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Writer: Sven Tore Jakobsen	(Writer's signature)
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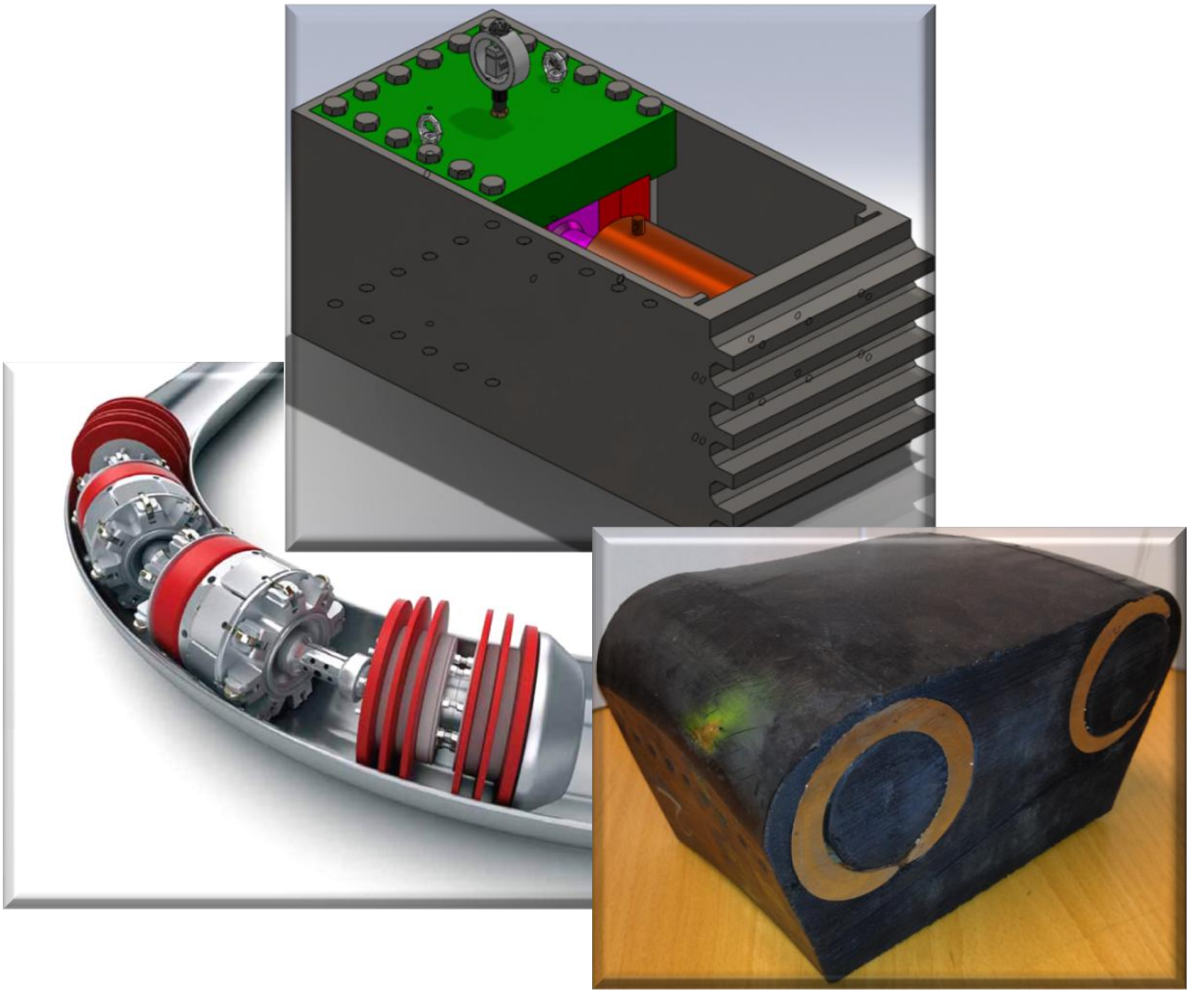
Metal and elastomer matrix composite test method



Sven Tore Jakobsen (213356)

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Abstract

The TDW Offshore Services' SmartPlug® tool is designed to isolate a section of pipeline while it is pressurized. This enables pipeline repair without reducing pipeline pressure or bleeding down the entire pipeline system. High risk connected to jobs of such type lead to high safety level on the SmartPlug® tool. When a sealing element for the plug modules (also called a "packer") is developed, it includes a lot of testing. The test methods TDW currently uses are not economically optimal, and room for improvements exists.

In this master thesis the issues with the existing methods are addressed, and a totally new concept of testing the sealing element is developed. The main idea behind the new concept is to test a linearized "pizza slice" out of the circular full scale packer. This will lower the production cost of the test elements, make it easier and faster for workshop personnel to perform tests, and increase the test frequency. The new test concept has only one test rig that is adjusted and adapted for all different packer sizes. This leads to no occupation of resources usable in other categories, as well as savings in workshop and storage area.

Volume and force calculation models had to be made to transfer the forces from full scale packers to the linearized coupons used in the packer test rig. A test duration estimate as well as a proposal for a practical test execution was done. The economical estimates for production of the test rig adapted for 3 outer limit configurations shows positive economical results.

The conclusion from the work is that the test rig fulfills all design requirements. In theory the test method will work as expected, but tests must be performed to verify the concept and identify any deviations when compared to full scale testing.

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Preface

The master degree in Constructions and materials at the University of Stavanger has in the 10th and last semester an obligatory master thesis. The thesis is a 30ects project and shall deal with a specific problem within the chosen specialism.

This master thesis represents a problem within the specialism “Offshore Systems”. The assignment was composed in collaboration between Sven Tore Jakobsen, Harald Wittersø (TDW), PhD. Jeff Wilson (TDW) and approved by PhD. Vikas Arora (UIS).

The work was carried out at the University of Stavanger in the spring 2012 under the supervision of Arora Vikas, Harald Wittersø and Jeff Wilson. Approximately 1000 man-hours are put down into this thesis.

All the references and captions in the report are hyperlinks and can be used if the thesis is read digitally.

It is emphasized that the reader without any further knowledge shall be able to understand the thesis and its relevant issues. However, the reader should be able to understand engineering problems and calculations. For readers that are not familiar with the SmartPlug® and packer theory the report should be read chronologically. For readers that are familiar with these topics, section 2.1 The SmartPlug and section 2.2. Acrylonitrile-Butadiene Rubber (NBR) will be redundant.

I would like to thank T.D. Williamson for welcoming the challenge, comitting time to the project and for providing working facilities with necessary equipment.

I would like to extend special thanks to:

- PhD. Jeff Wilson, TDW: For good supervision, motivation and ideas to the project.
- Harald Wittersø, TDW: For technical information, support, ideas and feedback on the project.
- PhD. Vikas Arora, UIS: For good supervision.
- Rafael Sulwinski, TDW: For technical help regarding simulation and analysis
- All of the engineers at the sustainable department at TDW Offshore Stavanger for support, ideas and for taking time to questions.

Stavanger 17.04.2012

Sven Tore Jakobsen

Nomenclature

Notation	Unit	Explanation
V_{packer}	MPa	Volume of packer
OD_{packer}	mm	Outer diameter of packer
ID_{packer}	mm	Inner diameter of packer
ID_{pipe}	mm	Inner diameter of pipeline
W_{packer}	mm	Inner width of packer
$W_{packer\ unset}$	mm	Inner width of packer in plug unset position
$W_{packer\ set}$	mm	Inner width of packer in plug set position
H_{packer}	mm	Radial height of packer
r_{spring}	mm	Radius of anti extrusion spring
β	Degrees	Angle between horizontal line and taper angle ($\alpha + \beta = 90^o$)
α	Degrees	Angle between vertical line and taper angle ($\alpha + \beta = 90^o$)
δ	mm	Deformation of packer
AD_{packer}	mm	Average diameter of packer
E	MPa	Modulus of elasticity
K	MPa	Bulk modulus
v	ratio	Poisson's ratio
ΔV	mm^3	Change in volume
V0	mm^3	Initial volume
p	MPa	Pressure
G		Rigidity modulus
d	mm	Amount of rubber bulging through a hole
r	mm	Radius of circular hole
F_{packer}	N	Force to deform packer
$F_{pretension}$	N	Pretension force from hydraulic cylinder
$P_{cylinder}$	Mpa	Pressure inside hydraulic cylinder
$A_{cylinder}$	mm^3	Area of piston in cylinder
$ID_{cylinder}$	mm	Inner diameter of hydraulic cylinder
$D_{piston\ rod}$	mm	Diameter of piston rod in hydraulic cylinder
N_{spring}	Amount	Number of springs
k	N/mm	Spring stiffness
$F_{spring\ plug\ unset}$	N	Spring force in plug unset /spring pretension position
$F_{spring\ plug\ set}$	N	Spring force in plug set position
$L_{spring\ unset}$	mm	Length of spring in spring unset position
$L_{spring\ pretension}$	mm	Length of spring in pretension position
$L_{spring\ plug\ set}$	mm	Length of spring in plug set position
F_{pipe}	N	Force on the plug due to pipeline pressure
P_{pipe}	MPa	Pipeline pressure
$F_{packer\ pipe}$	N	Total force acting on the packer due to pipeline pressure
F_{spring}	N	Total spring force from all springs
$P_{packer\ wall}$	Mpa	Pressure between packer and pipe wall at isolation pressure
A_{packer}	mm^3	Area of packer cross section

$A_{rectangular}$	mm^3	Area of rectangular part of packer cross section break down
$A_{triangular}$	mm^3	Area of triangular part of packer cross section break down
A_{corner}	mm^3	Area of corner part of packer cross section break down
V	mm^3	Volume
A	mm^2	Area
A_R	mm^2	Area of half corner cut out of packer cross section
A_c	mm^2	Area of corner
x_{center}	mm	Distance from orgio in corner coordinate system to center of spring
x_0,y_0		Lower integration limits
X_1,y_1		Upper integration limits
OR_{packer}	mm	Outer radius of packer
IR_{packer}	mm	Inner radius of packer
$\bar{r}$	mm	Radius from revolving axis to center of gravity in area of corner
$\bar{x}$	mm	Distance: Origo in corner coordinate system to center of gravity in corner
V_c	mm^3	Volume of one corner
H	mm	Inner with of packer plus one side in rectangular section
R	J/mol*K	Gas constant
n	Amount	Number of moles
T	K	Temperature
L_{coupon}	mm	Length of packer coupon
Mod 100	MPa	Modulus of elasticity for an elastomer
Shore A	Shore A	Hardness of rubber in packer
$r_{extrusion}$	mm	Extrusion gap (gap between packer unset and ID_{pipe})
$P_{packer\ def}$	MPa	Pressure to deform packer

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Chapter 1: Introduction

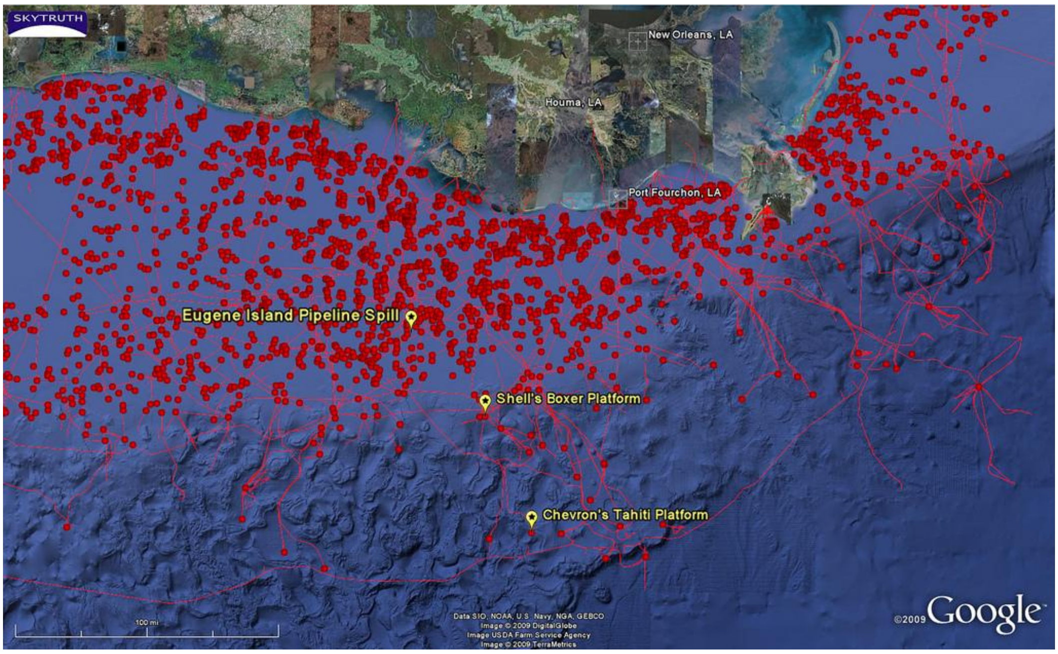
1.1 About T.D. Williamson, Inc

This section is inspired from (T.D Williamson 2011)

T.D.Williamson, Inc. is a well recognized name in pipeline equipment and services, TDW delivers safe integrity solutions for onshore and offshore applications. The company provides hot tapping & plugging, pipeline cleaning, geometry & MFL inspection, pigging and non-tethered plugging pig technology services for any pressurized pipeline system, anywhere in the world. TDW has offices all around the world, the TDW Offshore Stavanger office (Earlier PSI) was acquired by TDW in 2005 and specializes in offshore services, their biggest service is the "SmartPlug® pipeline isolation system".

1.2 Background

It is common to transport large quantities of oil and gas through pipelines. Investment costs are high, but the pipeline systems have long lifetimes, small operating costs, and large capacity. As the amount of pipeline systems in the world steadily increases, see Picture 1 from the GOM, focus on safety levels increases, and the already existing systems get older and worn out. The frequency of maintenance and inspection must increase also. To maintenance a pressurized pipeline containing hydrocarbons, includes a lot of challenges, especially if there is compressible fluid present.



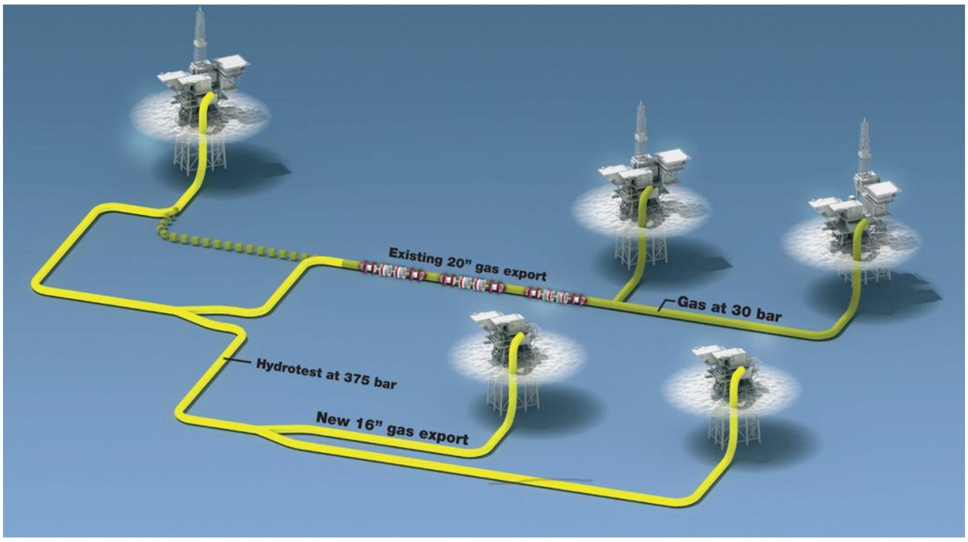
Picture 1: Pipelines in the Gulf of Mexico(SkyTruth's Photostream 2009)

To show why it is favorable to maintenance pipelines while pressurized, we look at an example with a ideal natural gas at 100bar in a 100km long 48inch pipeline. Pressurized to 100bar inside the pipeline the Volume of the gas is 116745,4m^3. When decompressed to atmospheric pressure the volume of the gas is 11674540,3m^3.

If this amount of hydrocarbons have to be disposed, it have to be done under controlled conditions, which is expensive and time-consuming, further advantages of isolating the pipeline while being serviced or repaired is:

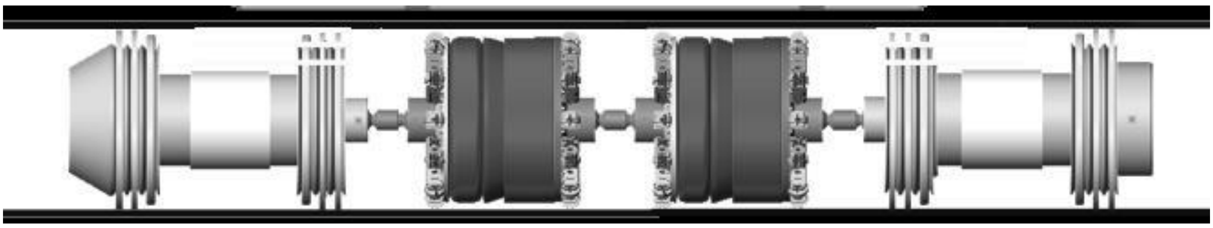
- No flaring of gas or displacement of pipeline contents
- Production continued during pipeline repairs
- No emissions of gas or hydrocarbon vapours to the atmosphere
- No danger of accidentally flooding offshore pipelines during construction
- No hole or future leak path is left at the isolation location
- No time or money spent on de-commissioning (bleeding down) and re-commissioning (refilling and re-pressurizing) pipelines
- No need to dispose of hydrates, chemicals and contaminated water
- Isolates short sections of pipeline anywhere in the pipeline systems

Because of this it is desirable to not bleed down or drain the pipeline before maintenance or inspection jobs are carried out, TDW uses a tool called a "SmartPlug®" to prevent this, see Picture 3.

