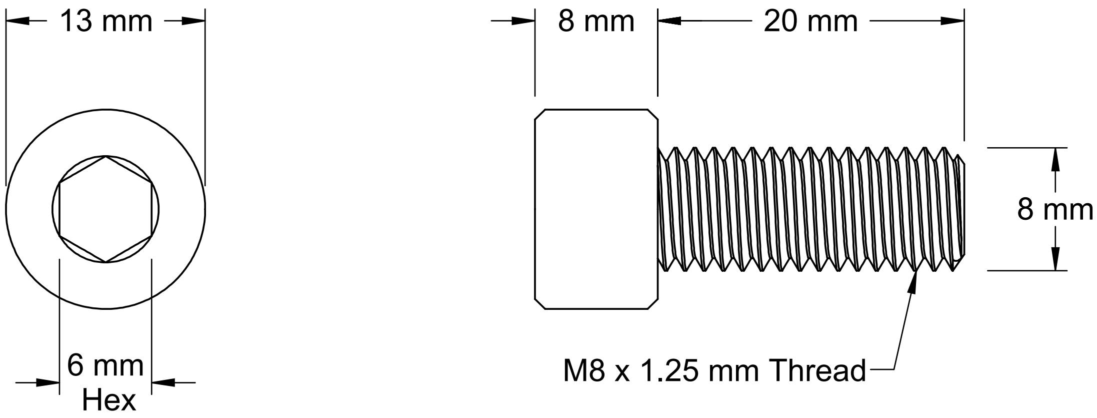
McMASTER-CARR® CAD	PART NUMBER	91292A282
http://www.mcmaster.com © 2014 McMaster-Carr Supply Company Information in this drawing is provided for reference only.	Stainless Steel Socket Head Cap Screw	





McMASTER-CARR® CAD	PART NUMBER 92290A622
http://www.mcmaster.com © 2014 McMaster-Carr Supply Company	Stainless Steel Socket Head Cap Screw
Information in this drawing is provided for reference only.	





McMASTER-CARR® CAD

PART NUMBER 92290A426

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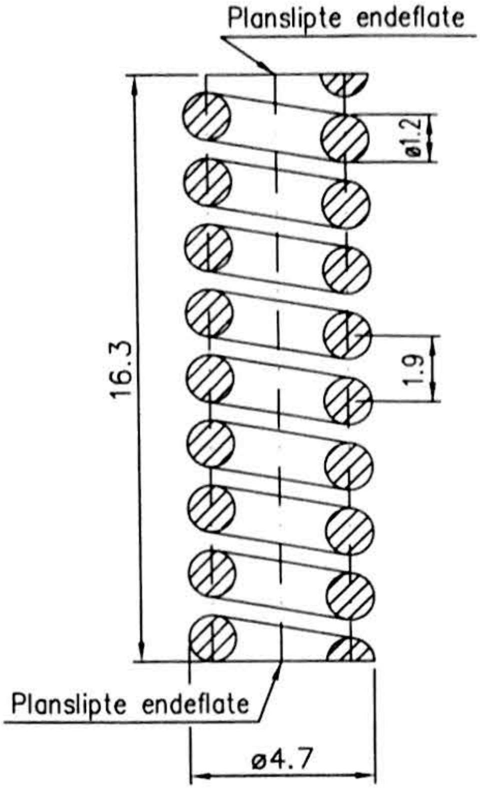
Stainless Steel
Socket Head Cap Screw



Ant. vindinger totalt: ig= 9.5 stk
Ant. fjærende vindinger: if= 8 stk
Minste tillatt lengde: Estimated to 13 mm

ESTIMERTE VERDIER:

Glidemodul: $G = \frac{E}{2(1 + v)} = 72782\ N/mm^2$

Fjærkonstant: $k = \frac{G \times d^4}{8 \times Dm^3 \times if} = 55\ N/mm$

(Fjærkraft målt til ca 5 Kg/mm 19.01.2010)

$l_1 = 13{,}9\ mm$

Rev.no.	Date	Alternations	Rev.by	Cont.by
01	12.09.2011	16.3 var 15.0 + 9.5 var 10.5	S.J.	S

Gasket art. no.	Size & Class	Material	O.no.	Number	Notes	Wght.
		UNS N10276			Hast. C276	
		Elgiloy				
		UNS N06625			Inc.625	
PVM00200471512		UNS N07750			Inc. X-750	

Date:	Design by:	Contr. by:	Scale:	Size:
19.01.10	S.J.	S	5:1	A4
Gen.tolerance NS 1430 med	Gen.roughn. Ra 1.6um	Break all edges (R all 45°)0.1-0.5	Projection	DIMS in mm

Norske Ventiler as
The Norwegian Valve Manufacturer
Postbox 115, N-5346 Ågotnes
Phone: 047 56315450
Telefax: 047 56315451
Email: info@norskeventiler.no
http://www.norskeventiler.no

Drawing title.
Trunnion Ball Valve
SPRING 4.7x15x1.2

Replaces:
Replaced by:
Drwg. no. 10.3732B
Rev. 01

Appendices

A Tolerances from ISO-2768

General Tolerances to DIN ISO 2768

- The latest DIN standard sheet version applies to all parts made to DIN standards.
- Variations on dimensions without tolerance values are according to "DIN ISO 2768- mk".

GENERAL TOLERANCES FOR LINEAR AND ANGULAR DIMENSIONS (DIN ISO 2768 T1)

LINEAR DIMENSIONS:

Permissible deviations in mm for ranges in nominal lengths	f (fine)	Tolerance class designation (description)		v (very coarse)
		m (medium)	c (coarse)	
0.5 up to 3	±0.05	±0.1	±0.2	-
over 3 up to 6	±0.05	±0.1	±0.3	±0.5
over 6 up to 30	±0.1	±0.2	±0.5	±1.0
over 30 up to 120	±0.15	±0.3	±0.8	±1.5
over 120 up to 400	±0.2	±0.5	±1.2	±2.5
over 400 up to 1000	±0.3	±0.8	±2.0	±4.0
over 1000 up to 2000	±0.5	±1.2	±3.0	±6.0
over 2000 up to 4000	-	±2.0	±4.0	±8.0

EXTERNAL RADIUS AND CHAMFER HEIGHTS

Permissible deviations in mm for ranges in nominal lengths	f (fine)	Tolerance class designation (description)		v (very coarse)
		m (middle)	c (coarse)	
0.5 up to 3	±0.2	±0.2	±0.4	±0.4
over 3 up to 6	±0.5	±0.5	±1.0	±1.0
over 6	±1.0	±1.0	±2.0	±2.0

ANGULAR DIMENSIONS

Permissible deviations in degrees and minutes for ranges in nominal lengths	f (fine)	Tolerance class designation (description)		v (very coarse)
		m (middle)	c (coarse)	
up to 10	±1º	±1º	±1º30'	±3º
over 10 up to 50	±0º30'	±0º30'	±1º	±2º
over 50 up to 120	±0º20'	±0º20'	±0º30'	±1º
over 120 up to 400	±0º10'	±0º10'	±0º15'	±0º30'
over 400	±0º5'	±0º5'	±0º10'	±0º20'

GENERAL TOLERANCES FOR FORM AND POSITION (DIN ISO 2768 T2)

STRAIGHTNESS AND FLATNESS

Ranges in nominal lengths in mm	**Tolerance class**		
	H	**K**	**L**
up to 10	0.02	0.05	0.1
over 10 up to 30	0.05	0.1	0.2
over 30 up to 100	0.1	0.2	0.4
over 100 up to 300	0.2	0.4	0.8
over 300 up to 1000	0.3	0.6	1.2
over 1000 up to 3000	0.4	0.8	1.6

PERPENDICULARITY

Ranges in nominal lengths in mm	**Tolerance class**		
	H	**K**	**L**
up to 100	0.2	0.4	0.6
over 100 up to 300	0.3	0.6	1
over 300 up to 1000	0.4	0.8	1.5
over 1000 up to 3000	0.5	0.8	2

SYMMETRY

Ranges in nominal lengths in mm	**Tolerance class**		
	H	**K**	**L**
up to 100	0.5	0.6	0.6
over 100 up to 300	0.5	0.6	1
over 300 up to 1000	0.5	0.8	1.5
over 1000 up to 3000	0.5	1	2

RUN-OUT

Tolerance class		
H	**K**	**L**
0.1	0.2	0.5

B Bearing material specs

Characteristics of rubbing bearing materials

Material	Maximum P loading lbf/in^2	Maximum P loading MN/m^2	PV value lbf/in^2 × ft/min	PV value MN/m^2 × m/s	Maximum temperature °C	Coefficient of friction	Coefficient of expansion $\times 10^{-6}$/°C	Comments	Application
Carbon/graphite	200–300	1.4–2	$\ngtr$3000 for continuous operation; 5000 for short period life	0.11; 0.18	350–500	0.10–0.25 dry	2.5–5.0	For continuous dry operation $P \ngtr$ 200 lbf/in^2 (1.4 MN/m^2), $V \ngtr$ 250 ft/min (1.25 m/s)	Food and textile machinery where contamination by lubricant inadmissible; furnaces, conveyors, etc. where temperature too high for conventional lubricants; where bearings are immersed in liquids, e.g. water, acid or alkaline solutions, solvents, etc.
Carbon/graphite with metal	450–600	3–4	4000 for continuous operation; 6000 for short period	0.145; 0.22	130–350	0.10–0.35 dry	4.2–5.0	Permissible peak load and temperature depend upon metal impregnant	Bearings working in dusty atmospheres, e.g. coal-mining, foundry plant, steel plant, etc.
Graphite impregnated metal	10 000	70	8000–10 000	0.28–0.35	350–600	0.10–0.15 dry; 0.020–0.025 grease lubricated	12–13 with iron matrix; 16–20 with bronze matrix	Operates satisfactorily dry within stated limits; benefits considerably if small quantity of lubricant present, i.e. higher *PV* values	
Graphite/thermo-setting resin	300	2	~10 000	~0.35	250	0.13–0.5 dry	3.5–5.0	Particularly suitable for operation in sea-water or corrosive fluids	
Reinforced thermo-setting plastics	5000	35	~10 000	~0.35	200	0.1–0.4 dry; 0.006 claimed with water lubrication	25–80 depending on plane of reinforcement	Values depend upon type of reinforcement, e.g. cloth, asbestos, etc. Higher *PV* values when lubricated	Water-lubricated roll-neck bearings (esp. hot rolling mills), marine sterntube and rudder bearings; bearings subject to atomic radiation
Thermo-plastic material without filler	1500	10	~1000	~0.035	100	0.1–0.45 dry	~100	Higher *PV* values acceptable if higher wear rates tolerated. With initial lubrication only, *PV* values up to 20 000 can be imposed	Bushes and thrust washers in automotive, textile and food machinery—linkages where lubrication difficult

Characteristics of rubbing bearing materials *(continued)*

Material	Maximum P loading lbf/in^2	Maximum P loading MN/m^2	PV value lbf/in^2 × ft/min	PV value MN/m^2 × m/s	Maximum temperature °C	Coefficient of friction	Coefficient of expansion $\times 10^{-6}$/°C	Comments	Applications
Thermo-plastic with filler or metal-backed	1500–2000	10–14	1000–3000	0.035–0.11	100	0.15–0.40 dry	80–100	Higher loadings and *PV* values sustained by metal-backed components, especially if lubricated	As above, and for more heavily loaded applications
Thermo-plastic with filler bonded to metal back	20 000	140	10 000	0.35	105	0.20–0.35 dry	27	With initial lubrication only. *PV* values up to 40 000 acceptable with re-lubrication at 500–1000 h intervals	For conditions of intermittent operation or boundary lubrication, or where lubrication limited to assembly or servicing periods, e.g. ball-joints, suspension and steering linkages, king-pin bushes, gearbox bushes, etc.
Filled PTFE	1000	7	Up to 10 000	Up to 0.35	250	0.05–0.35 dry	60–80	Many different types of filler used, e.g. glass, mica, bronze, graphite. Permissible *PV* and unit load and wear rate depend upon filler material, temperature, mating surface material and finish	For dry operation where low friction and low wear rate required, e.g. bushes, thrust washers, slideways, etc., may also be used lubricated
PTFE with filler, bonded to steel backing	20 000	140	Up to 50 000 continuous rating	Up to 1.75	280	0.05–0.30 dry	20 (lining)	Sintered bronze, bonded to steel backing, and impregnated with PTFE/lead	Aircraft controls, linkages; automotive gearboxes, clutch, steering suspension, bushes, conveyors, bridge and building expansion bearings
Woven PTFE reinforced and bonded to metal backing	60 000	420	Up to 45 000 continuous rating	Up to 1.60	250	0.03–0.30 dry	—	The reinforcement may be interwoven glass fibre or rayon	Aircraft and engine controls, linkages, automotive suspensions, engine mountings, bridge and building expansion bearings

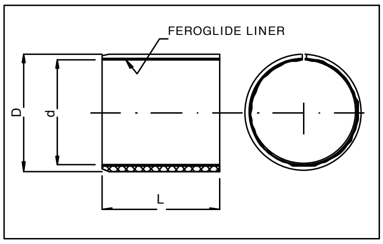
Notes: (1) Rates of wear for a given material are influenced by load, speed, temperature, material and finish of mating surface. The *PV* values quoted in the above table are based upon a wear rate of 0.001 in (0.025×10^{-3} m) per 100 h, where such data are available. For specific applications higher or lower wear rates may be acceptable—consult the bearing supplier.
(2) Where lubrication is provided, either by conventional lubricants or by process fluids, considerably higher *PV* values can usually be tolerated than for dry operation.

Plain bearing materials

C FEROGLIDE dimensions

FEROGLIDE

PA1 JOURNAL BEARINGS COILED



Standard Backing Metals:
1) Zinc coated mild steel
2) Stainless Steel (AISI 316)
3) Inconel 625

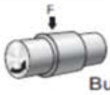
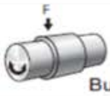
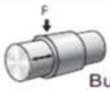
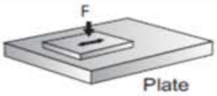
Sizes and Tolerances to DIN1494

Inner diameter tolerance H10 when installed in housing. Parts maybe out of round in their free state.

Dash No.	Inner d	Outer D	Length L	Weight g
P-A1-24.25	24	27	25	21.9
P-A1-24.30	24	27	30	26.3
P-A1-25.15	25	28	15	13.1
P-A1-25.20	25	28	20	18.3
P-A1-25.25	25	28	25	22.9
P-A1-25.30	25	28	30	27.6
P-A1-25.50	25	28	50	46.0
P-A1-28.20	28	32	20	27.7
P-A1-28.30	28	32	30	41.9
P-A1-30.15	30	34	15	22.4
P-A1-30.20	30	34	20	29.6
P-A1-30.25	30	34	25	37.0
P-A1-30.30	30	34	30	44.7
P-A1-30.40	30	34	40	59.7
P-A1-32.20	32	36	20	31.4
P-A1-32.30	32	36	30	47.5
P-A1-32.40	32	36	40	63.5
P-A1-35.20	35	39	20	34.2
P-A1-35.30	35	39	30	51.6
P-A1-35.40	35	39	40	68.4
P-A1-35.50	35	39	50	86.5
P-A1-40.20	40	44	20	38.8
P-A1-40.30	40	44	30	58.6
P-A1-40.40	40	44	40	77.6
P-A1-40.50	40	44	50	98.2
P-A1-45.20	45	50	20	55.1
P-A1-45.30	45	50	30	83
P-A1-45.40	45	50	40	111
P-A1-45.50	45	50	50	139
P-A1-50.20	50	55	20	62
P-A1-50.30	50	55	30	93
P-A1-50.40	50	55	40	123
P-A1-50.60	50	55	60	185
P-A1-55.40	55	60	40	137
P-A1-55.60	55	60	60	202
P-A1-60.30	60	65	30	109
P-A1-60.40	60	65	40	149
P-A1-60.60	60	65	60	220
P-A1-60.70	60	65	70	265
P-A1-65.50	65	70	50	198
P-A1-65.70	65	70	70	277
P-A1-70.40	70	75	40	170
P-A1-70.50	70	75	50	212