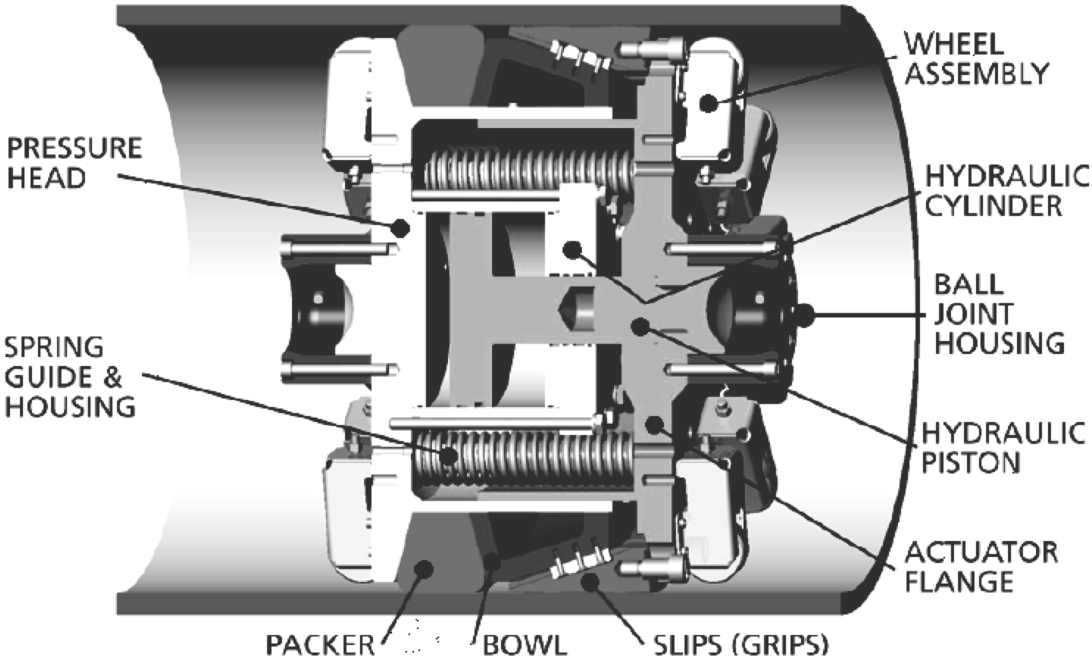


Picture 2: Example of pluging operation(Oilinfo 2012)

The TDW Offshore Services' SmartPlug® tool is designed to isolate a section of pipeline while it is at operating pressure see Picture 2. This enables pipeline repair without reducing pipeline pressure or bleeding down the entire pipeline system.



Picture 3: TDW SmartPlug® tool(Jostein Aleksandersen & Edd Tveit 2001)



Picture 4: Plug module(Jostein Aleksandersen & Edd Tveit 2001)

A SmartPlug® tool, see Picture 3, is custom built to fit each job's requirements such as:

- Pipeline pressure (the pressure the SmartPlug® needs to isolate)
- Pipeline fluid (some pipeline fluids have destructive impacts on some materials)
- Pipeline temperature (materials behave different under different temperatures)
- Maximum allowable stress implemented to the pipeline (under pigging and isolation the SmartPlug® creates stress in the pipeline)

There is one component on the plug module, see Picture 4, that these parameters affect more than others, the "packer" element. The packer is the element that seals between the pipe wall and the plug module itself. The design of this element will affect;

- The stress in the pipe wall (comes from the surface pressure between the packer and the pipe wall)
- Stress in the plug module
- The maximum differential pressure the plug module can take
- What fluid properties it can isolate against
- OD of the plug module in unset position, affect the piggability[1]
- Maximum pipe diameter the plug module can operate in i.e. the biggest possible OD of the plug in set position
- What temperature it can withstand
- The minimum differential pressure the plug module can take (also called; "self lock pressure")

[1] Pigability - The ability the plug train has to be transported through a pipeline and its bends, valves and other obstacles by using a fluid as transport medium.

Due to the advanced material properties of rubber compounds (elastomers), the development of a packer can be very challenging. The development includes a lot of trial and error which can include a lot of testing. Today TDW tests the packer elements in a full scale on a full scale plug module in a test rig similar to the pipeline on the job. Due to an increasing market for SmartPlug® tools, it is not ideal to occupy plug modules in testing, and the time-consuming and expensive packer development method used today have room for improvements.

1.3 Objectives and scope of work

The master thesis scope of work is:

- To address of the problems and challenges with TDW's existing packer test methods.
- To look into a totally new concept for testing the packers to find its attributes.
- To develop the new test concept.

1.4 Computer tools

Computer tools that are used during the thesis program:

- Internet explorer
- Microsoft Paint
- Microsoft Word with Chicago reference style add in
- Microsoft Project
- Microsoft Excel
- Microsoft PowerPoint
- Solid Works with Simulation add in
- Adobe acrobat reader

1.5 Refinements and simplifications

Sections simplifications and refinements is applied to is:

- Calculations; in the calculations some simplifications is done, these and their reason are mentioned where they are applied.
- Drawings; the drawings of the parts are mainly functional drawings and are not ready to be put into production.
- The thesis is focused on TDW's demand and therefore the theory, test methods, calculations and development are not universal but adapted to their request.

1.6 Organization of the work

The content of this master thesis is organized in five chapters. Chapter 1: Introduction, give an introduction to the thesis framework and objectives. Chapter 2: Existing technology, presents the existing technology of SmartPlug®, the rubber material used in the packers and the development methods of a packer. Chapter 3: Basic theory, contains a description of formulas used for calculating the forces acting in a packer, and a information part about the SolidWorks simulation. Chapter 4 is the biggest chapter in the thesis, this chapter describes chronologically the design phase of the packer tester. The chapter contains the conceptual design, the force calculations, SolidWorks simulations and analysis, overview of wanted result recordings from the test rig and at the end a cost estimate of the rig and its parts is done. Chapter 5: Discussion present the final design of the packer tester as well as a discussion and conclusion of the work done.

Chapter 2: Existing technology

2.1 The SmartPlug®

This section is inspired from (Jostein Aleksandersen & Edd Tveit 2001).

Pipeline isolation systems have evolved from the now outdated umbilical operated tools that were operated through penetrations in the pig trap doors with strippers. This method has great limitations both with respect to safety and operations. The development of remotely operated (tether less) tools (SmartPlug®) was initiated by PSI in the mid 90's and in 1999 the world First successful operation took place at the Dimlington Gas terminal in UK. The communication is based on Extremely Low Frequency electromagnetic waves, where digital signals are converted to suitable telegrams for transmission through the pipeline wall as well as any coating or burial mass.

The plug isolation system is a double block and bleed, high pressure, bi-directionally piggable pipeline isolation device.

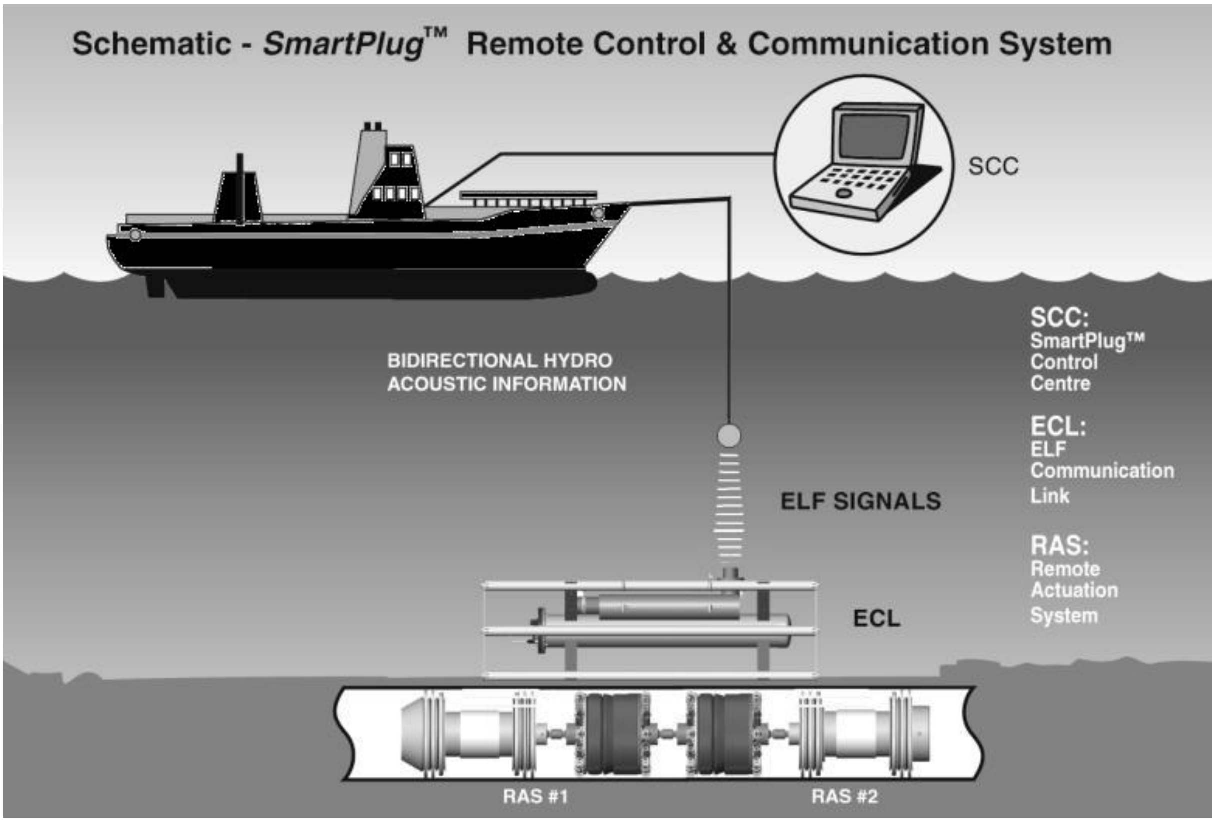
The TDW Plug is a remotely controlled and operated (umbilical-less) pipeline isolation system for use on oil and gas pipelines in all dimensions. They are designed, manufactured, and tested to isolate high pipeline operating pressures.

Communication with the tool for typical subsea application is done from a surface vessel, via acoustic signals to a subsea module, then through the pipeline wall via Extremely Low Frequency (ELF) electromagnetic waves.

All critical parameters such as pressures and temperatures are monitored. The tool design is fail safe. i.e. as long as there is a differential pressure over the isolation system it cannot unset. Thus any failure to the control system will not jeopardize its operation.

2.1.1 System description

The tool generally comprises two isolation Plug Modules and two Pigging Modules, see Picture 8. The remote control and communication system consists of the tool itself, the Surface Control Center (SCC). Acoustic modems, an ELF Communication Link (ECL) and Remote Actuation System (RAS) see Picture 5.



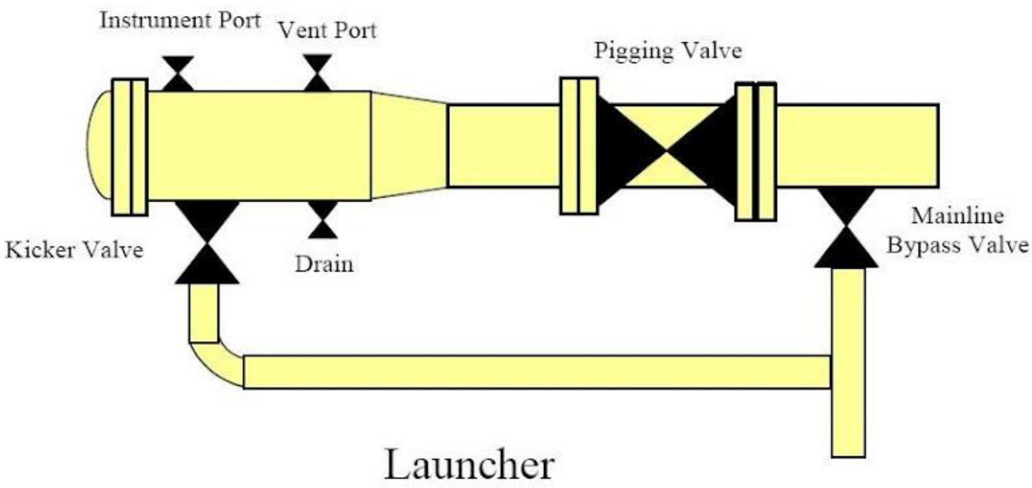
Picture 5: Remote control and communication system(Jostein Aleksandersen & Edd Tveit 2001)

The system component design is based upon proven technology and use of reliable components designed to ensure absolute redundancy. The Isolation Plug System design and operating philosophy emphasizes plug reliability, redundancy and safety. Safety aspects of the operation have been considered and back-up components incorporated for all essential systems.

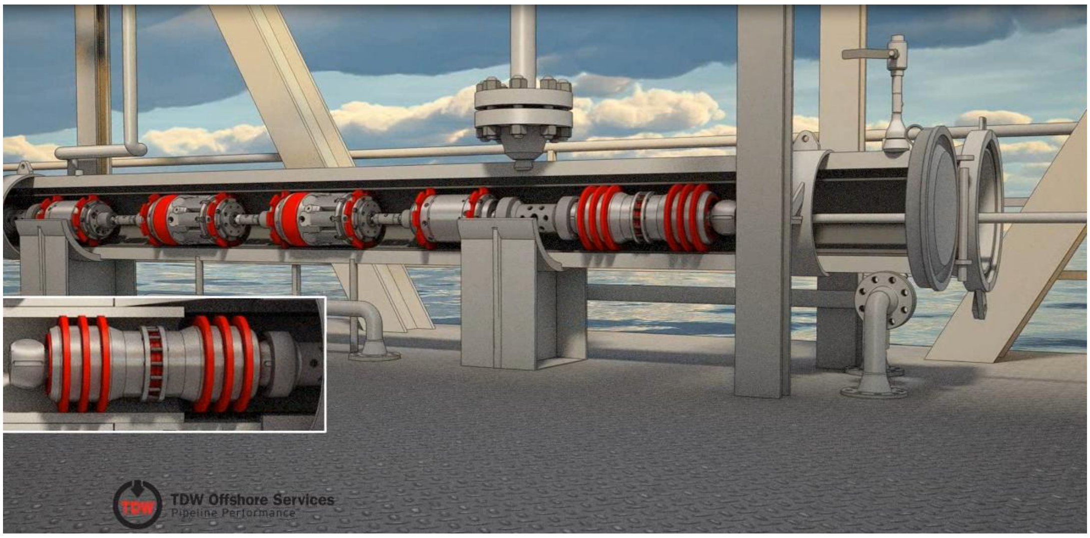
2.1.2 Launching

To get the plug train into the pipeline while pressurized it is inserted into a "pig launcher"see Picture 6. The pig launcher is a "door" into the pipeline that can be isolated. The pig launcher has bigger diameter than the rest of the pipeline due to the resistance in the seal discs on the pigging module. Roughly described the launching process is as follows see Picture 7:

- The launcher is isolated from the pipeline with pigging valves
- Pressure inside the launcher is bled down
- The launcher door is opened
- Plug train is inserted into the launcher
- The launcher door is then closed
- The launcher is pressurized equivalent to pipeline pressure
- Pigging valves is opened (there is no differential pressure over the plug train at this stage)
- Pressure behind the plug train is increased to make differential pressure over the plug train
- The differential pressure needed to move the train is individual for each job, but is in the range of 0,5- 6 bar
- When the plug is at set position the pumping of pigging medium stops



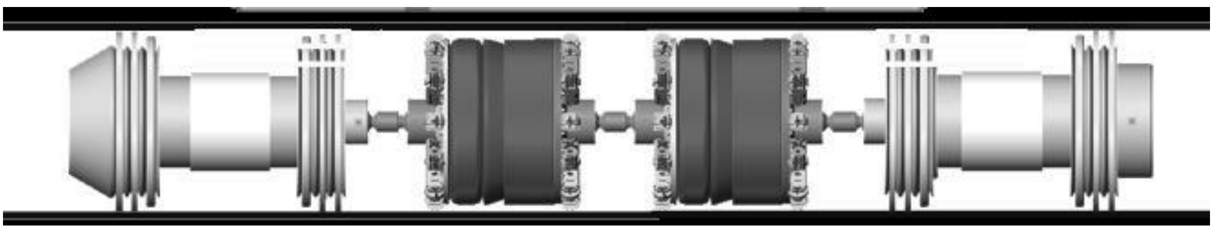
Picture 6: Pig launcher/reciever(Piping Guide 2012)



Picture 7: Launching prosess(T.D Williamson 2011)

2.1.3 Plug Modules

The two Plug Modules perform the seal and lock function see Picture 8, and each of the modules provides the function independent of the other, i.e. absolute redundancy.