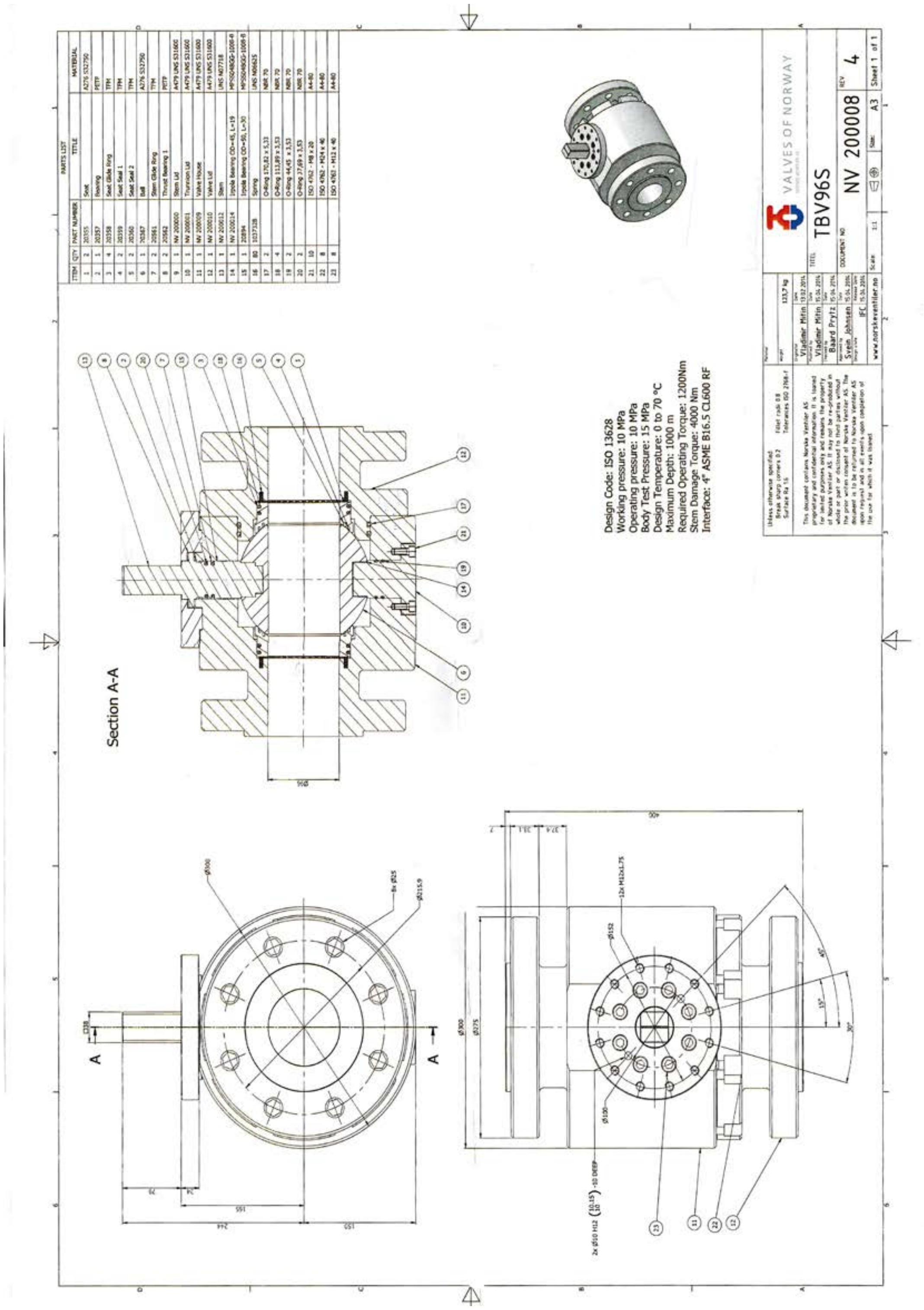
TENMAT LTD, Ashburton Road West, Trafford Park, Manchester, M17 1RU, 0161-872-2181 www.tenmat.com
Issue 2

D AST formulas

		Bearing Pressure P N/mm^2 $\{kgf/cm^2\}$	Velocity V m/s {m/min}	PV Value $N/mm^2 \cdot m/s$ $\{kgf/cm^2 \cdot m/min\}$
1. Rotating motion in single direction of radial journal	 Bushing	$\frac{F}{dL}$ $\left\{\frac{10^2F}{dL}\right\}$	$\frac{\pi dn}{10^3}$ $\left\{\frac{\pi dn}{10^3}\right\}$	$\frac{\pi Fn}{10^3L}$ $\left\{\frac{\pi Fn}{10L}\right\}$
2. Oscillating motion	 Bushing	$\frac{F}{dL}$ $\left\{\frac{10^2F}{dL}\right\}$	$\frac{dc\theta}{10^3}$ $\left\{\frac{\pi dc\theta}{180\times10^3}\right\}$	$\frac{Fc\theta}{10^3L}$ $\left\{\frac{\pi Fc\theta}{180\times10^2L}\right\}$
3. Reciprocating motion	 Bushing	$\frac{F}{dL}$ $\left\{\frac{10^2F}{dL}\right\}$	$\frac{2cS}{10^3}$ $\left\{\frac{2cS}{10^3}\right\}$	$\frac{2FcS}{10^3dL}$ $\left\{\frac{FcS}{5dL}\right\}$
4. Thrust motion	 Thrust washer; Rotation	$\frac{4F}{\pi(D^2-d^2)}$ $\left\{\frac{400F}{\pi(D^2-d^2)}\right\}$	$\frac{\pi Dn}{10^3}$ $\left\{\frac{\pi Dn}{10^3}\right\}$	$\frac{4FDn}{10^3(D^2-d^2)}$ $\left\{\frac{4FDn}{10(D^2-d^2)}\right\}$
	Oscillation	$\frac{4F}{\pi(D^2-d^2)}$ $\left\{\frac{400F}{\pi(D^2-d^2)}\right\}$	$\frac{Dc\theta}{10^3}$ $\left\{\frac{\pi Dc\theta}{180\times10^3}\right\}$	$\frac{4FDc\theta}{10^3\pi(D^2-d^2)}$ $\left\{\frac{4FDc\theta}{180\times10(D^2-d^2)}\right\}$
5. Plane reciprocating motion	 Plate	$\frac{F}{BL}$ $\left\{\frac{10^2F}{WL}\right\}$	$\frac{2cS}{10^3}$ $\left\{\frac{2cS}{10^3}\right\}$	$\frac{2FcS}{10^3BL}$ $\left\{\frac{FcS}{5WL}\right\}$

F : Vertical load.................................N {kgf}
N : Number of rotation .... S^{-1} {rpm}
c : Cyclic velocity of reciprocating
or oscillating motion.................S^{-1} {cpm}
S : Stroke distance .... M {mm}
θ : Oscillating angle .... rad
d : Bearing ID .... mm {mm}
D : Bearing OD .... mm {mm}
L : Bearing length .... mm {mm}
W : Bearing width .... mm {mm}

E Previous 2D assembly drawing



F Metric tap- and clearance drill dimensions

Metric Tap & Clearance Drill Sizes		Tap Drill				Clearance Drill			
		75% Thread for Aluminum, Brass, & Plastics		50% Thread for Steel, Stainless, & Iron		Close Fit		Standard Fit	
Screw Size (mm)	Thread Pitch (mm)	Drill Size (mm)	Closest American Drill	Drill Size (mm)	Closest American Drill	Drill Size (mm)	Closest American Drill	Drill Size (mm)	Closest American Drill
M1.5	0.35	1.15	56	1.25	55	1.60	1/16	1.65	52
M1.6	0.35	1.25	55	1.35	54	1.70	51	1.75	50
M 1.8	0.35	1.45	53	1.55	1/16	1.90	49	2.00	5/64
M 2	0.45	1.55	1/16	1.70	51	2.10	45	2.20	44
	0.40	1.60	52	1.75	50				
M 2.2	0.45	1.75	50	1.90	48	2.30	3/32	2.40	41
M 2.5	0.45	2.05	46	2.20	44	2.65	37	2.75	7/64
M 3	0.60	2.40	41	2.60	37	3.15	1/8	3.30	30
	0.50	2.50	39	2.70	36				
M 3.5	0.60	2.90	32	3.10	31	3.70	27	3.85	24
M 4	0.75	3.25	30	3.50	28	4.20	19	4.40	17
	0.70	3.30	30	3.50	28				
M 4.5	0.75	3.75	25	4.00	22	4.75	13	5.00	9
M 5	1.00	4.00	21	4.40	11/64	5.25	5	5.50	7/32
	0.90	4.10	20	4.40	17				
	0.80	4.20	19	4.50	16				
M 5.5	0.90	4.60	14	4.90	10	5.80	1	6.10	B
M 6	1.00	5.00	8	5.40	4	6.30	E	6.60	G
	0.75	5.25	4	5.50	7/32				
M 7	1.00	6.00	B	6.40	E	7.40	L	7.70	N
	0.75	6.25	D	6.50	F				
M 8	1.25	6.80	H	7.20	J	8.40	Q	8.80	S
	1.00	7.00	J	7.40	L				
M 9	1.25	7.80	N	8.20	P	9.50	3/8	9.90	25/64
	1.00	8.00	O	8.40	21/64				
M 10	1.50	8.50	R	9.00	T	10.50	Z	11.00	7/16
	1.25	8.80	11/32	9.20	23/64				
	1.00	9.00	T	9.40	U				
M 11	1.50	9.50	3/8	10.00	X	11.60	29/64	12.10	15/32
M 12	1.75	10.30	13/32	10.90	27/64	12.60	1/2	13.20	33/64
	1.50	10.50	Z	11.00	7/16				
	1.25	10.80	27/64	11.20	7/16				
M 14	2.00	12.10	15/32	12.70	1/2	14.75	37/64	15.50	39/64
	1.50	12.50	1/2	13.00	33/64				
	1.25	12.80	1/2	13.20	33/64				
M 15	1.50	13.50	17/32	14.00	35/64	15.75	5/8	16.50	21/32
M 16	2.00	14.00	35/64	14.75	37/64	16.75	21/32	17.50	11/16
	1.50	14.50	37/64	15.00	19/32				
M 17	1.50	15.50	39/64	16.00	5/8	18.00	45/64	18.50	47/64
M 18	2.50	15.50	39/64	16.50	41/64	19.00	3/4	20.00	25/32
	2.00	16.00	5/8	16.75	21/32				
	1.50	16.50	21/32	17.00	43/64				
M 19	2.50	16.50	21/32	17.50	11/16	20.00	25/32	21.00	53/64
M 20	2.50	17.50	11/16	18.50	23/32	21.00	53/64	22.00	55/64
	2.00	18.00	45/64	18.50	47/64				
	1.50	18.50	47/64	19.00	3/4				

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G 316 Stainless steel properties

Mechanical Properties

Table 2. Mechanical properties of 316 grade stainless steels.

Grade	Tensile Str (MPa) min	Yield Str 0.2% Proof (MPa) min	Elong (% in 50mm) min	Hardness	
				Rockwell B (HR B) max	Brinell (HB) max
316	515	205	40	95	217
316L	485	170	40	95	217
316H	515	205	40	95	217

Note: 316H also has a requirement for a grain size of ASTM no. 7 or coarser.

Physical Properties

Table 3. Typical physical properties for 316 grade stainless steels.

Grade	Density (kg/m^3)	Elastic Modulus (GPa)	Mean Co-eff of Thermal Expansion (μm/m/°C)			Thermal Conductivity (W/m.K)		Specific Heat 0-100°C (J/kg.K)	Elec Resistivity (nΩ.m)
			0-100°C	0-315°C	0-538°C	At 100°C	At 500°C		
	8000	193	15.9	16.2	17.5	16.3	21.5	500	740

http://www.azom.com/article.aspx?ArticleID=863

H Friction coefficient for stainless steel bolts

Screw made from	Nut made from	Lubricant in thread	Lubricant under head	Resilience of connection	Friction coefficient in thread μ_G	Friction coefficient under head μ_K
A 2	A 2	none	none	very high	**0.26** to 0.50	**0.35** to 0.50
		Special lubricant (chloroparaffin base)			**0.12** to 0.23	**0.08** to 0.12
		Anticorrosive grease			**0.26** to 0.45	**0.25** to 0.35
		none	none	low	**0.23** to 0.35	**0.12** to 0.16
		Special lubricant (chloroparaffin base)			**0.10** to 0.16	**0.08** to 0.12
	AlMgSi	none		very high	**0.32** to 0.43	**0.08** to 0.11
		Special lubricant (chloroparaffin base)			**0.28** to 0.35	**0.08** to 0.11

I Torque Constant K

Coefficient of Friction										
Between Threads, μ_s	Between Bearing Surfaces, μ_w									
	0.08	0.10	0.12	0.15	0.20	0.25	0.30	0.35	0.40	0.45
0.08	0.117	0.130	0.143	0.163	0.195	0.228	0.261	0.293	0.326	0.359
0.10	0.127	0.140	0.153	0.173	0.206	0.239	0.271	0.304	0.337	0.369
0.12	0.138	0.151	0.164	0.184	0.216	0.249	0.282	0.314	0.347	0.380
0.15	0.153	0.167	0.180	0.199	0.232	0.265	0.297	0.330	0.363	0.396
0.20	0.180	0.193	0.206	0.226	0.258	0.291	0.324	0.356	0.389	0.422
0.25	0.206	0.219	0.232	0.252	0.284	0.317	0.350	0.383	0.415	0.448
0.30	0.232	0.245	0.258	0.278	0.311	0.343	0.376	0.409	0.442	0.474
0.35	0.258	0.271	0.284	0.304	0.337	0.370	0.402	0.435	0.468	0.500
0.40	0.285	0.298	0.311	0.330	0.363	0.396	0.428	0.461	0.494	0.527
0.45	0.311	0.324	0.337	0.357	0.389	0.422	0.455	0.487	0.520	0.553

J Email regarding PEEK

Helene Foldnes Skage <heleneskage@gmail.com>

Bachelorprosjekt spørsmål

Vladimir Mitin <vladimir@norskeventiler.no> 8. mars 2016 kl. 12.46
Til: Helene Foldnes Skage <heleneskage@gmail.com>
Kopi: Håkon Malkenes <h.m.malkenes@gmail.com>

Hei,

Preloading for PEEK materialet liger på ca 5% av max belastning som regnes ca 0,6 av tensile stress

Vennlig hilsen/ Best regards

Vladimir Mitin

Design engineer
Mob. +47 900 77 982
www.norskeventiler.no

VALVES OF NORWAY

Fra: Helene Foldnes Skage [mailto:heleneskage@gmail.com]
Sendt: 1. mars 2016 19:17
Til: Vladimir Mitin
Kopi: Håkon Malkenes
Emne: Bachelorprosjekt spørsmål

[Sitert tekst skjult]

2 vedlegg

feroglide-gen-info-fb.pdf
123K

feroglide_technical_manual_172_2.pdf
136K

K Calculations torsion, shear and moment.

Calculations of the stresses in each section were Q_1 and U_1 is the von mises and utility for the closed position, and Q_2 and U_2 is for the open position.

K.1 Reaction forces and moments

$$F_{act} = 30000 \text{ N} \tag{K.1}$$
$$F_{us} = 31599 \text{ N} \tag{K.2}$$
$$F_{ls} = 44438 \text{ N} \tag{K.3}$$
$$F_t = 81852 \text{ N} \tag{K.4}$$
$$F_b = 124689 \text{ N} \tag{K.5}$$

$$M_{us_{FEM}} = 8890 \text{ Nmm} \tag{K.6}$$
$$M_{ls_{FEM}} = 584308 \text{ Nmm} \tag{K.7}$$
$$M_{t_{FEM}} = -604929 \text{ Nmm} \tag{K.8}$$

K.2 Shear forces and moments

$$V_A = F_t = 81852 \text{ N} \tag{K.9}$$
$$V_B = V_A - F_b = -42837 \text{ N} \tag{K.10}$$
$$V_C = V_B + F_l s = 1601 \text{ N} \tag{K.11}$$
$$V_D = V_C - F_u s = -29998 \text{ N} \tag{K.12}$$
$$V_F = V_D + F_a ct = 2 \text{ N} \tag{K.13}$$

$$M_1 = V_A \cdot L_1 + M_{t_{FEM}} = 5533971 \text{ Nmm} \tag{K.14}$$
$$M_2 = M_1 + V_B \cdot L_4 = 1250271 \text{ Nmm} \tag{K.15}$$
$$M_3 = M_2 + M_{ls_{FEM}} = 1834579 \text{ Nmm} \tag{K.16}$$
$$M_4 = M_3 + V_C \cdot L_5 = 1896858 \text{ Nmm} \tag{K.17}$$
$$M_5 = M_4 + M_{us_{FEM}} = 1905748 \text{ Nmm} \tag{K.18}$$
$$M_6 = M_5 + V_D \cdot L_6 = 3875 \text{ Nmm} \tag{K.19}$$
$$M_h = M_1 + V_B \cdot L_2 = 3135099 \text{ Nmm} \tag{K.20}$$
$$M_{lc} = M_1 + V_B \cdot L_3 = 2706729 \text{ Nmm} \tag{K.21}$$
$$M_{cs} = M_2 + V_C \cdot L_{cs} = 1297901 \text{ Nmm} \tag{K.22}$$