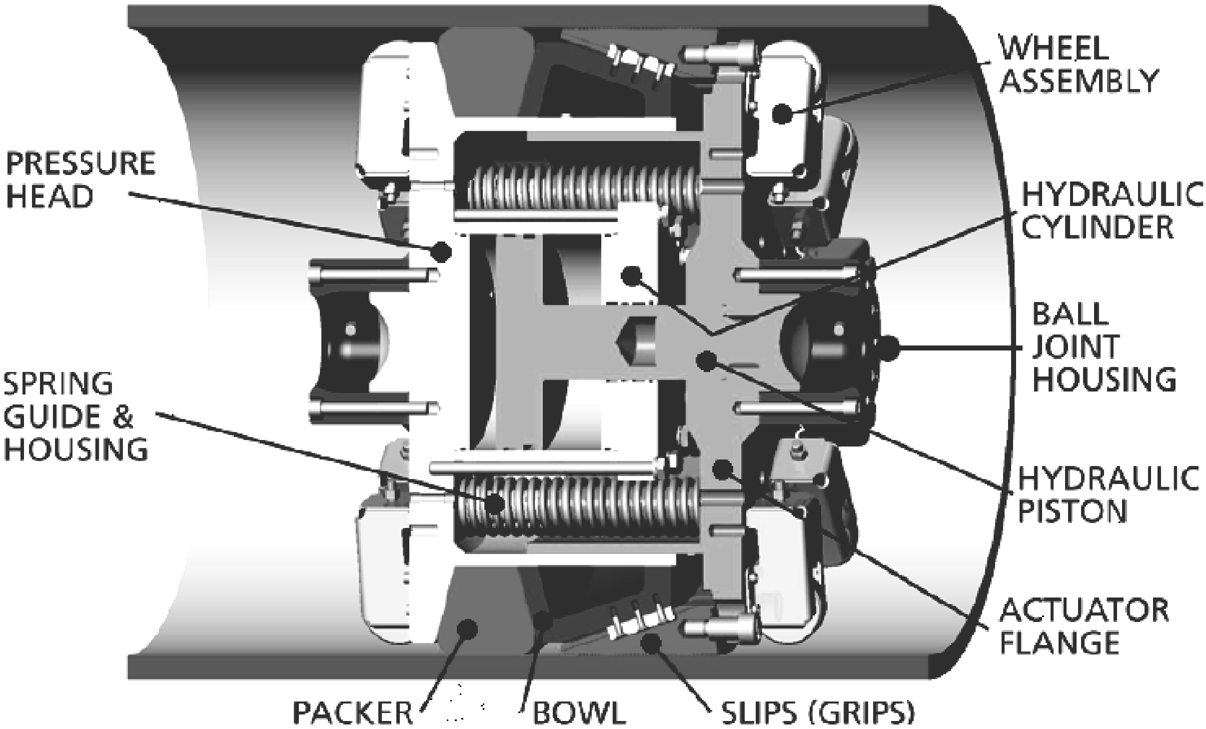


Picture 8: TDW SmartPlug® tool(Jostein Aleksandersen & Edd Tveit 2001)

The Plug Modules, linked together by ball joints, have two primary functions see Picture 9:

1. Set and lock to the inner pipe wall via the threaded metal segments or slips.

2. Differential pressure sealing with the large Packer volume.

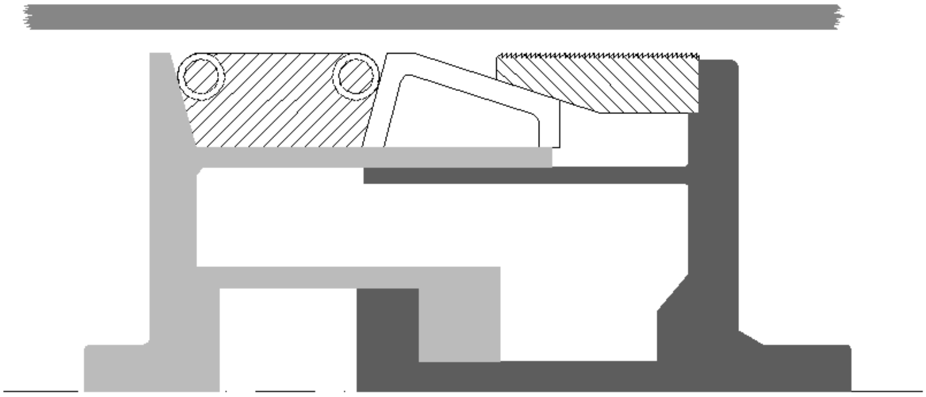


Picture 9: Plug module in locked (set) position(Jostein Aleksandersen & Edd Tveit 2001)

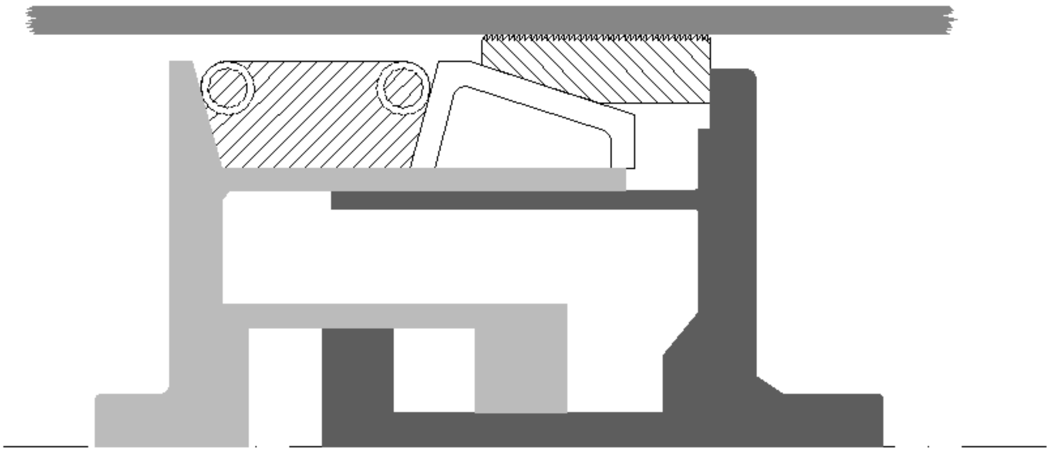
Load transfer is effected by the use of threaded radial-oriented metal segments or slips made from hardened steel. The slips are activated by pressurizing the set side of the hydraulic cylinder resulting in the piston moving to the left and the slips sliding on the bowl and expanding radially, see Picture 10 and Picture 11.

Once the slips are in contact with the pipe wall the Packer will start being compressed thus radially expanding to seal against the inner diameter of the pipeline, see Picture 11 and Picture 12. The outer surfaces of the slips are machined with teeth that are made as sharp as possible to enable the slip teeth to penetrate the surface of the pipeline inner wall. This penetration is less than a tenth of a mm when the slips make uniformly distributed contact with the pipe wall. This value is well within the tolerances specified for scratch marks as published by API.

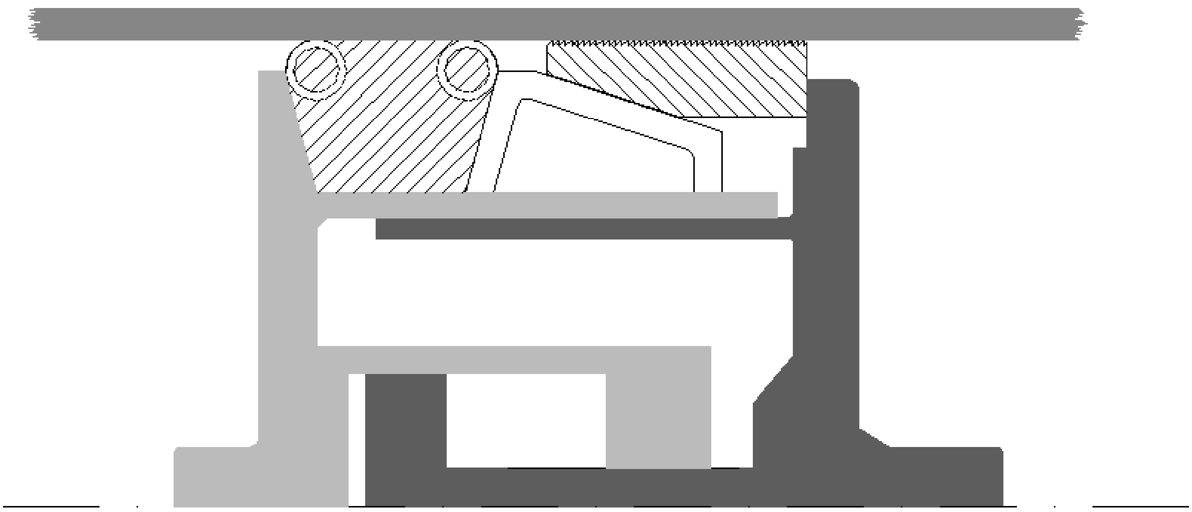
The Plug Modules are self-locking, i.e. once they have been expanded against the pipe wall, a continued application of differential pressure will maintain or intensify their sealing and gripping ability without use of an actuation load. Differential pressure is acting on the pressure head and produces a thrust load against each Plug Module, which transfers the load into the pipe wall through the slips, see Picture 13.



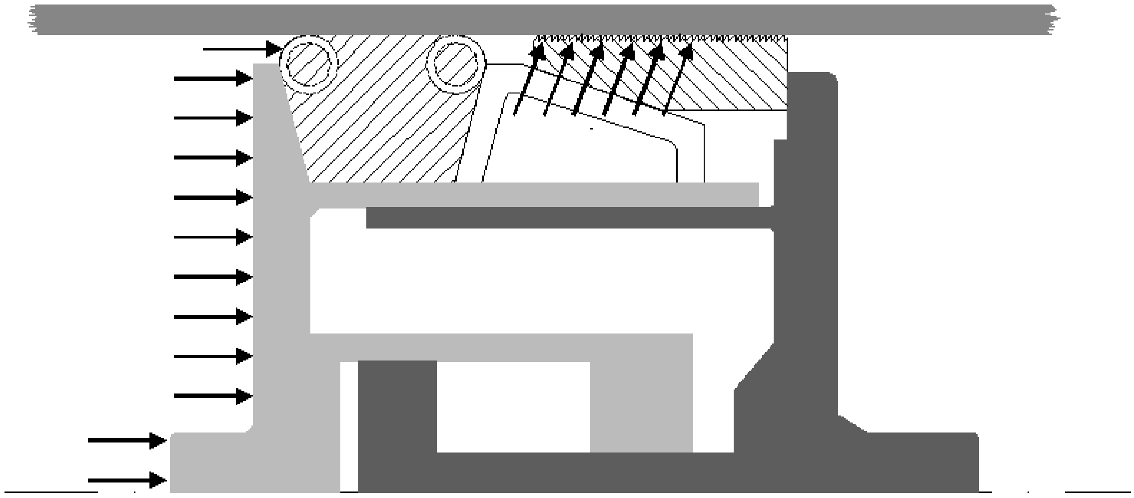
Picture 10: Unset position(Strømsmo, Design Analysis of Packer 2004)



Picture 11: Half set position(Strømsmo, Design analysis of packer 2004)

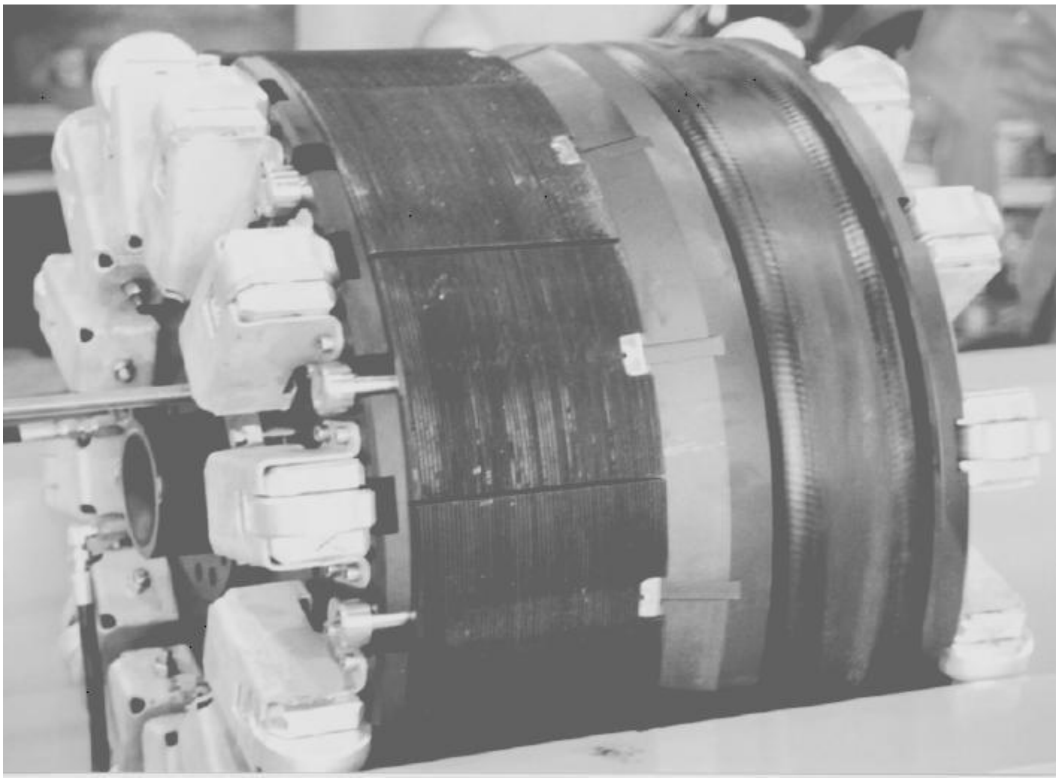


Picture 12: Set position(Strømsmo, Design analysis of packer 2004)



Picture 13: Load transfer in a plug module(Strømsmo, Design analysis of packer 2004)

Each Plug Module is equipped with spring-loaded wheels at each end providing the means of supporting and centralizing the plug module inside the pipeline see Picture 14. In addition the wheels also prevent seal distortion and improve the efficiency of the Packer by keeping the system central in the pipeline, as well as facilitate smooth travel in the pigging mode.



Picture 14: Plug module

Unsetting the tool can be done in several ways, the primary mode is to equalize pressure across the Plug Module and unset it via remote hydraulic actuation. Pressure to the rod side of the hydraulic cylinder is vented and the piston side of the hydraulic cylinder is pressurized, causing the piston to retract the actuator flange. This forces the actuator flange to pull the slips off of the slip bowl ramp and the pressure head to relieve the set on the Packer, thereby releasing the module. The secondary means of unsetting the Plug Module is to vent both the rod and the piston sides of the hydraulic cylinder and then equalize differential pressure across the system. By doing so a number of compressed springs will force the actuator flange and pressure head apart, allowing the slips and the seal to retract to their pigging positions.

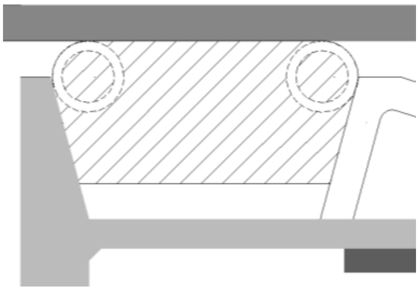
A third means of unsetting the plug modules, should the communication and control systems fail is to use the mechanical fail-safe unsetting mode. Pressure on the actuator flange side of the Plug Module is increased to a pre-set operating pressure to activate a dump valve in the hydraulic circuit, thereby venting pressure from both sides of the hydraulic cylinder. The compressed springs will force the actuator flange and pressure head apart, allowing the slips and the seal to retract to their pigging positions.

Each Plug Module has one Packer, made from a medium hard elastomeric material (acrylonitrile-butadiene rubber). The outer edges of the Packer have steel springs molded in to give the Packer anti-extrusion capability. Picture 15 show a cross section cut of a packer, and Picture 16 show how the spring prevent the rubber to extrude between the pipe wall and the plug module.

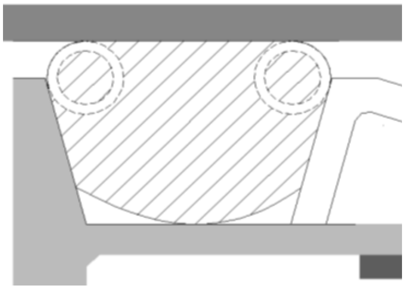
Picture 15: Cross section of packer

Picture 16: Extrusion spring

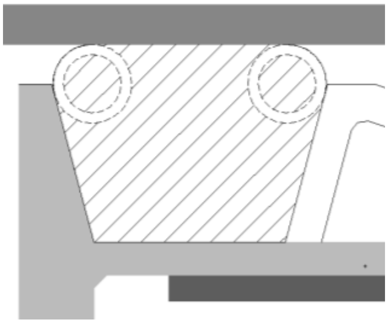
The Packer is actuated (set) by applying the system hydraulic operating pressure to the rod side of the actuator cylinder. This applies a compression load over the annular area of the Packer, and results in compression of the Packer. This compressive action causes the Packer to expand radially and form high-pressure contact between the ID of the pipeline and the plug module, see Picture 13, Picture 17, Picture 18 and Picture 19.