K.3 Friction torques

$$d_{us} = 50 \text{ mm} \tag{K.23}$$
$$d_{ls} = 50 \text{ mm} \tag{K.24}$$
$$d_t = 44 \text{ mm} \tag{K.25}$$

$$\mu_{us} = 0.03 \tag{K.26}$$
$$\mu_{ls} = 0.1 \tag{K.27}$$
$$\mu_t = 0.1 \tag{K.28}$$

$$D_g = 126 \text{ mm} \tag{K.29}$$
$$d_g = 116.7 \text{ mm} \tag{K.30}$$
$$p = 10 \text{ N/mm}^2 \tag{K.31}$$

$$A_{pg} = \frac{\pi \cdot \left(D_g^2 - d_g^2\right)}{4} = 1773 \text{ mm}^2 \tag{K.32}$$
$$F_{pg} = A_{pg} \cdot p = 17727 \text{ N} \tag{K.33}$$
$$F_{sg} = 0.05 \cdot 0.6 \cdot 110\ N/mm^2 \cdot A_{pg} = 5850 \text{ N} \tag{K.34}$$
$$D_b = 176 \text{ mm} \tag{K.35}$$
$$y = 0.4 \tag{K.36}$$

$$M_{us} = \frac{F_{us}}{2} \cdot u_{us} \cdot d_{us} = 21329 \text{ Nmm} \tag{K.37}$$
$$M_{ls} = \frac{F_{ls}}{2} \cdot u_{ls} \cdot d_{ls} = 99986 \text{ Nmm} \tag{K.38}$$
$$M_t = \frac{F_t}{2} \cdot u_t \cdot d_t = 163704 \text{ Nmm} \tag{K.39}$$
$$M_{ss} = 4875 \text{ Nmm} \tag{K.40}$$
$$M_{tb} = 60377 \text{ Nmm} \tag{K.41}$$
$$M_{pg} = F_{pg} \cdot y \cdot \frac{1}{2} \cdot D_b \cdot \frac{(1 + sin(50)) \cdot 0.5}{sin(50) + 0.05 \cdot cos(50)} = 690326 \text{ Nmm} \tag{K.42}$$
$$M_{2pg} = 2 \cdot M_{pg} = 1380651 \text{ Nmm} \tag{K.43}$$
$$M_{sg} = 2 \cdot F_{sg} \cdot y \cdot \frac{1}{2} \cdot D_b \cdot \frac{(1 + sin(50)) \cdot 0.5}{sin(50) + 0.05 \cdot cos(50)} = 455615 \text{ Nmm} \tag{K.44}$$

K.4 Allowed stress

$$\sigma_{stem_{allowed}} = \frac{827}{1.5} = 551.33 \text{ N/mm}^2 \tag{K.45}$$

$$\sigma_{trunnion_{allowed}} = \frac{204}{1.5} = 136 \text{ N/mm}^2 \tag{K.46}$$

K.5 Torsion

Torsion for the different sections that are checked. For both the open and closed position. The values for moments used in calculations below is from section 4.1.

$$T_t = M_t = 163704 \text{ Nmm} \tag{K.47}$$

$$T_h = M_t + M_{pg} + M_{sg} = 1308645 \text{ Nmm} \tag{K.48}$$

$$T_{lc} = M_t + M_{pg} + M_{sg} = 1308645 \text{ Nmm} \tag{K.49}$$

$$T_{ls} = M_t + M_{ls} + M_{pg} + M_{sg} = 1409630 \text{ Nmm} \tag{K.50}$$

$$T_{us} = M_t + M_{ls} + M_{pg} + M_{sg} + M_{gr}M_{us} = 1435834 \text{ Nmm} \tag{K.51}$$

$$T_{tot} = M_t + M_{pg} + M_{sq} + M_{ls} + M_{gr} + M_{us} + M_{tb} = 1496211 \text{ Nmm} \tag{K.52}$$

$$\begin{aligned}
T_t &= 0\ Nmm \\
T_h &= 2\left(M_{pg} + M_{sg}\right) = 1836266 \text{ Nmm} \\
T_{lc} &= 2\left(M_{pg} + M_{sg}\right) = 1836266 \text{ Nmm} \\
T_{ls} &= 2\left(M_{pg} + M_{sg}\right) = 1836266 \text{ Nmm} \\
T_{us} &= 2\left(M_{pg} + M_{sg}\right) + M_{gr} = 1841141 \text{ Nmm} \\
T_{tot} &= 2\left(M_{pg} + M_{sg}\right) + M_{gr} + M_{tb} = 1901518 \text{ Nmm}
\end{aligned}$$

K.6 Stress lower Hexagon

$$w_h = 35.8 \text{ mm} \tag{K.53}$$

$$a_h = 29.67 \text{ mm} \tag{K.54}$$

$$y_h = \frac{w}{2} = 17.9 \text{ mm} \tag{K.55}$$

$$I_h = \frac{5 \cdot \sqrt{3} \cdot a_h^4}{16} = 419451 \text{ mm}^4 \tag{K.56}$$

$$Q_h = \frac{1}{8} \cdot a_h \cdot w_h^2 + \frac{1}{12} \cdot w_h^2 \cdot \sqrt{a_h^2 - \frac{w_h}{2}^2} = 7280 \text{ mm}^3 \tag{K.57}$$

$$q_h = \frac{M_h}{I_h} \cdot y_h = 134 \text{ N/mm}^2 \tag{K.58}$$

$$t_{hs} = \frac{V_B \;\; \cdot Q_h}{I_h \cdot w_h} = -21 \text{ N/mm}^2 \tag{K.59}$$

$$t_{ht} = 5.297 \cdot \frac{T_h x}{w_h^3} = 151 \text{ N/mm}^2 \tag{K.60}$$

$$t_{htot} = t_{ht} - t_{hs} = 172 \text{ N/mm}^2 \tag{K.61}$$

$$Q_{h1} = \sqrt{(q_h)^2 + 3 \cdot (t_{htot})^2} = 327 \text{ N/mm}^2 \tag{K.62}$$

$$U_{h1} = Q_{h1}/Q_{stem_{allowed}} \cdot 100\% = 67\% \tag{K.63}$$

$$t_{htot2} = 5.297 \cdot \frac{T_{hx2}}{w_h^3} = 212 \text{ N/mm}^2 \tag{K.64}$$

$$Q_{h2} = \sqrt{3 \cdot (t_{htot2})^2} = 367 \text{ N/mm}^2 \tag{K.65}$$

$$U_{h2} = Q_{h2}/Q_{stem_{allowed}} \cdot 100\% = 67\% \tag{K.66}$$

K.7 Stress lower circular section

$$d_{lc} = 45\ mm \tag{K.67}$$

$$W_{lc} = \frac{pi}{32} \cdot (d_{lc} mm)^3 = 8946\ \mathrm{mm}^3 \tag{K.68}$$

$$A_{lc} = pi \cdot \frac{1}{4} \cdot d_{lc}^2 = 1590\ \mathrm{mm}^2 \tag{K.69}$$

$$q_{lc} = \frac{M_l c}{W_{lc}} = 303\ \mathrm{N/mm}^2 \tag{K.70}$$

$$t_{lcs} = \frac{16}{3} \cdot \frac{V_B}{\pi \cdot d_{lc}^2} = -36\ \mathrm{N/mm}^2 \tag{K.71}$$

$$t_{lct} = \frac{16 \cdot T_{lc}}{\pi \cdot d_{lc}^3} = 73\ \mathrm{N/mm}^2 \tag{K.72}$$

$$t_{lctot} = t_{lct} - t_{lcs} = 109\ \mathrm{N/mm}^2 \tag{K.73}$$

$$Q_{lc1} = \sqrt{(q_{lc})^2 + 3 \cdot (t_{lctot})^2} = 357\ \mathrm{N/mm}^2 \tag{K.74}$$

$$U_{lc1} = \frac{Q_{lc1}}{Q_{stem_{allowed}}} \cdot 100\ \% = 65\% \tag{K.75}$$

$$t_{lct2} = \frac{16 \cdot T_{lc2}}{\pi \cdot d_{lc}^3} = 103\ \mathrm{N/mm}^2 \tag{K.76}$$

$$Q_{lc2} = \sqrt{3 \cdot (t_{lct2})^2} = 178\ \mathrm{N/mm}^2 \tag{K.77}$$

$$U_{lc2} = \frac{Q_{lc2}}{Q_{stem_{allowed}}} \cdot 100\% = 32\% \tag{K.78}$$

K.8 Stress lower stem

$$d_{ls} = 45\ mm \tag{K.79}$$

$$W_{ls} = \frac{pi}{32} \cdot (d_{ls})^3 = 8946\ \text{mm}^3 \tag{K.80}$$

$$A_{ls} = pi \cdot \frac{1}{4} \cdot d_{ls}^2 1590\ \text{mm}^2 \tag{K.81}$$

$$q_{ls} = \frac{M_2}{W_{ls}} = 140\ \text{N/mm}^2 \tag{K.82}$$

$$t_{lst} = \frac{16 \cdot T_{ls}}{\pi \cdot d_{ls}^3} = 79\ \text{N/mm}^2 \tag{K.83}$$

$$Q_{ls1} = \sqrt{(q_{ls})^2 + 3 \cdot (t_{lst})^2} = 195\ \text{N/mm}^2 \tag{K.84}$$

$$U_{ls1} = \frac{Q_{ls1}}{Q_{stem_{allowed}}} \cdot 100\% = 35\% \tag{K.85}$$

$$t_{lst2} = \frac{16 \cdot T_{ls2}}{\pi \cdot d_{ls}^3} = 103\ \text{N/mm}^2 \tag{K.86}$$

$$Q_{ls2} = \sqrt{3 \cdot (t_{lst2})^2} = 178\ \text{N/mm}^2 \tag{K.87}$$

$$U_{ls2} = \frac{Q_{ls2}}{Q_{stem_{allowed}}} \cdot 100\% = 32\% \tag{K.88}$$

K.9 Stress mid upper seal/glide ring

$$d_{cs} = 38.8\ mm \tag{K.89}$$

$$W_{cs} = \frac{pi}{32} \cdot (d_{cs}mm)^3 = 5734\ \text{mm}^4 \tag{K.90}$$

$$A_{cs} = pi \cdot \frac{1}{4} \cdot d_{cs}^2 1182\ \text{mm}^2 \tag{K.91}$$

$$q_{cs} = \frac{M_c s}{W_c s} = 226\ \text{N/mm}^2 \tag{K.92}$$

$$t_{css} = \frac{16}{3} \cdot \frac{V_C}{\pi \cdot d_{cs}^2} = 2\ \text{N/mm}^2 \tag{K.93}$$

$$t_{cst} = \frac{16 \cdot T_{cs}}{\pi \cdot d_{cs}^3} = 123\ \text{N/mm}^2 \tag{K.94}$$

$$t_{cstot} = t_{cst} - t_{css} = 122\ \text{N/mm}^2 \tag{K.95}$$

$$Q_{cs1} = \sqrt{(q_{cs})^2 + 3 \cdot (t_{cstot})^2} = 309\ \text{N/mm}^2 \tag{K.96}$$

$$U_{cs1} = \frac{Q_{cs1}}{Q_{stem_{allowed}}} \cdot 100\% = 56\% \tag{K.97}$$

$$t_{cst2} = \frac{16 \cdot T_{cs2}}{\pi \cdot d_{cs}^3} = 161\ \text{N/mm}^2 \tag{K.98}$$

$$Q_{cs2} = \sqrt{3 \cdot (t_{cst2})^2} = 278\ \text{N/mm}^2 \tag{K.99}$$

$$U_{cs2} = \frac{Q_{cs2}}{Q_{stem_{allowed}}} \cdot 100\% = 50\% \tag{K.100}$$

K.10 Stress upper stem

$$d_{us} = 45\ mm \tag{K.101}$$

$$W_{us} = \frac{pi}{32} \cdot (d_{us} mm)^3 = 8946\ \mathrm{mm}^3 \tag{K.102}$$

$$A_{us} = pi \cdot \frac{1}{4} \cdot d_{us}^2 = 1590\ \mathrm{mm}^2 \tag{K.103}$$

$$q_{us} = \frac{M_3}{W_{us}} = 205\ \mathrm{N/mm}^2 \tag{K.104}$$

$$t_{ust} = \frac{16 \cdot T_{us}}{\pi \cdot d_{us}^3} = 80\ \mathrm{N/mm}^2 \tag{K.105}$$

$$Q_{us1} = \sqrt{(q_{us})^2 + 3 \cdot (t_{ust})^2} = 248\ \mathrm{N/mm}^2 \tag{K.106}$$

$$U_{us1} = \frac{Q_{us1}}{Q_{stem_{allowed}}} \cdot 100\% = 45\% \tag{K.107}$$

$$t_{ust2} = \frac{16 \cdot T_{us2}}{\pi \cdot d_{us}^3} = 103\ \mathrm{N/mm}^2 \tag{K.108}$$

$$Q_{us2} = \sqrt{3 \cdot (t_{ust2})^2} = 178\ \mathrm{N/mm}^2 \tag{K.109}$$

$$U_{us2} = \frac{Q_{us2}}{Q_{stem_{allowed}}} \cdot 100\% = 32\% \tag{K.110}$$

K.11 Stress trunnion

$$d_t = 40\ mm \tag{K.111}$$

$$W_t = \frac{pi}{32} \cdot (d_t mm)^3 = 1257\ \mathrm{mm}^3 \tag{K.112}$$

$$A_t = pi \cdot \frac{1}{4} \cdot d_t^2 = 1257\ \mathrm{mm}^2 \tag{K.113}$$

$$q_t = \frac{M_{t_{FEM}}}{W_t} = -96\ \mathrm{N/mm}^2 \tag{K.114}$$

$$t_{ts} = \frac{V_A}{A_t} = 65\ \mathrm{N/mm}^2 \tag{K.115}$$

$$t_{tt} = \frac{16 \cdot T_t}{\pi \cdot d_t^3} = 13\ \mathrm{N/mm}^2 \tag{K.116}$$

$$t_{ttot} = t_{tt} + t_{ts} = 78\ \mathrm{N/mm}^2 \tag{K.117}$$

$$Q_{t1} = \sqrt{3 \cdot (t_{ttot})^2} = 135\ \mathrm{N/mm}^2 \tag{K.118}$$

$$U_{t1} = \frac{Q_{t1}}{Q_{trunnion_{allowed}}} \cdot 100\% = 100\% \tag{K.119}$$

$$t_{tt2} = \frac{16 \cdot T_{t2}}{\pi \cdot d_t^3} = 0\ \mathrm{N/mm}^2 \tag{K.120}$$

$$Q_{t2} = \sqrt{3 \cdot (t_{t2})^2} = 0\ \mathrm{N/mm}^2 \tag{K.121}$$

$$U_{t2} = \frac{Q_{t2}}{Q_{trunnion_{allowed}}} \cdot 100\% = 0\% \tag{K.122}$$

K.12 Allowed stress bearings

$$\sigma_y = 140 \ \ \text{N/mm}^2 \tag{K.123}$$

$$\gamma = 1.5 \tag{K.124}$$

$$\sigma_{bearing_{allowed}} = \frac{sigma_y}{\gamma} = 93.33 \text{ N/mm}^2 \tag{K.125}$$

K.13 stress lower stem bearing

$$b_{lsb} = 50 \ \ \text{mm} \tag{K.126}$$

$$h_{lsb} = 30 \text{ mm} \tag{K.127}$$

$$A_{lsb} = b_{lsb} \cdot h_{lsb} = 1500 \text{ mm}^2 \tag{K.128}$$

$$q_{lsb} = F_{ls}/A_{lsb} = 30 \text{ N/mm}^2 \tag{K.129}$$

$$U_{lsb} = \frac{q_{lsb}}{q_{totb}} \cdot 100\% = 32\% \tag{K.130}$$

K.14 stress upper stem bearing

$$b_{usb} = 50 \text{ mm} \tag{K.131}$$

$$h_{usb} = 30 \text{ mm} \tag{K.132}$$

$$A_{usb} = b_{usb} \cdot h_{usb} = 390 \text{ mm}^2 \tag{K.133}$$

$$q_{usb} = F_{us}/A_{usb} = 81 \text{ N/mm}^2 \tag{K.134}$$

$$U_{usb} = \frac{q_{usb}}{q_{totb}} \cdot 100\% = 87\% \tag{K.135}$$

K.15 stress trunnion bearing

$$b_{tb} = 50 \text{ mm} \tag{K.136}$$

$$h_{tb} = 30 \text{ mm} \tag{K.137}$$

$$A_{tb} = b_{tb} \cdot h_{tb} = 880 \text{ mm}^2 \tag{K.138}$$

$$q_{tb} = F_t/A_{tb} 93 \text{ N/mm}^2 \tag{K.139}$$

$$U_{tb} = \frac{q_{tb}}{q_{totb}} \cdot 100\% = 100\% \tag{K.140}$$

$$L_{act} = \frac{T_{tot}}{F_{act}} = 50 \text{ mm} \tag{K.141}$$

$$L_{act_a} = \frac{T_{tot} \cdot \gamma}{F_{act}} = 75 \text{ mm} \tag{K.142}$$

L Calculation method for distributed loads

This section shows the calculation method to find the moment if the reaction forces from the bearings was calculated as distributed loads. This will give a lower moment then for point loads, and for that reason not used.

The lengths that the force is distributed on is trunnion, gasket and the lower and upper bearing, denoted as L_t, L_g, L_{ls}, L_{us}. For these lengths there will be a linearly inclining or declining shear force. The lengths L_{g-ls}, L_{ls-us} and L_{us-act} is the lengths in between the loads, where there will be a constant shear force.