Picture 17: Packer movement in radial direction(Strømsmo, Design Analysis of Packer 2004)



Picture 18: Packer starts to fill available free space(Strømsmo, Design Analysis of Packer 2004)



Picture 19: Packer is fully deformed(Strømsmo, Design Analysis of Packer 2004)

2.2. Acrylonitrile-Butadiene Rubber (NBR)

2.2.1. Introduction

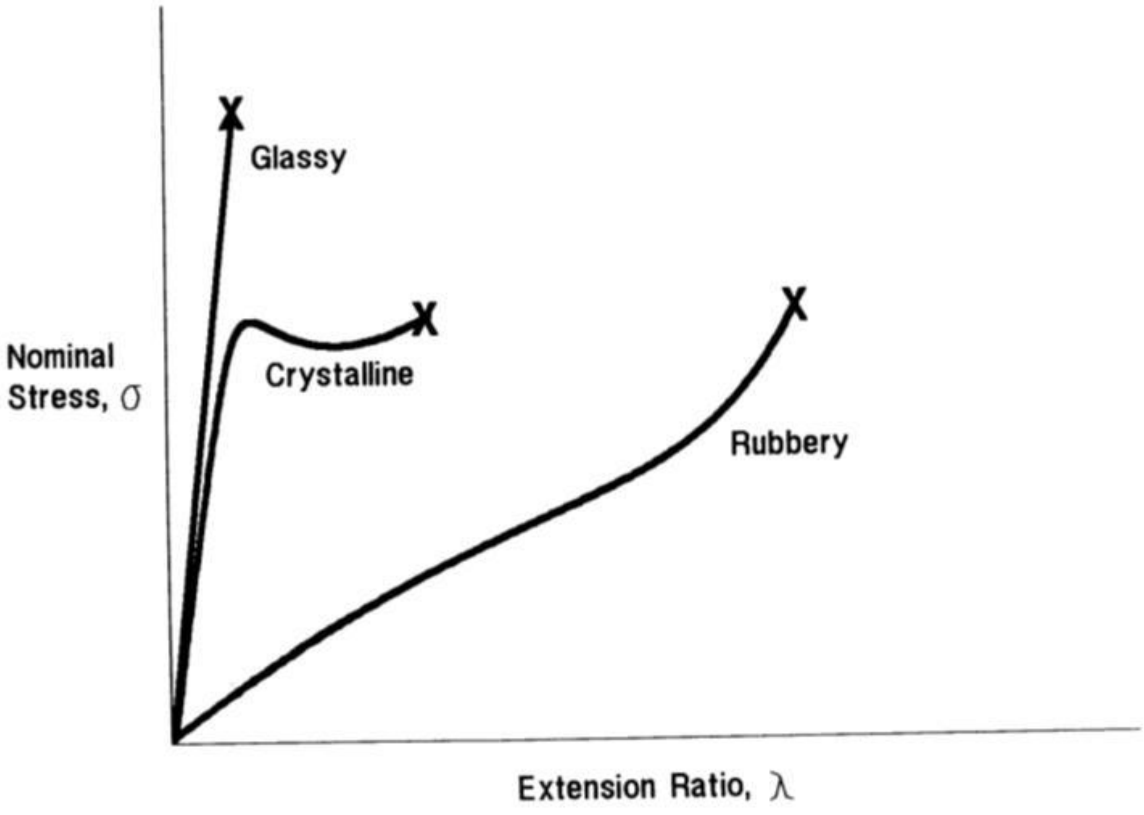
Section 2.2.1. Introduction, is inspired from (William D. Callister 2007), (Brydson 1988) and (Stevenson 1984).

A simplistic way of describing acrylonitrile-butadiene rubbers, commonly known as nitrile rubbers, is to say that they are special purpose rubbers with the conventional technology. Commercially they have been available for over 60 years, they are known primarily for their resistance to liquid fuels such as petrol and other hydrocarbons. The rubbers are the product of the work carried out over many years by Farbenfabriken Bayer, which was for part of the time a constituent of the industrial giant IG Farben. Acrylonitrile-butadiene copolymers were first prepared in 1930 and full-scale production started in 1937, the product being marketed as Buna N. Around 1940 BunaN became renamed to Perbunan, this name is retained by Bayer to this day.

2.2.2. Properties and Applications

Section 2.2.2. Properties and Applications, is inspired from (Gent 2001) and (International Institute of Synthetic Rubber Producers, Inc 2012).

Elastomers are essentially super-condensed gases. The molecules are arranged as an amorphous, glassy or crystalline phase. Glassy polymers are hard and brittle. When Crystalline polymers are subjected to strain they go through a series of changes: elastic, yield, plastic flow, necking, strain hardening, and fracture. Failure (rupture) point (X) is shown in Picture 20. Elastomers are unique in being soft, well-extensible, and very elastic.



Picture 20: Tensile stress-strain diagrams for polymers in three physical states(Gent 2001)

Nitrile (NBR) is usually considered the workhorse of the industrial and automotive rubber products industries. NBR is a complex family of unsaturated copolymers of acrylonitrile and butadiene. By choosing an elastomer with appropriate acrylonitrile content in balance with other additives, the NBR has a wide specter of application areas including oil, fuel, and chemical resistance. With a temperature range of -40^{o}C to 125^{o}C, the NBR materials, can withstand the harshest environments.

On the industrial side NBR is used in roll covers, hydraulic hoses, conveyor belting, graphics, packer-elements, and seals for all types of pipelines and appliance applications. Global consumption of NBR was about 370,000 tonnes in 2005.

NBR rubber is an elastomer which is an amorphous solid that behaves isotropically. The three basic types of loads for isotropic materials are:

- Simple tension
- Simple shear
- Uniform (hydrostatic) compression

The elastic behavior of these loads is defined by the coefficients:

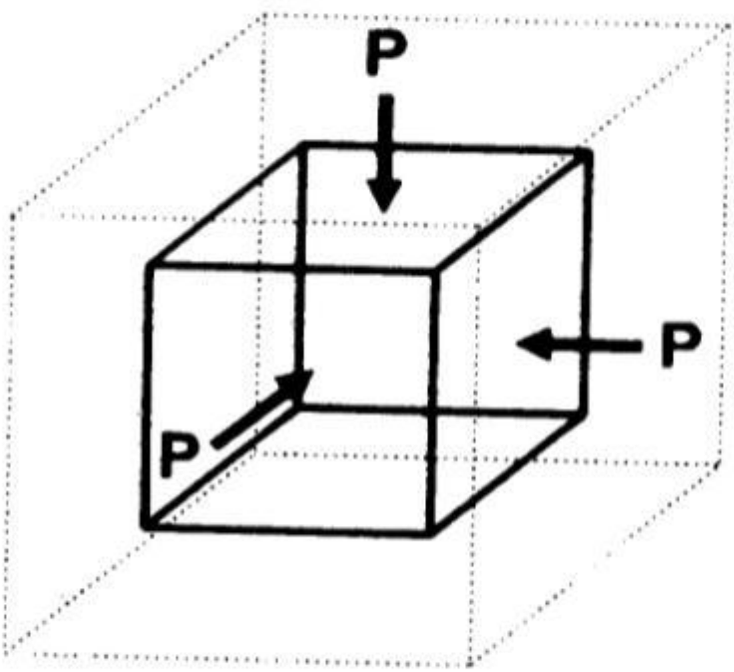
- Young's modulus E (tensile)
- Rigidity modulus G (shear)
- Bulk modulus K (compression)
- Poisson's ratio ν.

Due to the scope of the thesis the stiffness modulus G will not be discussed further.

2.2.4. Bulk Modulus

Section 2.2.4. Bulk Modulus, is inspired from (Gent 2001) and (Brydson 1988).

Elastic materials that are isotropic in the non-deformed state, can be described by a constant that tells something about their resistance to compression in volume under hydrostatic pressure. The constant is called the K modulus of bulk compression, it is defined by the relationship between the applied pressure p and the subsequent shrinkage ΔV of the original volume V0, see Picture 21. The rubber must be subjected to high hydrostatic pressure before volume shrinkage occurs.



Picture 21: Bulk compression(Brydson 1988)

Applied pressure (Brydson 1988):

$$p = K * \frac{\Delta V}{V0} \qquad \text{(Eq. 1)}$$

Solving Eq. 1 with respect to K gives us an expression for the Bulk Modulus (Brydson 1988):

$$K = \frac{p * V0}{\Delta V} \qquad \text{(Eq. 2)}$$

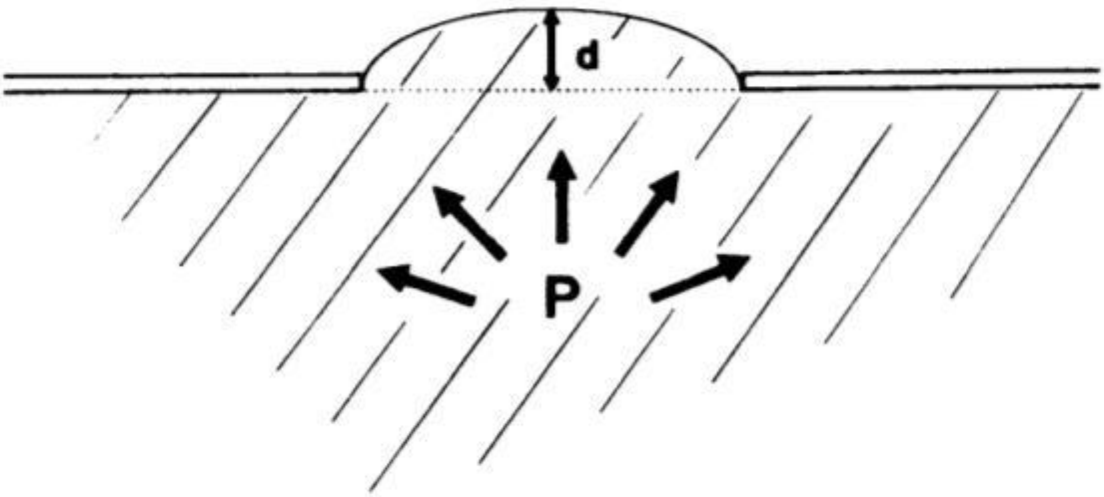
Relationships between Modulus of Elasticity (E), Poisson's ratio (v) and Bulk Modulus (K) for rubber are related by (Simrit 2012):

$$E = 3 * K * (1 - 2v) \qquad \text{(Eq. 3)}$$

Poisson's ratio for rubber is estimated to 0.499 (Gent 2001).

2.2.5. Protrusion of Rubber

Section 2.2.5. Protrusion of Rubber, are inspired from (Gent 2001).



Picture 22: Protrusion through an aperture(Gent 2001)

When a rubber block is contained within a rigid container with a small hole in one end, the rubber will bulge through the hole by an amount d, which depends on the pressure inside the rubber:

Internal pressure (Gent 2001):

$$p = \frac{\pi * d * E}{2r(1 + v)} \qquad \text{(Eq. 4)}$$

d: amount of rubber bulging through the hole

r: radius for a circular hole

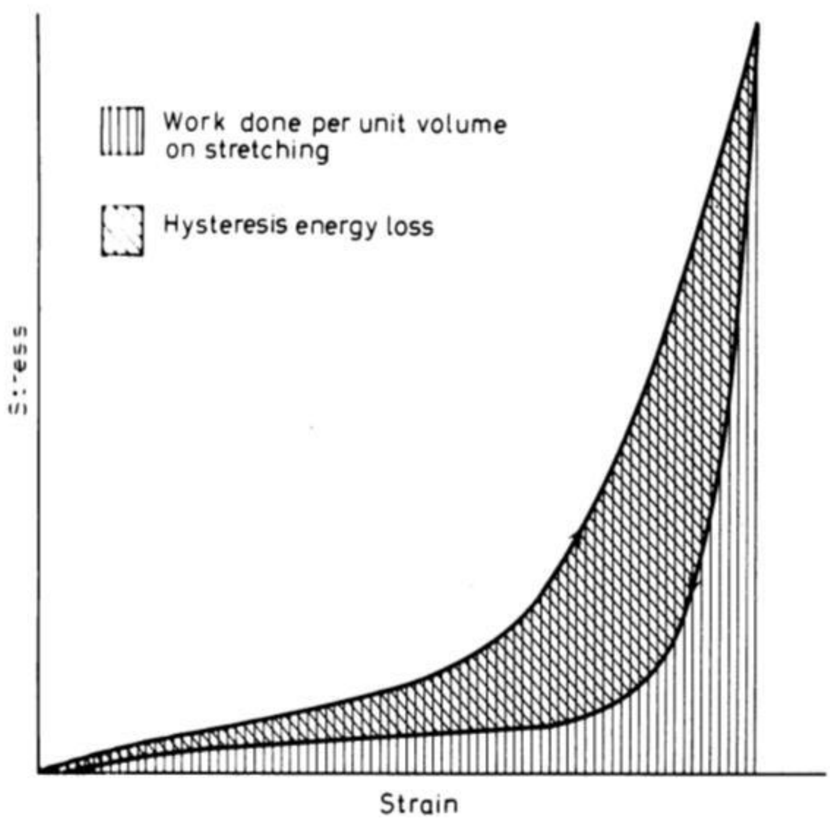
E: Modulus of elasticity

v: Poisson's ratio for rubber

2.2.6. Hysteresis Energy Loss

Section 2.2.6. Hysteresis Energy Loss, is inspired from (Ciesielski 1999).

The required force to deform the rubber is higher than the retraction force on the deformed rubber. Energy loss is a result of internal friction. Picture 23 shows a typical stress-strain extension cycle for rubber. This cycle is called the hysteresis energy loss, the lost energy is converted into heat. The longer the period of the rubber is deformed, the greater the energy loss. When the rubber is exposed to stress and pressure over time the material will not fully recover its original shape.



Picture 23: Typical stress-strain extension cycle(R. K. Flitney 1984)

2.2.7. Modulus of Elasticity for an Elastomer

Modulus of elasticity for an elastomer, (also called "Mod 100") is stress required to produce a given elongation. In the case of "Mod 100", the modulus would be the stress required to elongate the sample 100%. In elastomers, the stress is not linear with strain as described in 2.2.2. Properties and Applications. Therefore the modulus is neither a ratio nor a constant slope-but rather denotes a point on the stress-strain curve (Azom 2012). However as described is rubber a non-linear material under tensile stress, but the material behaves almost linearly in shear or compression deformation (Azom 2012).

2.3 Packer development

2.3.1 Today's method

The current method for developing packers is (roughly described) carried out as follows:

1. The packer dimensions, profile, rubber compounds and spring configuration are chosen with background in past experience and job specs.
2. The packer is calculated, and if necessary changes to the above are implemented.
3. The packer is then drawn in Solid Works and some simulation with the packer "mounted" on the plug module is done. If necessary the design is tuned.
4. Then an external company produces the packer:
   - They make a special made mold, fitted only for this specific packer.
   - And then cast the packer in the mold.
5. Due to the advanced material properties of the elastomer used in the packer, it has to go through several full scale tests to approve it's attributes.
   - The packer is mounted on a plug module and subjected to several qualitative and quantitative tests. Both outside and inside a test pipe.
6. If the packer does not fulfill the design requirements you have to start over at point 1.

The process described above can be a very expensive process if you don't get it right the first time, in the worst case have to make several new and expensive molds and packers. In addition it's a rather expensive test process due to the man-hours needed for mounting the packers onto a plug module and putting it in and taking it out of the test pipe. Unfortunately history shows that the process of developing new packers requires some trial and error, which again leads us to the potential of saving money in this process.

2.3.2 Future methods

Three ways to improve this system, save money, and make it easier to handle and work with will be discussed:

2.3.2.1 Method 1

The first method is to make a more accurate computer model to simulate the packer's attributes.

- The benefits of this method:
  - When/if the method gives accurate answers, it is significantly less expensive to use than anything else.
  - It is faster to use, an engineer can simulate the packer and get results in a couple of hours.
- The disadvantages with this method:
  - The computer model can be expensive and time consuming to develop.
  - Most likely a test rig used to develop the computer model is needed.
  - Due to the advanced material properties and that there is two materials (rubber and steel) interacting. Real life tests are still required after the packer is produced.
  - Full scale tests also have to be done to secure the many variables and human factors connected to the fabrication phase.
  - A test rig is still needed to test the final design.

2.3.2.2 Method 2

The second opportunity is to make a variable test rig for the whole circular packer.

- The benefits of this method:
  - Man-hour savings in the test phase as a result of easier and faster handling and mounting.
  - Do not occupy a plug module for testing, or don't need to wait for plug module coming back from work.
  - Inexpensive flexibility to different geometries.
  - Due to flexible test rigs, a small amount of rigs are needed to cover all the packer dimensions.
- The disadvantages with this method:
  - The method does not deal with the fact that a new mold and packer have to be made if the existing packer does not fulfill the requirements.
  - To cover the whole range of packers, several test rigs may be needed

2.3.2.3 Method 3

The idea behind the third and most innovative method comes from finite element method analysis (FEM-analysis). The idea is to slice the packer into a finite number of "pizza-slices" and test only one of these slices. But to test only one small element in a real life test is not possible, so an element length of for example 100-200 mm is needed. The error by taking an element of this length is not known but it is believed that through tests it can be estimated. The element is put into a test rig that is a rectangular pressure chamber with a hydraulic activated piston in one end. The packer element will be "set" inside the chamber and the pipeline pressure will be simulated with the hydraulic piston. When a packer configuration tests well, a full scale packer can be made and full scale tests can be done to verify the packers attributes.

- The benefits of this method:
  - A full-scale mold and packer don't need to be made during the developing phase, if the chosen geometry don't fulfill the requirements, less money is lost.
  - Smaller parts makes it easier and faster to handle, and man-hours are saved.
  - Need only one rig for a very wide range of packers.
  - The rig is fairly cheap to make and due to the history of trial and error to develop packers, this method out of the three, gives potential to save the most money.
  - The test should give very close to real results with respect to packer pressure, spring configurations and other quantitative results such as packer failure.
- The disadvantages with this method:
  - The fact that this is not a full-scale test leads to some uncertainties connected to the results.
  - The tests do not take into account the change in volume and as a result the hoop stresses that exist in a full-scale packer when it's set.
  - The tests do not take into account that the extrusion springs are stretched when a full-scale packer is "set".
  - The test considers a straight packer piece without curvature.
  - The test method may have difficulty in getting real self lock pressure results, but it should give good indications on differences between different configurations.

2.3.3 Conclusion

To meet TDW's requests about a test method that;

- take care of the economical issue with existing methods.
- have opportunity for a higher test frequency.
- do not occupy resources within the company that are usable in other activities.
- is easier and faster for workshop personnel to execute.

It is decided to make a new test rig, and since the method described in chapter 2.3.2.3 Method 3 has potential for the biggest money savings, seems most flexible to geometries, and that a test rig most likely is needed regardless of the three methods described in chapter 2.3.2 Future methods. It is decided to look more into 2.3.2.3 Method 3, to see how it should be done, and to settle its limitations.

Chapter 3: Basic theory

3.1 Packer calculations

The method of conservation of volume is used to calculate the packers and find the expected factors such as:

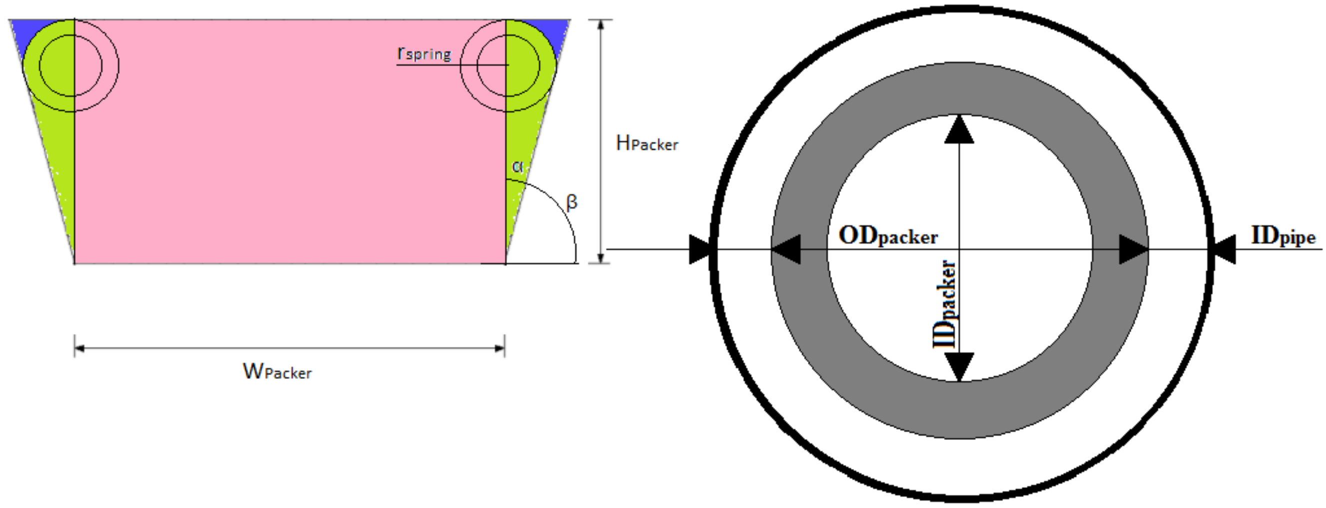
- Packer pressure against the pipe wall
- Force needed to deform the packer
- Packer deformation with given pipeline pressure

The assumptions for using the method is:

- The volume of the rubber in the packer element is the same in unset and set position.
- The rubber in the packers behave as a incompressible fluid.

Formulas for accurate volume and area calculation are derived in appendix B, however due to simplicity, linearized formulas for packer volume are used in the thesis.

3.1.1 Packer Volume



Picture 24: Packer dimension notations

$$V_{packer} = \frac{\pi}{4}\left[\left(\left(OD_{packer}^2 - ID_{packer}^2\right) * W_{packer\ unset}\right) + \left(\frac{\left(OD_{packer} - ID_{packer}\right)^2}{\tan(\beta)} * OD_{packer}\right) - \left(\frac{1}{3} * \frac{\left(OD_{packer} - ID_{packer}\right)^3}{\tan(\beta)}\right)\right] \quad \text{(Eq. 5)}$$

3.1.2 Packer deformation

$$W_{packer\ set} = \frac{V_{packer} - \frac{\pi}{4} * \left[\left(\frac{\left(ID_{pipe} - ID_{packer}\right)^2}{\tan(\beta)} * ID_{pipe}\right) - \left(\frac{1}{3} * \frac{\left(ID_{pipe} - ID_{packer}\right)^3}{\tan(\beta)}\right)\right]}{\frac{\pi}{4} * \left(ID_{pipe}^2 - ID_{packer}^2\right)} \quad \text{(Eq. 6)}$$

$$\delta_{packer} = W_{packer\ unset} - W_{packer\ set} \quad \text{(Eq. 7)}$$

3.1.3 Force to deform packer

Average diameter of packer:

$$AD_{packer} = \frac{OD_{packer} + ID_{packer}}{2} \quad \text{(Eq. 8)}$$

Radial height of packer:

$$H_{packer} = \frac{OD_{packer} - ID_{packer}}{2} \quad \text{(Eq. 9)}$$

Force to deform packer:

$$F_{packer} = \delta_{packer} * \frac{4}{3} * E * \pi * AD_{packer} * \frac{H_{packer}}{W_{packer\ unset}} * \left(1 + \frac{K * H_{packer}^2}{4 * W_{packer\ unset}^2}\right) \quad \text{(Eq. 10)}$$

3.1.4 Cylinder pretension and "unset spring" force

Pretension force from hydraulic cylinder:

$$F_{pretension} = P_{cylinder} * A_{cylinder} \quad \text{(Eq. 11)}$$

Area of cylinder:

$$A_{cylinder} = \left(\frac{\pi * ID_{cylinder}^2}{4}\right) - \left(\frac{\pi * D_{piston\ rod}^2}{4}\right) \quad \text{(Eq. 12)}$$

The unsetting springs can consist of two different springs, one laying inside the other.

Spring force in plug unset/ spring pretension position:

$$F_{spring\ plug\ unset} = k * \left(L_{spring\ unset} - L_{spring\ pretension}\right) * N_{spring} \qquad \text{(Eq. 13)}$$

Spring force in plug set position:

$$F_{spring\ plug\ set} = k * \left(L_{spring\ unset} - L_{spring\ plug\ set}\right) * N_{spring} \qquad \text{(Eq. 14)}$$

Total spring force:

$$F_{spring} = F_{spring\ plug\ set\ (1)} + F_{spring\ plug\ set\ (2)} + \cdots + F_{spring\ plug\ set\ (n)} \qquad \text{(Eq. 15)}$$

3.1.5 Pressure between packer and pipe wall

Force on the plug due to pipeline pressure:

$$F_{pipe} = P_{pipe}\left[\frac{\pi}{4} * \left(ID_{pipe}^2 - D_{piston\ rod}^2\right)\right] \qquad \text{(Eq. 16)}$$

Total force acting on the packer due to pipeline pressure:

$$F_{packer\ pipe} = F_{pipe} + F_{pretension} - F_{packer} - F_{spring} \qquad \text{(Eq. 17)}$$

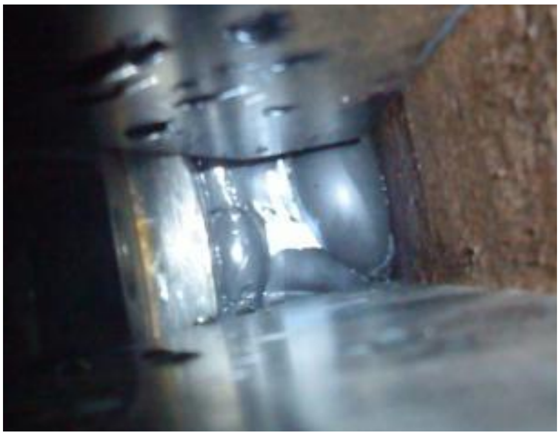
Pressure between packer and pipe wall at isolation pressure, also called packer pressure:

$$P_{packer\ wall} = \frac{F_{packer\ pipe}}{\frac{\pi}{4}\left(ID_{pipe}^2 - ID_{packer}^2\right)} \qquad \text{(Eq. 18)}$$

3.1.6 Errors in volume calculations

The errors by using linearized formulas for volume calculation is considered and is relatively small compared to other volume related variables such as:

- Extrusion of rubber through the springs when pressurized, see Picture 25. This is a incident that rarely happens, but it is important to know about it.



Picture 25: Extrusion of rubber through anti extrusion spring

- The geometry of the spring is not constant in all the stages from unset to set position, this can clearly be seen on Picture 26. This gives an error in the conservation of volume method that is much bigger than the error in the numerical calculation. The calculation method do not take care of this issue.

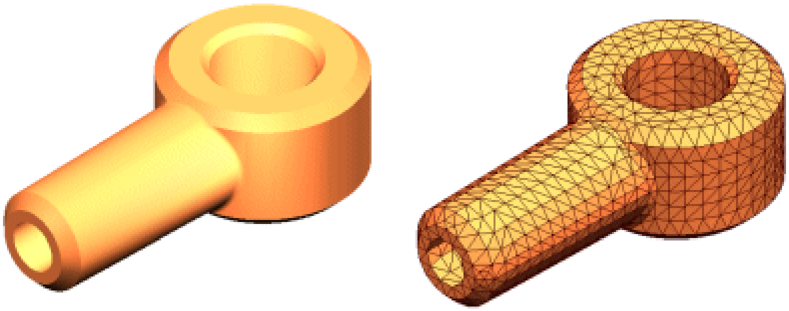
Picture 26: Change in anti extrusion spring geometry

3.2 SolidWorks simulation

3.2.1 Basic concepts of analysis

The following section is an extract from (SolidWorks 2012)

The software uses the Finite Element Method (FEM). FEM is a numerical technique for analyzing engineering designs. FEM is accepted as the standard analysis method due to its generality and suitability for computer implementation. FEM divides the model into many small pieces of simple shapes called elements effectively replacing a complex problem by many simple problems that need to be solved simultaneously.

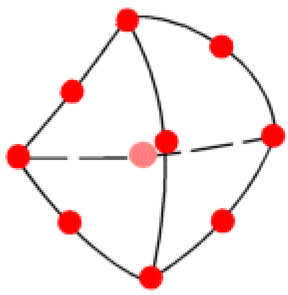


Picture 27: CAD model of a part and Model subdivided into small pieces (elements)

Elements share common points called nodes. The process of dividing the model into small pieces is called meshing.

The behavior of each element is well-known under all possible support and load scenarios. The finite element method uses elements with different shapes.

The response at any point in an element is interpolated from the response at the element nodes. Each node is fully described by a number of parameters depending on the analysis type and the element used. For example, the temperature of a node fully describes its response in thermal analysis. For structural analyses, the response of a node is described, in general, by three translations and three rotations. These are called degrees of freedom (DOFs). Analysis using FEM is called Finite Element Analysis (FEA).



Picture 28: A tetrahedral element. Red dots represent nodes. Edges can be curved or straight.

The software formulates the equations governing the behavior of each element taking into consideration its connectivity to other elements. These equations relate the response to known material properties, restraints, and loads.

Next, the program organizes the equations into a large set of simultaneous algebraic equations and solves for the unknowns.

In stress analysis, for example, the solver finds the displacements at each node and then the program calculates strains and finally stresses.

The software offers the following types of studies:

- Static (or Stress) Studies. Static studies calculate displacements, reaction forces, strains, stresses, and factor of safety distribution. Material fails at locations where stresses exceed a certain level. Factor of safety calculations are based on one of four failure criterion.
  Static studies can help you avoid failure due to high stresses. A factor of safety less than unity indicates material failure. Large factors of safety in a contiguous region indicate low stresses and that you can probably remove some material from this region.
- Frequency Studies. A body disturbed from its rest position tends to vibrate at certain frequencies called natural, or resonant frequencies. The lowest natural frequency is called the fundamental frequency. For each natural frequency, the body takes a certain shape called mode shape. Frequency analysis calculates the natural frequencies and the associated mode shapes.
  In theory, a body has an infinite number of modes. In FEA, there are theoretically as many modes as degrees of freedom (DOFs). In most cases, only a few modes are considered.
  Excessive response occurs if a body is subjected to a dynamic load vibrating at one of its natural frequencies. This phenomenon is called resonance. For example, a car with an out-of-balance tire shakes violently at a certain speed due to resonance. The shaking decreases or disappears at other speeds. Another example is that a strong sound, like the voice of an opera singer, can cause a glass to break.
  Frequency analysis can help you avoid failure due to excessive stresses caused by resonance. It also provides information to solve dynamic response problems.
- Dynamic Studies. Dynamic studies calculate the response of a model due to loads that are applied suddenly or change with time or frequency.
  Linear dynamic studies are based on frequency studies. The software calculates the response of the model by accumulating the contribution of each mode to the loading environment. In most cases, only the lower modes contribute significantly to the response. The contribution of a mode depends on the load's frequency content, magnitude, direction, duration, and location.
  The objectives of a dynamic analysis include:
    - the design of structural and mechanical systems to perform without failure in dynamic environments,
    - the reduction of vibration effects.
- Buckling Studies. Buckling refers to sudden large displacements due to axial loads. Slender structures subject to axial loads can fail due to buckling at load levels lower than those required to cause material failure. Buckling can occur in different modes under the effect of different load levels. In many cases, only the lowest buckling load is of interest.
  Buckling studies can help you avoid failure due to buckling.

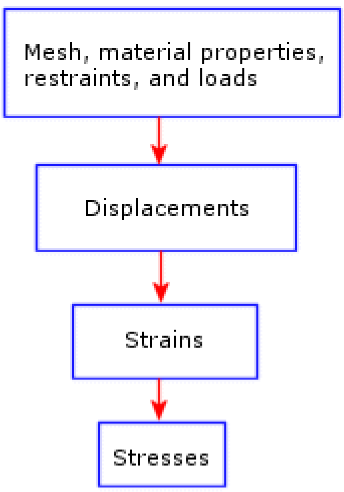
- Thermal Studies. Thermal studies calculate temperatures, temperature gradients, and heat flow based on heat generation, conduction, convection, and radiation conditions. Thermal studies can help you avoid undesirable thermal conditions like overheating and melting.
- Design Studies. Optimization design studies automate the search for the optimum design based on a geometric design. The software is equipped with a technology to quickly detect trends and identify the optimum solution using the least number of runs. Optimization design studies require the definition of the following:
    - Goals or Objectives. State the objective of the study. For example, minimum material to be used.
    - Design Variables. Select the dimensions that can change and set their ranges. For example, the diameter of a hole can vary from 0.5” to 1.0” while the extrusion of a sketch can vary from 2.0” to 3.0”.
    - Constraints. Set the conditions that the optimum design must satisfy. For example, you can require that a stress component does not exceed a certain value and the natural frequency to be within a specified range.

3.2.2 Sequence of Calculations

The following section is an extract from (SolidWorks 2012)

Given a meshed model with a set of displacement restraints and loads, the linear static analysis program proceeds as follows:

1. The program constructs and solves a system of linear simultaneous finite element equilibrium equations to calculate displacement components at each node.
2. The program then uses the displacement results to calculate the strain components.
3. The program uses the strain results and the stress-strain relationships to calculate the stresses.



Picture 29: Squence of calculation in SolidWorks

3.2.3 Stress Calculations

The following section is an extract from (SolidWorks 2012)

Stress results are first calculated at special points, called Gaussian points or Quadrature points, located inside each element. These points are selected to give optimal numerical results. The program calculates stresses at the nodes of each element by extrapolating the results available at the Gaussian points.

After a successful run, nodal stress results at each node of every element are available in the database. Nodes common to two or more elements have multiple results. In general, these results are not identical because the finite element method is an approximate method. For example, if a node is common to three elements, there can be three slightly different values for every stress component at that node.

3.2.4 Linear static analysis

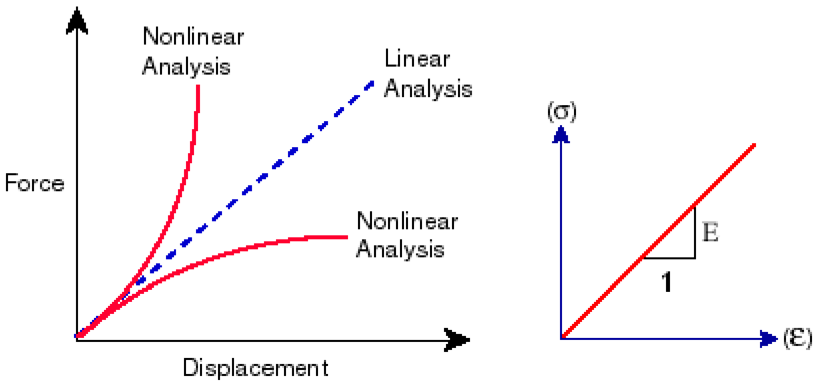
The following section is an extract from (SolidWorks 2012)

When loads are applied to a body, the body deforms and the effect of loads is transmitted throughout the body. The external loads induce internal forces and reactions to render the body into a state of equilibrium.