Section	Length $[mm]$
L_t	20
L_g	65
L_{g-ls}	23.3
L_{ls}	30
L_{ls-us}	19.8
L_{us}	7.8
L_{us-act}	59.5

To find the points where the declining shear force intercepts the neutral line, which there is three of. The relationship $A/a = B/b$ is used, where A and B is the two legs of the full triangle, and a and b is the legs of the triangle above or below the neutral line, inside the full triangle. The length from the end of the trunnion to the interception is denoted as L_{zero-g}, And for the length from start of the lower bearing to the interception, and from the start of the lower bearing to the interception are denoted as $L_{zero-ls}$ and $L_{zero-us}$.

$$L_{zero-g} = \frac{|V_A| \cdot L_g}{|V_A| + |V_B|} \tag{L.1}$$

$$L_{rest-g} = L_g - L_{zero-g} \tag{L.2}$$

$$L_{zero-ls} = \frac{|V_B| \cdot L_{ls}}{|V_B| + |V_C|} \tag{L.3}$$

$$L_{rest-ls} = L_{ls} - L_{zero-ls} \tag{L.4}$$

$$L_{zero-us} = \frac{|V_C| \cdot L_{us}}{|V_C| + |V_D|} \tag{L.5}$$

$$L_{rest-us} = L_{us} - L_{zero-us} \tag{L.6}$$

The moments at the different lengths are then:

$$
\begin{aligned}
M_1 &= M_{t-FEM} + \frac{V_A \cdot L_t}{2} && \text{(L.7)}\\
M_2 &= M_1 + \frac{V_A \cdot L_{zero-g}}{2} && \text{(L.8)}\\
M_3 &= M_2 + \frac{V_B \cdot L_{rest-g}}{2} && \text{(L.9)}\\
M_4 &= M_3 + V_B \cdot L_g && \text{(L.10)}\\
M_5 &= M_4 + M_{ls-FEM} + L_{zero-ls} \cdot V_B && \text{(L.11)}\\
M_6 &= M_5 + L_{rest-ls} \cdot V_C && \text{(L.12)}\\
M_7 &= M_6 + L_{ls-us} \cdot V_C && \text{(L.13)}\\
M_8 &= M_7 + M_{us-FEM} + L_{zero-us} \cdot V_C && \text{(L.14)}\\
M_9 &= M_8 + L_{rest-us} \cdot V_D && \text{(L.15)}\\
M_{10} &= M_9 + L_{us-act} \cdot V_D && \text{(L.16)}
\end{aligned}
$$

In addition to this, the moments at the interesting sections are as followed. Where M_h is the hexagon, M_c is the lower circular, and M_s is the circular section under the seal/glide ring. M_{tc}, M_{lsc} and M_{usc} is the circular sections beneath the trunnion, lower stem bearing and the upper stem bearing.

$$
\begin{aligned}
M_{tc} &= M_1 && \text{(L.17)}\\
M_h &= M_3 + L_h \cdot V_B && \text{(L.18)}\\
M_c &= M_3 + L_c \cdot V_B && \text{(L.19)}\\
M_{lsc} &= M_5 && \text{(L.20)}\\
M_s &= M_5 + L_s \cdot V_C && \text{(L.21)}\\
M_{usc} &= M_8 && \text{(L.22)}
\end{aligned}
$$

M FEM report for the Bearing forces

Real simulation v8:1

As before

Study Properties

Study Type	Static Stress
Last Modification Date	2016-04-13, 11:39:09

Constraints

Upper

Type	Pinned
Radial	Yes
Axial	No
Tangential	No

Lower

Type	Pinned
Radial	Yes
Axial	No
Tangential	No

Trunnion

Type	Pinned
Radial	Yes
Axial	Yes
Tangential	No

Loads

Force Ball

Type	Force
X Value	0 N
Y Value	124689 N
Z Value	0 N
Radius	70.59 mm

Contacts

Bonded

Name
Bonded9 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded10 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded11 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded12 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded13 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded14 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded15 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded16 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded17 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded18 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded19 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded20 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded21 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded22 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded23 [20367 - Ball:1||NV 200012 - Stem:1]

Separation + Sliding

Name
Separation + Sliding1 [NV 200012 - Stem:1||20894 - Bearing:1]
Separation + Sliding3 [NV 200012 - Stem:1||20357 - Bearing:1]
Separation + Sliding4 [NV 200012 - Stem:1||20357 - Bearing:1]
Separation + Sliding5 [NV 200012 - Stem:1||20357 - Bearing:1]
Separation + Sliding6 [NV 200012 - Stem:1||20894 - Bearing:1]
Separation + Sliding24 [20367 - Ball:1||NV 200014 - Bearing:1]

Results

Reaction Forces

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Upper	3586 N	0 N	1155 N mm	-1154 N mm
		-3586 N		-37.36 N mm
		6.436 N		0 N mm
Lower	48349 N	0 N	481827 N mm	-481827 N mm
		-48349 N		0 N mm
		0 N		0 N mm
Trunnion	72756 N	0 N	548649 N mm	548647 N mm
		-72756 N		1420 N mm
		0 N		0 N mm

N FEM report for mount with bend

Mount bend new v5:1

Bend new

Study Properties

Study Type	Static Stress
Last Modification Date	2016-04-11, 18:02:35

Constraints

Stem

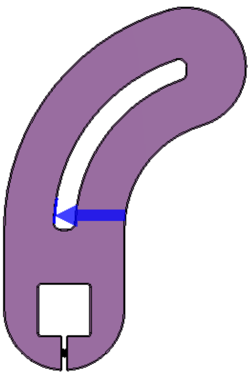
Type	Fixed
Ux	Yes
Uy	Yes
Uz	Yes

Loads

Actuator

Type	Force
X Value	-4.702E-11 N
Y Value	-28000 N
Z Value	0 N
Radius	12.5 mm

Selected Entities



Results

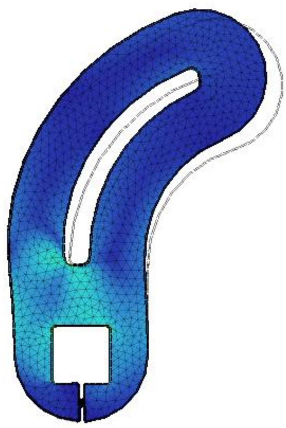
Result Summary

Name	Minimum	Maximum
Safety Factor		
Per Body	0.9125	15
Stress		
Von Mises	0.02678 MPa	224.7 MPa
1st Principal	-59.31 MPa	272.9 MPa
3rd Principal	-219.4 MPa	36.52 MPa
Normal XX	-216.6 MPa	193.1 MPa
Normal YY	-186.9 MPa	199.9 MPa
Normal ZZ	-70.52 MPa	68.65 MPa
Shear XY	-56.48 MPa	119.8 MPa
Shear XZ	-15.19 MPa	24.39 MPa
Shear YZ	-31.02 MPa	25.54 MPa
Displacement		
Total	0 mm	0.1161 mm
X	-0.06764 mm	0.02008 mm
Y	-0.105 mm	0.007412 mm
Z	-0.01042 mm	0.004867 mm
Strain		
Equivalent	1.163E-07	0.001035
1st Principal	-1.975E-05	0.001213
3rd Principal	-9.627E-04	5.666E-06
Normal XX	-9.611E-04	8.728E-04
Normal YY	-8.684E-04	8.72E-04
Normal ZZ	-2.637E-04	1.726E-04
Shear XY	-3.671E-04	7.788E-04
Shear XZ	-9.872E-05	1.585E-04
Shear YZ	-1.92E-04	1.581E-04
Contact Pressure		
Total	0 MPa	55.52 MPa
X	-32.43 MPa	10.53 MPa
Y	-55.43 MPa	34.33 MPa
Z	-23.39 MPa	17.68 MPa