Linear Static analysis calculates displacements, strains, stresses, and reaction forces under the effect of applied loads.

Linear static analysis makes the following assumptions:

- Static Assumption. All loads are applied slowly and gradually until they reach their full magnitudes. After reaching their full magnitudes, loads remain constant (time-invariant). This assumption allows us to neglect inertial and damping forces due to negligibly small accelerations and velocities. Time-variant loads that induce considerable inertial and/or damping forces may warrant dynamic analysis. Dynamic loads change with time and in many cases induce considerable inertial and damping forces that cannot be neglected.
- Linearity Assumption. The relationship between loads and induced responses is linear. For example, if you double the loads, the response of the model (displacements, strains, and stresses), will also double. You can make the linearity assumption if:
    - all materials in the model comply with Hooke's law, that is; stress is directly proportional to strain.
    - the induced displacements are small enough to ignore the change in stiffness caused by loading.
    - boundary conditions do not vary during the application of loads. Loads must be constant in magnitude, direction, and distribution. They should not change while the model is deforming.



Picture 30: Assumptions in linear static analysis

3.2.5 Meshing

The following section is an extract from (SolidWorks 2012) and (SolidWorks 2012)

Finite Element Analysis (FEA) provides a reliable numerical technique for analyzing engineering designs. The process starts with the creation of a geometric model. Then, the program subdivides the model into small pieces of simple shapes (elements) connected at common points (nodes). Finite element analysis programs look at the model as a network of discrete interconnected elements.

The Finite Element Method (FEM) predicts the behavior of the model by combining the information obtained from all elements making up the model.

Meshing is a very crucial step in design analysis. The automatic mesher in the software generates a mesh based on a global element size, tolerance, and local mesh control specifications. Mesh control lets you specify different sizes of elements for components, faces, edges, and vertices.

The software estimates a global element size for the model taking into consideration its volume, surface area, and other geometric details. The size of the generated mesh (number of nodes and elements) depends on the geometry and dimensions of the model, element size, mesh tolerance, mesh control, and contact specifications. In the early stages of design analysis where approximate results may suffice, you can specify a larger element size for a faster solution. For a more accurate solution, a smaller element size may be required.

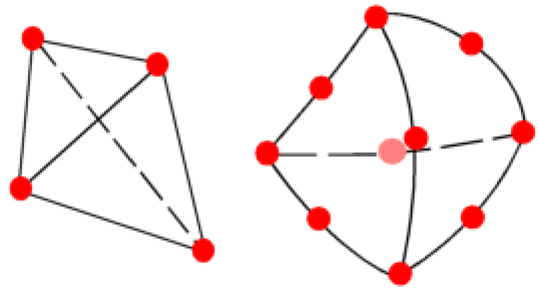
Meshing generates 3D tetrahedral solid elements, 2D triangular shell elements, and 1D beam elements. A mesh consists of one type of elements unless the mixed mesh type is specified. Solid elements are naturally suitable for bulky models. Shell elements are naturally suitable for modeling thin parts (sheet metals), and beams and trusses are suitable for modeling structural members.

In meshing a part or an assembly with solid elements, the software generates one of the following types of elements based on the active mesh options for the study:

- Draft quality mesh. The automatic mesher generates linear tetrahedral solid elements.
- High quality mesh. The automatic mesher generates parabolic tetrahedral solid elements.

Linear elements are also called first-order, or lower-order elements. Parabolic elements are also called second-order, or higher-order elements.

A linear tetrahedral element is defined by four corner nodes connected by six straight edges. A parabolic tetrahedral element is defined by four corner nodes, six mid-side nodes, and six edges. The following figures show schematic drawings of linear and parabolic tetrahedral solid elements.



Picture 31: Linear solid element and Parabolic solid element

In general, for the same mesh density (number of elements), parabolic elements yield better results than linear elements because:

- they represent curved boundaries more accurately
- they produce better mathematical approximations.

However, parabolic elements require greater computational resources than linear elements.

For structural problems, each node in a solid element has three degrees of freedom that represent the translations in three orthogonal directions. The software uses the X, Y, and Z directions of the global Cartesian coordinate system in formulating the problem.

Chapter 4: Design of the packer tester

4.1 Conceptual design

The conceptual design is based on the method described in chapter 2.3.2.3 Method 3. The method seems to take best care of;

- The economical issue with existing methods.
- The opportunity for a higher test frequency.
- A faster and easier execution for the workshop personnel.
- The flexibility to geometries .

4.1.1 Desirable goals and results

4.1.1.1 Goals

The ultimate goals for a test rig adapted to TDW's needs is:

- Only one packer test rig that covers all the packer dimensions and geometries with little configurations.
- An inexpensive to produce and operate test rig.
- Have a test rig that gives consistent real and accurate results.
- A test system that limits the loss when a packer fails or does not fulfill the requirements.
- A test rig that test packers with 30% expansion, and also is robust enough to test a packer to failure (Very high pressure; 400-600 bar packer pressure).

4.1.1.2 Qualitative results

The qualitative results that is desirable from the test rig is:

- Spring configurations
- Measured force needed to deform the packer from unset to set position
- The packer pressure which implement the stress to the pipe wall (comes from the surface pressure between the packer and the pipe wall)
- Stress implemented to the plug module
- The maximum differential pressure the packer can withstand before it leaks or total failure
- What fluid properties it can isolate against
- Maximum pipe diameter the packer can operate in i.e. the biggest possible OD of the packer in set position without failure of anti extrusion springs
- The minimum differential pressure the packer can withstand before it leaks (also called; "self lock pressure")

4.1.1.3 Quantitative results

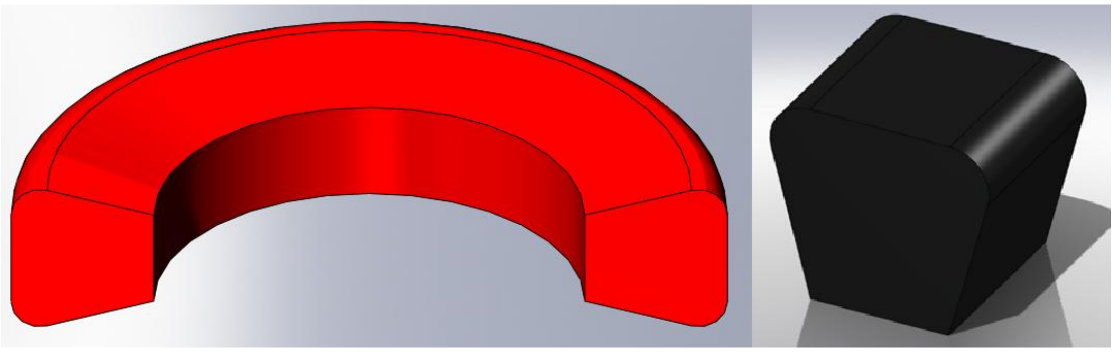
The quantitative results that is desirable from the rest rig is:

- Total packer failure i.e. the pressure in the rubber is too big for the anti extrusion springs to hold it→rubber squeezes out of the initially closed area
- The rubber has cracks
- The springs have plastic deformation
- The packer starts to leak

- The packer has folds and an unnatural shape after setting
- Spring behavior
- Rubber behavior

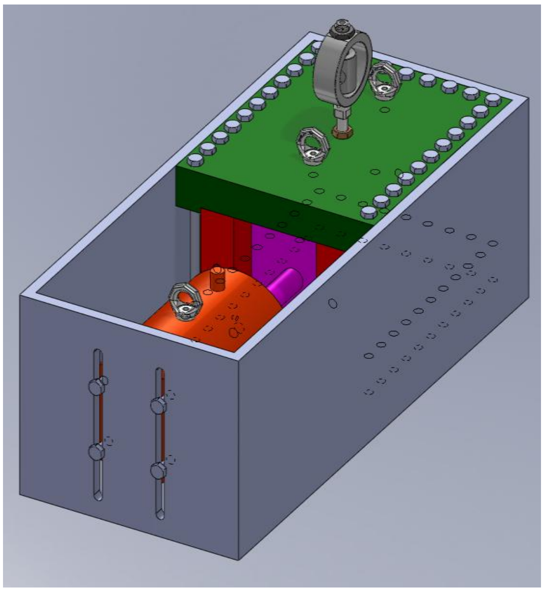
4.1.2 Concept

As described in chapter 2.3.2.3 Method 3, the concept for the new packer test rig uses a similar approach as FEM. The idea is to test a linearized rectangular coupon, see Picture 32, with same cross section geometries as the equivalent packer. The coupon length are set to 200mm.

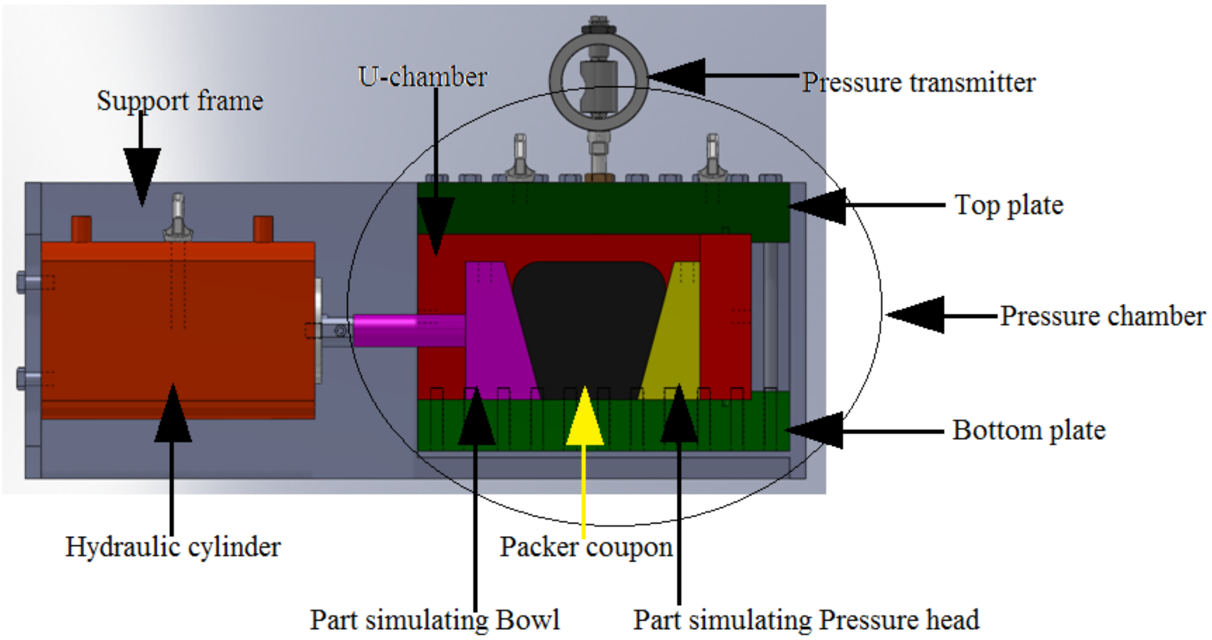


Picture 32: Linearized packer coupon

The coupon will be “set” in a rectangular pressure chamber, and the pipeline pressure will be simulated with a hydraulic piston. To test sealing, the pressure chamber will be pressurized with an incompressible fluid. The conceptual design of the test rig, see Picture 33 and Picture 34, shows the coupon inside the pressure chamber that sits in the support frame.



Picture 33: Conseptual design of packer test rig



Picture 34: Overview of conseptual design

Test rig concept dimensions:

- Overall length 1,24m
- Overall width 0,53m
- Overall height (without load cell (Pressure transmitter) and lifting eyes) 0,46m

4.1.3 Flexibility

To make the test rig flexible to as many packer dimensions as possible, some parts have to be packer-size specific and need to be replaced for each packer dimension. The challenge is to have as low amount of these size specific components in the test rig as possible, to keep the test costs low. The main parts in the test rig are shown in Table 2.

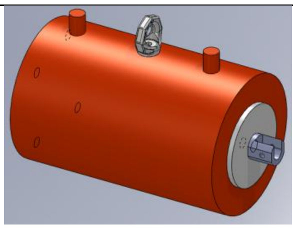
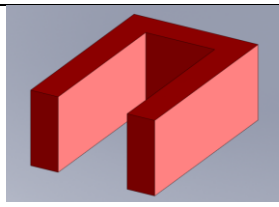
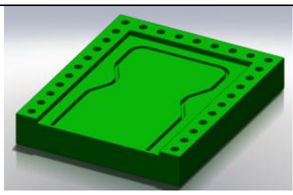
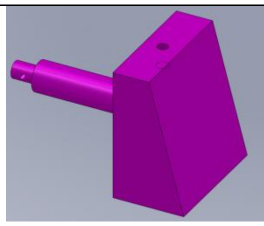
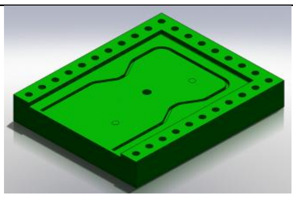
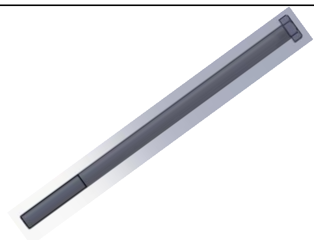
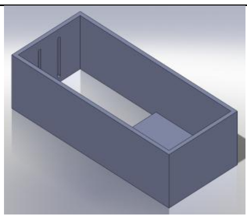
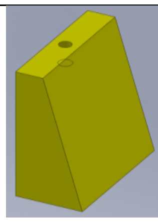
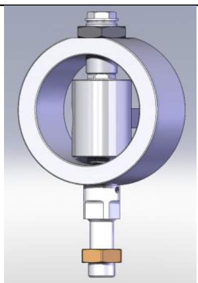
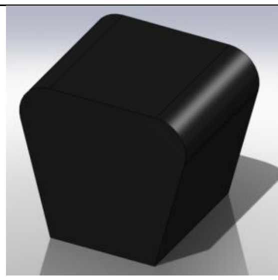
Fixed parts		Size specific parts	
Hydraulic cylinder with coupling		U-chamber: Replaced per pipe size	
Bottom plate with groove for seal		Piston/Bowl: Easily reconfigured	
Top plate with groove for seal		Bolts: Replaced per pipe size	
Support frame		Pressure head: Easily reconfigured	
Load cell		Packer coupon: Easy to make a many different configurations	

Table 2: Main components in packer test rig

The size specific parts that need to be changed when the test rig is set up for another dimension can be easily made with little machining and are inexpensive.

Table 3 show the rig configured for the 8 inch packer and the 48 inch packer which is the outer limits on packer sizes TDW use today. The hydraulic cylinder is lowered for smaller packer dimension to apply force in the centre of the bowl.

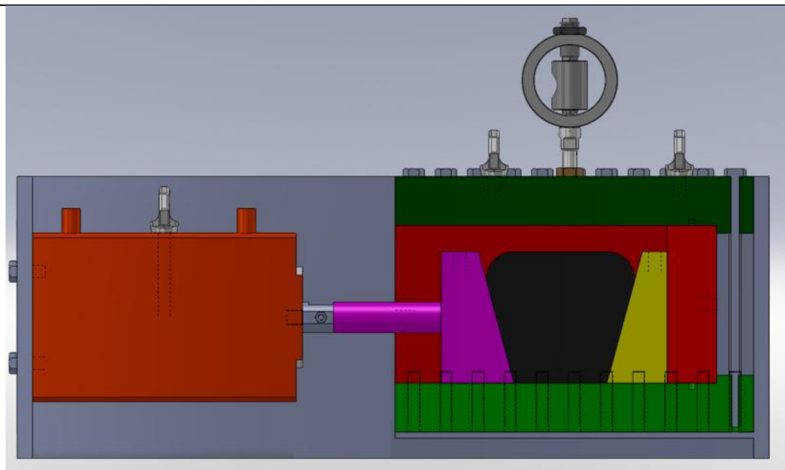
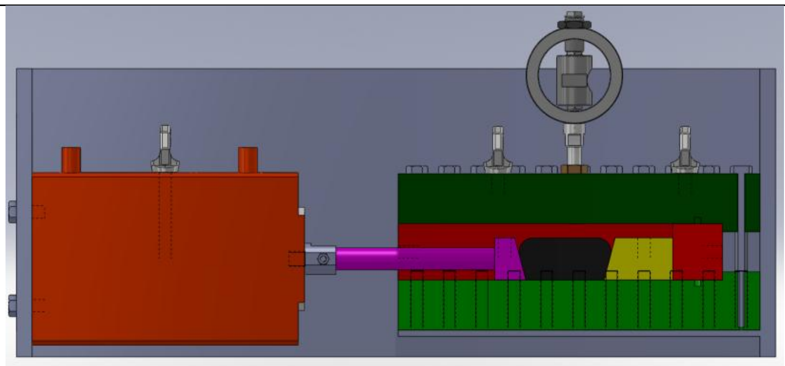
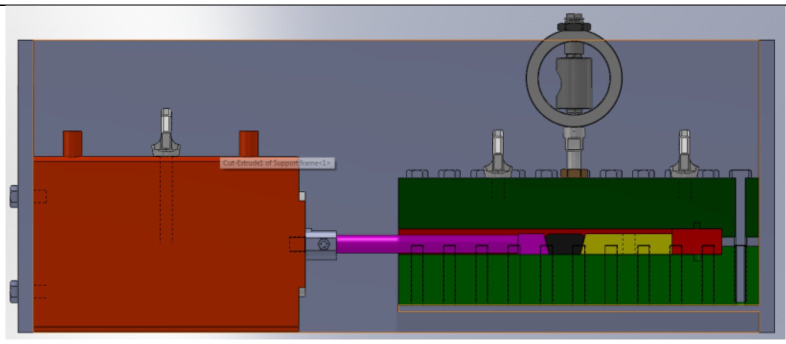
Packer test rig configured for different packer dimensions	
48 inch configuration	
24 inch configuration	
8 inch configuration	

Table 3: Packer test rig configured for three different packer coupon dimensions

4.1.4 Example of test preparation, execution and duration

4.1.4.1 Test preparation

Table 4 show a rough example of the packer test rig being prepared for the 48” packer coupon. To prepare the test rig a crane and two people are needed due to heavy parts. The parts that needs to be craned into place have threaded holes for lifting eyes, the lifting eyes need to be removed after the lifting of each part is finished.