Stress

Von Mises

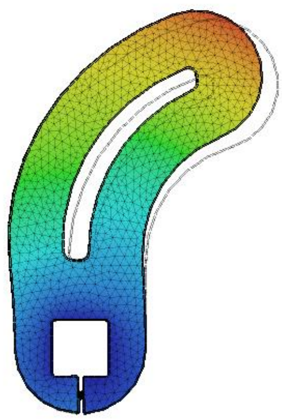
[MPa] 0 224.7, Threshold: 0 - 130.6



Displacement

Total

[mm] 0 0.1161



O FEM report for closed valve

Real simulation v8:1

Negative Y + torsion Closed

Study Properties

Study Type	Static Stress
Last Modification Date	2016-04-13, 12:52:31

Constraints

Upper

Type	Pinned
Radial	Yes
Axial	No
Tangential	No

Lower

Type	Pinned
Radial	Yes
Axial	No
Tangential	No

Trunnion

Type	Pinned
Radial	Yes
Axial	Yes
Tangential	No

Actuator

Type	Pinned
Radial	No
Axial	No
Tangential	Yes

Loads

Force Ball

Type	Force
X Value	0 N
Y Value	124689 N
Z Value	0 N
Radius	70.59 mm

Force actuator

Type	Force
X Value	0 N
Y Value	-30000 N
Z Value	3.886E-12 N

Trunnion friction

Type	Moment
X Value	0 N mm
Y Value	0 N mm
Z Value	163704 N mm

Gasket friction

Type	Moment
X Value	0 N mm
Y Value	0 N mm
Z Value	1.146E+06 N mm

Lower bearing friction

Type	Moment
X Value	0.1 N mm
Y Value	0 N mm
Z Value	99986 N mm

Glide ring

Type	Moment
X Value	0 N mm
Y Value	0 N mm
Z Value	4875 N mm

Upper bearing friction

Type	Moment
X Value	0 N mm
Y Value	0 N mm
Z Value	21329 N mm

Thrust bearing friction

Type	Moment
X Value	0 N mm
Y Value	0 N mm
Z Value	60377 N mm

Contacts

Bonded

Name
Bonded11 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded12 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded13 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded14 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded15 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded16 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded17 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded18 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded19 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded20 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded21 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded23 [20367 - Ball:1||NV 200012 - Stem:1]

Separation + Sliding

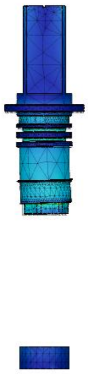
Name
Separation + Sliding1 [NV 200012 - Stem:1||20894 - Bearing:1]
Separation + Sliding3 [NV 200012 - Stem:1||20357 - Bearing:1]
Separation + Sliding4 [NV 200012 - Stem:1||20357 - Bearing:1]
Separation + Sliding5 [NV 200012 - Stem:1||20357 - Bearing:1]
Separation + Sliding6 [NV 200012 - Stem:1||20894 - Bearing:1]
Separation + Sliding24 [20367 - Ball:1||NV 200014 - Bearing:1]

Results

Stress

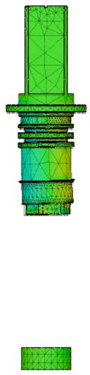
Von Mises

[MPa] 0.3 887.2



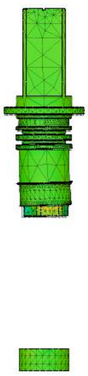
Shear XZ

[MPa] -195.8 184.3



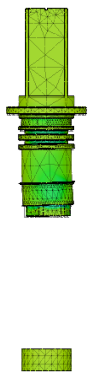
Shear XY

[MPa] -180 166



Shear YZ

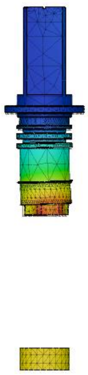
[MPa] -399.6 236.4



Displacement

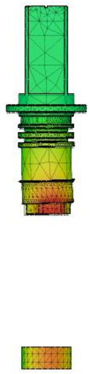
Total

[mm] 0.0203 0.133



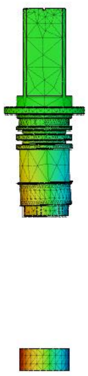
Y

[mm] -0.1066 0.1257



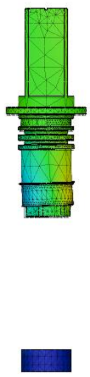
X

[mm] -0.1085 0.1088



Z

[mm] -0.00035 0.04357



P FEM report for open valve

Real simulation v9:1

Negative Y + torsion Open

Study Properties

Study Type	Static Stress
Last Modification Date	2016-04-13, 12:53:06

Constraints

Upper

Type	Pinned
Radial	Yes
Axial	No
Tangential	No

Lower

Type	Pinned
Radial	Yes
Axial	No
Tangential	No

Trunnion

Type	Pinned
Radial	Yes
Axial	Yes
Tangential	No

Actuator

Type	Pinned
Radial	No
Axial	No
Tangential	Yes

Loads

Gasket friction

Type	Moment
X Value	0 N mm
Y Value	0 N mm
Z Value	1.836E+06 N mm

Glide ring

Type	Moment
X Value	0 N mm
Y Value	0 N mm
Z Value	4875 N mm

Thrust bearing friction

Type	Moment
X Value	0 N mm
Y Value	0 N mm
Z Value	60377 N mm

Contacts

Bonded

Name
Bonded11 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded12 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded13 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded14 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded15 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded16 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded17 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded18 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded19 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded20 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded21 [20367 - Ball:1||NV 200012 - Stem:1]
Bonded23 [20367 - Ball:1||NV 200012 - Stem:1]

Separation + Sliding

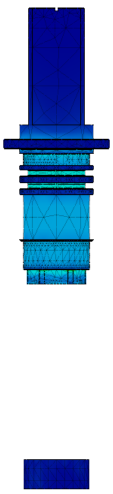
Name
Separation + Sliding1 [NV 200012 - Stem:1||20894 - Bearing:1]
Separation + Sliding3 [NV 200012 - Stem:1||20357 - Bearing:1]
Separation + Sliding4 [NV 200012 - Stem:1||20357 - Bearing:1]
Separation + Sliding5 [NV 200012 - Stem:1||20357 - Bearing:1]
Separation + Sliding6 [NV 200012 - Stem:1||20894 - Bearing:1]
Separation + Sliding24 [20367 - Ball:1||NV 200014 - Bearing:1]

Results

Stress

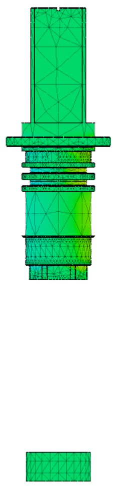
Von Mises

[MPa] 0 1319



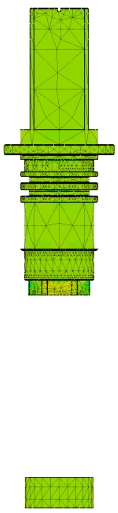
Shear XZ

[MPa] -453.1 556.8



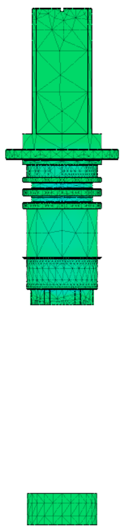
Shear XY

[MPa] -341.1 219.9



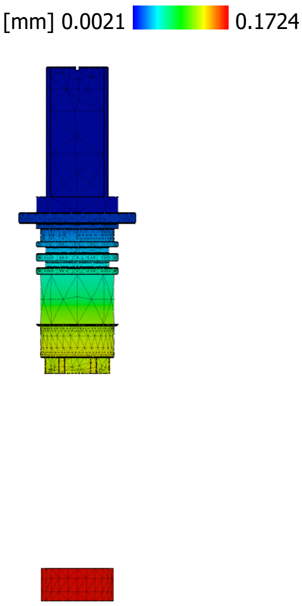
Shear YZ

[MPa] -592.7 725.4

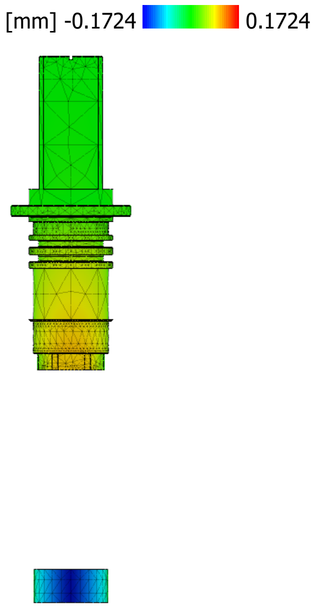


Displacement

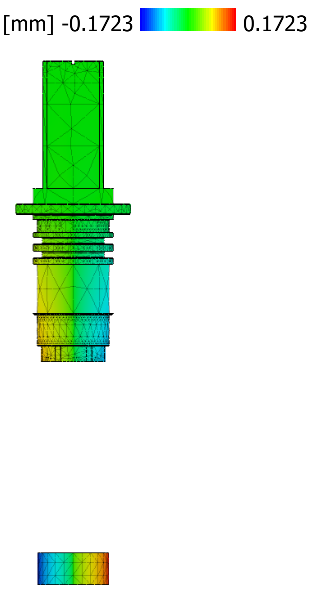
Total



Y



X



Z