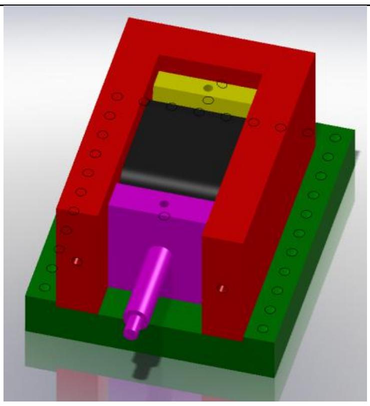
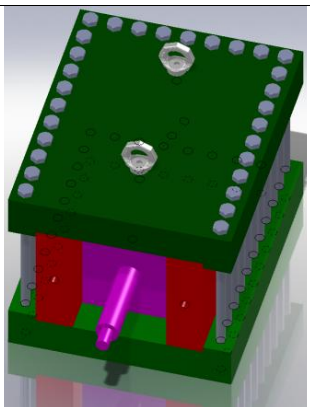
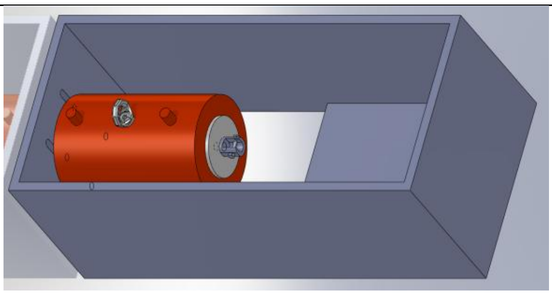
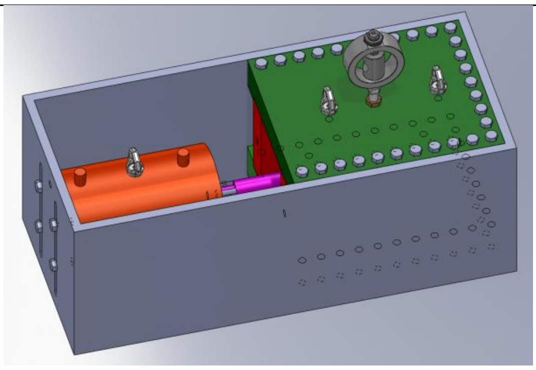
Test rig preparation	
1. Load the U-chamber, the wedges and the packer coupon onto the bottom plate.	
2. Put on the top plate and bolt the pressure chamber.	
3. Adjust the hydraulic cylinder in the support frame to hit center of piston/bowl.	
4. Crane the whole pressure chamber assembly into the support frame.	

Table 4: Preparation of test rig

4.1.4.2 Test execution

When the test rig is prepared, the execution of the test can start, roughly described it will be conducted as follows:

1. Set the packer:
    a. The packer pressure and equivalent hydraulic cylinder pressure are calculated from the desired pipeline pressure by using the excel model seen in 4.2 Volume and force calculations.
    b. To simulate the setting of the packer best, it should be set by increasing the cylinder pressure linearly to the equivalent water pressure in annulus. This can be hard to implement in real life, therefore the alternative is to do the setting in steps, for example 5-10% alternating between water pressure and hydraulic pressure.
    c. Before the water pressure can be increased the packer has to seal.
2. Under the setting operation:
    a. A visual inspection of the packer is done with a camera.
    b. Packer pressure should be monitored and recorded.
    c. The water pressure and hydraulic pressure (which gives the measurement on the forces) should be monitored and recorded.
3. A self lock test may be possible. If so, reduce cylinder pressure and water pressure linearly and inspect both visual, and by monitoring the pressure to find the point when it starts to leak.
4. Increase cylinder pressure and water pressure after self lock test, up to the limit of the rig or to the packer fails to find failure point.

4.1.4.3 Estimate of test duration

The estimates of the duration of a test can be seen in Table 5. This estimates is strongly dependent on what kind of test and how it is done. This estimates do not take into account the fabrication time of new size specific parts.

Duration estimate packer test			
Test type	***Tests run on the same geometry and same OD/ID.***	***Tests run on the same OD/ID but different geometry***	***Tests run on different OD/ID and/or different geometries***
Practical work	No parts of the rig needs to be changed. The "pressure head" needs to be removed to change the packer coupon. Different rubber compounds and spring configurations can be tested.	The pressure vessel do not need to be dismantled, only "bowl" and "Pressure head" and packer coupon need to be replaced.	The pressure vessel needs to be dismantled for changing size specific parts.
Duration estimate of assembling and preparing rig first test (starting with dismantled rig)	90 min	90 min	90 min
Duration estimate of assembling and preparing rig next test (starting after previous test with assembled rig)	15-30 min	30-60 min	180 min
Duration estimate of on test run (Starting from assembled and prepared rig)	30-60 min	30-60 min	30-60 min
Duration estimate of one test run including assembling and preparation	45-90min	60-120 min	210-240 min

Table 5: Test duration estimates

4.1.5 Conceptual design review meeting

When the conceptual design phase was done, a "Conceptual design review meeting" was held in the Research and Development (R&D) department of TDW offshore Stavanger to validate the design so far, and to get suggestions and instructions for the next phase of development and master thesis program. The meeting summary can be seen in Appendix A: Conceptual design review meeting summary.

The main concern in the meeting was how to deal with the volume change and hoop stress in the rubber, and the strain in the springs that occurs in a real life situation. After a discussion round there was an agreement on that the initial simplified idea for the test rig with a linearized rectangular packer coupon with no stretching would give the results that we are looking to get out of the first prototype. It was decided to widen the test rig from 200mm inside width to 250mm inside width to make an opening for making and applying a mechanism that can address the volume change and stretching in the coupon at a later time.

4.2 Volume and force calculations

The volume and force calculations follow the equations derived in chapter 3.1 Packer calculations. The method of conservation of volume is used to calculate the packers and find the expected factors such as:

- Packer pressure against the pipe wall
- Force needed to deform the packer
- Packer deformation with given pipeline pressure

The assumptions for using the method and for applying the linearized method of conservation of volume to the packer coupon is:

- The volume of the rubber in the packer element is the same in unset and set position.
- The rubber in the packer behave as an incompressible fluid.
- Pressure to deform the packer is the same as on the whole circular packer
- Pressure between packer and pipe wall is the same as on the whole circular packer

The calculation is done on the biggest packer profile in the TDW system, the 48" packer, to find the outer limits of dimensions and forces the packer tester have to handle. The results are used in the further design of the packer tester. The input and results from the calculations is presented in Table 6 and Table 7. The maximum pipe ID is the outer diameter of the packer with half the extrusion spring radius in expansion ref Eq. 19.

$$ID_{pipe\ max} = OD_{packer} + 2 * r_{spring} \qquad \text{(Eq. 19)}$$

It is important to note that the absolute highest pressure the chamber have to withstand is limited by the maximum force the hydraulic cylinder apply. In this case a hydraulic cylinder with a maximum force of 350T is chosen, see 4.3.3 Hydraulic cylinder.

Inndata Packer			
Packer no.			**167734**
Description	**Unit**	**Notation**	**Value**
Segment length	[mm]	Lcoupon	238,5
Packer width at ID	[mm]	Wpacker	150
Outer diameter packer	[mm]	ODpacker	1080
Inner diameter packer	[mm]	IDpacker	650
Angle side faces	[deg]	β	75
Hardness	[shore A]	Shore A	65
Max id pipe	[mm]	IDpipe	1166,2
Gap between packer and pipe	[mm]	rextrusion	43,1
Radius of spring	[mm]	rspring	43,1
Pipeline pressure	[Mpa]	Ppipe	37,55241767
Inndata Cylinder			
Description	**Unit**	**Notation**	**Value**
Radius piston rod	[mm]	rpiston rod	75
Radius piston	[mm]	rcylinder	200
Hydraulic pressure	[Mpa]	Pcylinder	23,5
Inndata unset springs			
Spring no.			**123428**
Description	**Unit**	**Notation**	**Value**
Unset length	[mm]	Lspring unset1	1120
Pretension length	[mm]	Lspring pretension1	670
Set position length	[mm]	Lspring plug set1	422
Spring constant	[N/mm]	k1	7,5
Number of springs	Amount	Nspring1	12
Spring no.			**123433**
Description	**Unit**	**Notation**	**Value**
Unset length	[mm]	Lspring unset2	1120
Pretension length	[mm]	Lspring pretension2	670
Set position length	[mm]	Lspring plug set2	422
Spring constant	[N/mm]	k2	4,1
Number of springs	Amount	Nspring2	12

Table 6: Inndata for volume and force calculations

The calculation for hydraulic pretension force and for unset spring force is considered to get as real results as possible to transfer to the packer coupon. The plug module for the 48” packer have 12 springs inside 12 springs with bigger radius.

Output				
Description	**Unit**	**Notation**	**Value packer (Linearized formula)**	**Value packer coupon**
Hydraulic cylinder force	[kN]	Fpretension	2537,8	
Force from pipeline pressure	[kN]	Fpipe	38938,2	
Unset Spring force in pretension position (Spring no: 123428)	[kN]	Fspring plug unset1	40,5	
Unset Spring force in pretension position (Spring no: 123433)	[kN]	Fspring plug unset2	22,1	
Unset Spring force in set position (Spring no: 123428)	[kN]	Fspring plug set1	62,8	
Unset Spring force in set position (Spring no: 123433)	[kN]	Fspring plug set2	34,3	
Angle alfa	[deg]	α	15	
Modulus of elasticity for the rubber	[Mpa]	Mod 100	2,413	
K-factor	ratio	K	0,54	
Average diameter packer	[mm]	ADpacker	865	
Radial height packer	[mm]	Hpacker	215	
Volume packer	[mm^3]	Vpacker	124085866	
Packer width at ID at set position	[mm]	Wpacker set	92,8	103,8
Deformation of packer	[mm]	δpacker	57,2	46,2
Force to deform packer	[kN]	Fpacker	915,5	81,4
Force to deform packer	[T]	Fpacker	93,3	8,3
Pressure to deform packer	[Mpa]	Ppacker def	1,567	1,567
Total force acting on the packer due to pipeline pressure (With unset springs)	[kN]	Fpacker pipe	40463,4	
Total force acting on the packer due to pipeline pressure (Without unset springs)	[kN]	Fpacker pipe	40560,5	3433,5
Total force acting on the packer due to pipeline pressure (With unset springs)	[T]	Fpacker pipe	4124,7	
Total force acting on the packer due to pipeline pressure (Without unset springs)	[T]	Fpacker pipe	4134,6	350,0
Pressure between packer and pipe wall at isolation pressure (With unset springs)	[Mpa]	Ppacker wall	54,95	
Pressure between packer and pipe wall at isolation pressure (Without unset springs)	[Mpa]	Ppacker wall	55,08	55,08

Table 7: Output from volume and force calculations

There are some key results from Table 7 that form the bases for the calculations on the test rig. The key data is:

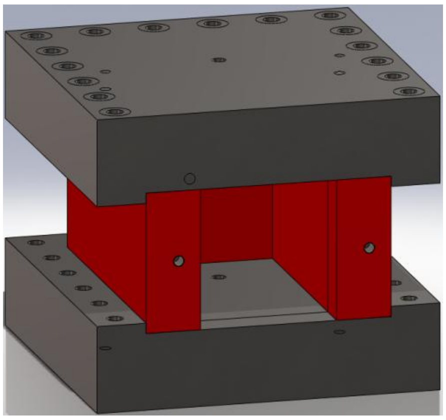
- "Pressure between packer and pipe wall at isolation pressure (Without unset springs)" at 55,08MPa. This is the maximum pressure the Pressure chamber have to be designed for. The assumption that this pressure shall be simulated to be the same in the test rig as in a real life situation, is the reason for this pressure to be the same for the coupon and for the packer. The results marked "Without unset springs" is used because there is not unsetting springs in the test rig.
- "Total force acting on the packer due to pipeline pressure (Without unset springs)" at 350T for the packer coupon. This is the actual maximum force the hydraulic cylinder in the test rig applies. Also here is the results marked "Without unset springs" used because there is not unsetting springs in the test rig as there is on the plug module. The maximum force from the hydraulic cylinder is equivalent to 37MPa pipeline pressure in a real life situation. This means that with the 350T hydraulic cylinder, it is possible to simulate a pipeline pressure of 37MPa on the 48" packer.
- The internal width of the chamber or coupon length have not been widened from 200 to 250 but from 200 to 238,5. The background for this come from the stress hand calculations on the pressure chamber, weight issue and the fabrication method due to availability of plates. More about this in 4.3.1.1 Material selection and 4.3.1.2 Hand calculations
- Another parameter that is interesting is the "Packer width at ID at set position" that on the plug module after setting is 92,8mm and in the test rig is 103,8mm, with the same pipeline pressure. This is a reasonable result due to the fact that the packer coupon in the test rig is not stretched in longitudinal direction as the circular packer will be in a real life situation.

4.3 Analysis

The analysis part of the packer tester design phase is started with a basic hand calculation to use as a starting position for the SolidWorks simulation that follows. The SolidWorks simulation is an iterative process that starts where the hand calculations stops and then by several simulations are run and corrected to optimize the design. The Packer test rig is designed according to the standard NS-EN 13445-3:2009 "Unfired pressure vessels".

4.3.1 Pressure vessel

The pressure vessel module of the test rig, see Picture 35, is the chamber the packer coupon is pressurized in. The vessel have to withstand high pressure, have to be dismountable for size flexibility and have to be easy for personnel to handle and work with.



Picture 35: Pressure vessel

4.3.1.1 Material selection

The material selected for the pressure chamber is selected with background in:

- The high pressure that leads to high wall thickness
- Price
- Availability at steel suppliers
- Machining properties

The material with a high yield and tensile strength, that meet the requirements best, is S690QL see Appendix E: S690QL . The S690QL is a high yield structural steel grade produced in compliance with EN 10025:6:2004. The material is heat treated using the quench and temper process and has good bending and welding properties. TDW's main supplier "Maskinering & Sveiseservice" has plates in S690QL with maximum thickness of 160mm in stock, and provides plates in S690QL with thickness up to 200mm on order. Due to weight and delivery time reasons it is decided to design the packer tester with 160mm thick plates.

The material meets the requirements in NS-EN 13445-2:2009 for composition, minimum elongation of 14% after fracture and impact energy measured on a Charpy-V-notch impact test. See Appendix C: Extracts from NS-EN13445:2009

According to NS-EN13445-3:2009 Table 8 shows the allowable design stress level, membrane stress and sum of membrane and bending stress. The material input data is according to Appendix E: S690QL

Material inndata				
Description	**Notation**	**Unit**	**Value**	**Note**
Yield strength	σy	N/mm^2	630	
Tensile strength	σu	N/mm^2	900	
Allowable stress and safety level				
Description	**Notation**	**Unit**	**Value**	**Note**
z-factor	z		1	1 if no longitudinal or circumferential weld
Allowable design stress level	f	N/mm^2	375,0	NS-EN 13445-3:2009 chapter 6.2
Allowable membrane stress	f * z	N/mm^2	375,0	NS-EN 13445-3:2009 eq: 15.5.3-1
Allowable sum of membran and bending stress	1,5 * f * z	N/mm^2	562,5	NS-EN 13445-3:2009 eq: 15.5.3-2

Table 8: Allowable design stress calculation for pressure vessel

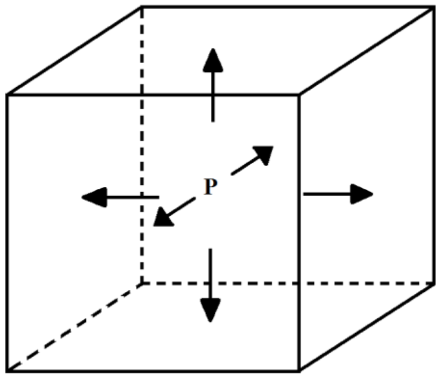
4.3.1.2 Hand calculations

The hand calculations are done to have a reference to compare the simulation results with

The calculations on the pressure vessel are done according to the standard "NS-EN 13445-3:2009 Issue 1". See Appendix C: Extracts from NS-EN13445:2009 for the relevant parts from the standard. The verification for using the methods in the standard to this specific case can be seen in Appendix D: Letter from CEN regarding NS-EN 13445-3:2009.

Assumptions for making the calculations according to NS-EN 13445-3:2009 is:

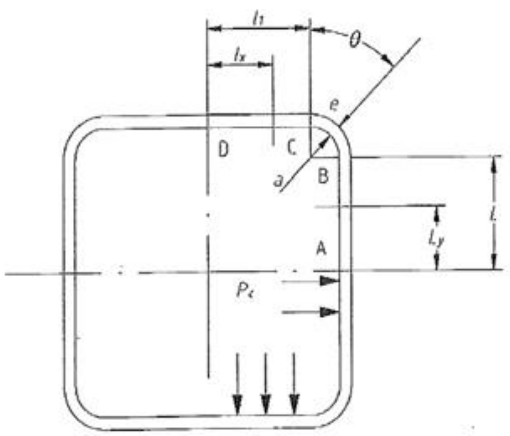
- A rectangular pressure vessel simplified as shown in pic ref



Picture 36: Rectangular assumption for hand calculation on pressure vessel

- No corner radius
- No longitudinal or circumferential welds
- No fatigue loads
- Pressure vessel is located in area with:
    - pressure 1atm
    - temperature 20°C

The results from the hand calculations seen in Table 9, correspond with the nomenclature in Picture 37.



Picture 37: Nomenclature on pressure vessel hand calculation

Inndata pressure vessel walls				
Description	**Notation**	**Unit**	**Value**	**Note**
Internal height of chamber	L	mm	258,1	
Internal width chamber	l1	mm	238,5	
Plate thickness	t	mm	160	
Yield strength	σy	N/mm^2	630	
Tensile strength	σu	N/mm^2	900	
Desired packer pressure	Ppacker wall	N/mm^2	55,08	
Second moment of inertia	I1,I2	mm^4/mm	341333,33	NS-EN 13445-3:2009 eq: 15.5.1.2-4
Inside radius of corner	a	mm	0,00	
Output pressure vessel walls according NS-EN 13445-3:2009				
Description	**Notation**	**Unit**	**Value**	**Note**
Alfa factor 1	α1		0,95	NS-EN 13445-3:2009 eq: 15.5.1.2-13
Alfa factor 3	α3		0,92	NS-EN 13445-3:2009 eq: 15.5.1.2-14
Phi factor	φ		0,00	NS-EN 13445-3:2009 eq: 15.5.1.2-15
K factor	K3	mm^3/mm	39936,49	NS-EN 13445-3:2009 eq: 15.5.1.2-12
Bending moment A	MA	Nmm/mm	-2199701,78	NS-EN 13445-3:2009 eq: 15.5.1.2-11
Membrane stress point C	σmC	Mpa (N/mm^2)	44,43	NS-EN 13445-3:2009 eq: 15.5.1.2-1
Membrane stress point D	σmD	Mpa (N/mm^2)	44,43	NS-EN 13445-3:2009 eq: 15.5.1.2-2
Membrane stress point B	σmB	Mpa (N/mm^2)	41,05	NS-EN 13445-3:2009 eq: 15.5.1.2-2
Membrane stress point A	σmA	Mpa (N/mm^2)	41,05	NS-EN 13445-3:2009 eq: 15.5.1.2-3
Bending stress point C	σbC	Mpa	-408,06	NS-EN 13445-3:2009 eq: 15.5.1.2-5
Bending stress point D	σbD	Mpa	-499,85	NS-EN 13445-3:2009 eq: 15.5.1.2-6
Bending stress point A	σbA	Mpa	-515,56	NS-EN 13445-3:2009 eq: 15.5.1.2-7
Bending stress point B	σbB	Mpa	-408,06	NS-EN 13445-3:2009 eq: 15.5.1.2-8
Allowable stress and safety level for unreinforced vessels				
Description	**Notation**	**Unit**	**Value**	**Note**
z-factor	z		1	1 if no longitudinal or circumferential weld
Allowable design stress level	f	N/mm^2	375,0	NS-EN 13445-3:2009 chapter 6.2
Allowable membrane stress	f * z	N/mm^2	375,0	NS-EN 13445-3:2009 eq: 15.5.3-1
Allowable sum of membran and bending stress	1,5 * f * z	N/mm^2	562,5	NS-EN 13445-3:2009 eq: 15.5.3-2
Maximum membrane stress	σmmax	N/mm^2	44,43	
Maximum sum of membrane and bending stress	σbmax	N/mm^2	559,98	
Safety level: Membrane stress			8,44	(σm ≤ f * z) => OK!
Safety level: Sum of membrane and bending stress			1,00	(σm + σb ≤ 1,5 * f * z) => OK!

Table 9: Results from hand calculations on the pressure vessel, according to NS-EN 13445-3:2009

The key results from the hand calculations on the pressure vessel are:

- The biggest possible internal width with wall thickness 160mm and internal pressure 55,08MPa is 238,5mm.

4.3.1.3 SolidWorks simulations

In this chapter the process of the SolidWorks simulation on the pressure chamber in the packer test rig will be described. In the iteration process several simulations have been run, however it is only the final design simulation that is described in this chapter. The main results from the iteration process is:

- Internal width is 250mm; due to the conservative methods of hand calculation the internal width of the chamber initially was 238,5, the simulation shows that the internal width of the chamber could be widened to the desired width. This again leads to a recalculation of the packer coupon forces. See 4.3.1.3.1 Recalculation of packer coupon forces.
- The wall thickness of the U-chamber is 100mm; the fixture mode of this part allow the wall thickness to be lowered as low as 100mm.
- The wall thickness in the top and bottom plate is 141mm; this measurement is not 160mm as the hand calculation initiated because of the fabrication method. These parts is machined out of a 160mm plate.

4.3.1.3.1 Recalculation of packer coupon forces

Due to the revisions of pressure chamber width, a new hand calculation of the packer coupon forces had to be run. The innput data to the calculations can be seen in Table 10.

Inndata Packer			
Packer no.			**167734**
Description	**Unit**	**Notation**	**Value**
Segment length	[mm]	Lcoupon	250
Packer width at ID	[mm]	Wpacker	150
Outer diameter packer	[mm]	ODpacker	1080
Inner diameter packer	[mm]	IDpacker	650
Angle side faces	[deg]	β	75
Hardness	[shore A]	Shore A	65
Max id pipe	[mm]	IDpipe	1166,2
Gap between packer and pipe	[mm]	rextrusion	43,1
Radius of spring	[mm]	rspring	43,1
Pipeline pressure	[Mpa]	Ppipe	35,75396468
Inndata Cylinder			
Description	**Unit**	**Notation**	**Value**
Radius piston rod	[mm]	rpiston rod	75
Radius piston	[mm]	rcylinder	200
Hydraulic pressure	[Mpa]	Pcylinder	23,5
Inndata unset springs			
Spring no.			**123428**
Description	**Unit**	**Notation**	**Value**
Unset length	[mm]	Lspring unset1	1120
Pretension length	[mm]	Lspring pretension1	670
Set position length	[mm]	Lspring plug set1	422
Spring constant	[N/mm]	k1	7,5
Number of springs	Amount	Nspring1	12
Spring no.			**123433**
Description	**Unit**	**Notation**	**Value**
Unset length	[mm]	Lspring unset2	1120
Pretension length	[mm]	Lspring pretension2	670
Set position length	[mm]	Lspring plug set2	422
Spring constant	[N/mm]	k2	4,1
Number of springs	Amount	Nspring2	12

Table 10: Inndata to packer coupon forces recalculations

Output				
Description	**Unit**	**Notation**	**Value packer (Linearized formula)**	**Value packer coupon**
Hydraulic cylinder force	[kN]	Fpretension	2537,8	
Force from pipeline pressure	[kN]	Fpipe	37559,1	
Unset Spring force in pretension position (Spring no: 123428)	[kN]	Fspring plug unset1	40,5	
Unset Spring force in pretension position (Spring no: 123433)	[kN]	Fspring plug unset2	22,1	
Unset Spring force in set position (Spring no: 123428)	[kN]	Fspring plug set1	62,8	
Unset Spring force in set position (Spring no: 123433)	[kN]	Fspring plug set2	34,3	
Angle alfa	[deg]	α	15	
Modulus of elasticity for the rubber	[Mpa]	Mod 100	2,413	
K-factor	ratio	K	0,54	
Average diameter packer	[mm]	ADpacker	865	
Radial height packer	[mm]	Hpacker	215	
Volume packer	[mm^3]	Vpacker	124085866	
Packer width at ID at set position	[mm]	Wpacker set	92,8	103,8
Deformation of packer	[mm]	δpacker	57,2	46,2
Force to deform packer	[kN]	Fpacker	915,5	84,2
Force to deform packer	[T]	Fpacker	93,3	8,6
Pressure to deform packer	[Mpa]	Ppacker def	1,567	1,567
Total force acting on the packer due to pipeline pressure (With unset springs)	[kN]	Fpacker pipe	39084,3	
Total force acting on the packer due to pipeline pressure (Without unset springs)	[kN]	Fpacker pipe	39181,5	3433,5
Total force acting on the packer due to pipeline pressure (With unset springs)	[T]	Fpacker pipe	3984,1	
Total force acting on the packer due to pipeline pressure (Without unset springs)	[T]	Fpacker pipe	3994,0	350,0
Pressure between packer and pipe wall at isolation pressure (With unset springs)	[Mpa]	Ppacker wall	53,08	
Pressure between packer and pipe wall at isolation pressure (Without unset springs)	[Mpa]	Ppacker wall	53,21	53,21

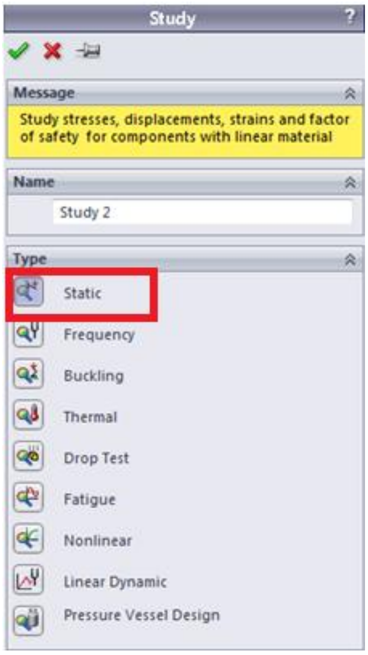
Table 11: Output from the packer coupon forces recalculations

The results from the packer coupon force calculation rerun can be seen in Table 11. The key results from the calculation which are the input parameters for the following simulation is:

- Maximum force acting on the packer is 350T which is limited by the hydraulic cylinder
- Maximum packer coupon pressure is 53,21MPa
- The simulated pipeline pressure equivalent to the packer coupon pressure is 35,75MPa

4.3.1.3.2 Linear static study set up

1. Linear static analysis, see Picture 38, is used in the simulation of the pressure chamber, see 3.2.4 Linear static analysis for an explanation of the method and the assumption.

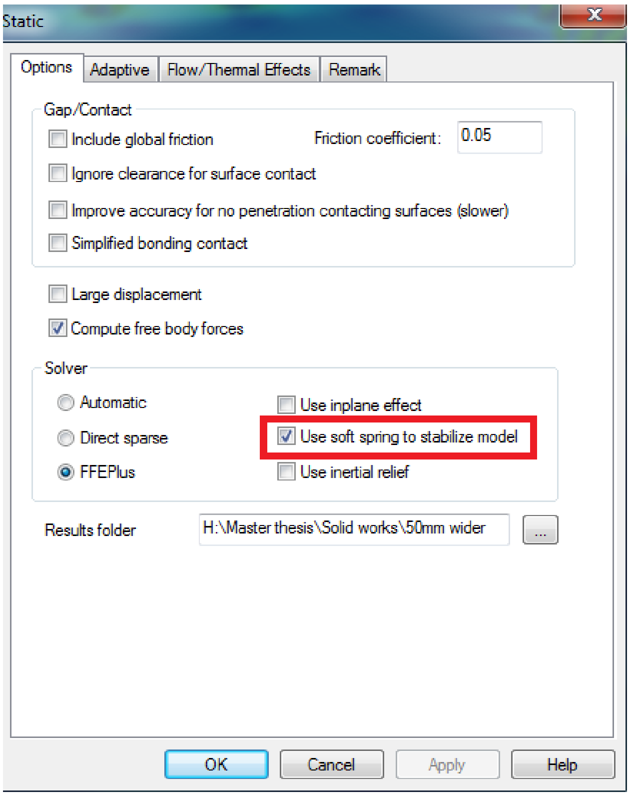


Picture 38: Linear static study

2. For the contact between the components in the assembly it is assumed no penetration see Picture 39.

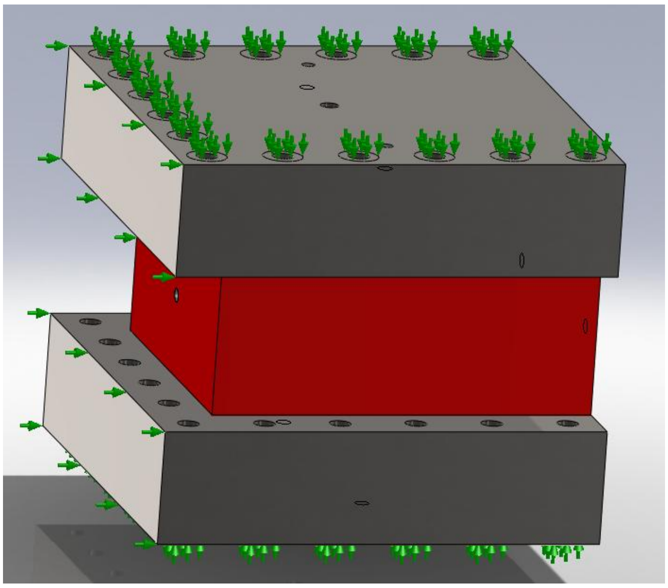
Picture 39: Contact between components

3. The fixtures of the pressure chamber is:
   a. The pressure chamber rest against the bottom plate, this is simulated by using soft springs to stabilize the model see Picture 40.



Picture 40: Stabilzation of the model

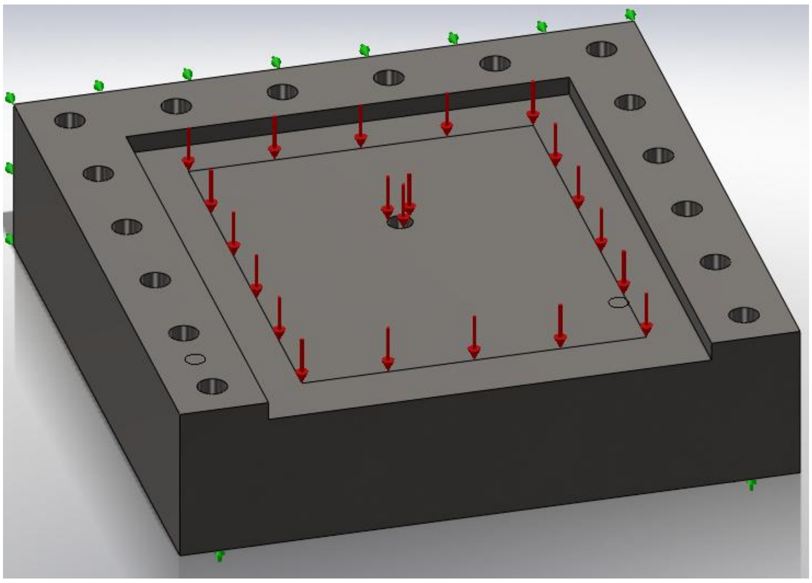
   b. The chamber sits against the back wall in the support frame and have pressure from the hydraulic cylinder from the opening or front of the chamber. This is simulated by supporting it on the back end of top and bottom plate, see Picture 41.



Picture 41: Fixtures of pressure chamber in simulation

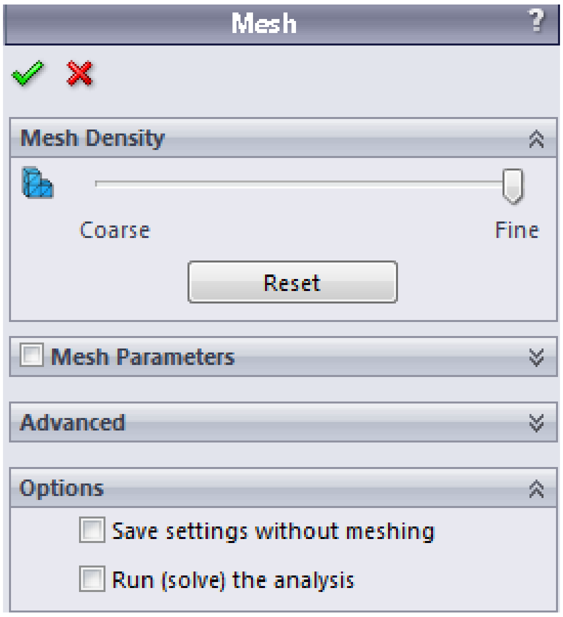
c. With background the calculations seen in 4.3.7 Bolt calculation, it is assumed that the main bolts can hold the pressure chamber parts perfectly together with no deformation of the bolts, the bolts rests on the washers where the top and bottom plate is fixed, see Picture 41.

4. The external load applied to the chamber is 53,21MPa internal pressure. The internal pressure is limited within the seal groves along the top of the U-chamber and is also limited a distance corresponding to the smallest thickness of the bowl/piston inside the edge of the opening, see Picture 42. The reason for doing this, is that pressure never will appear outside this area.

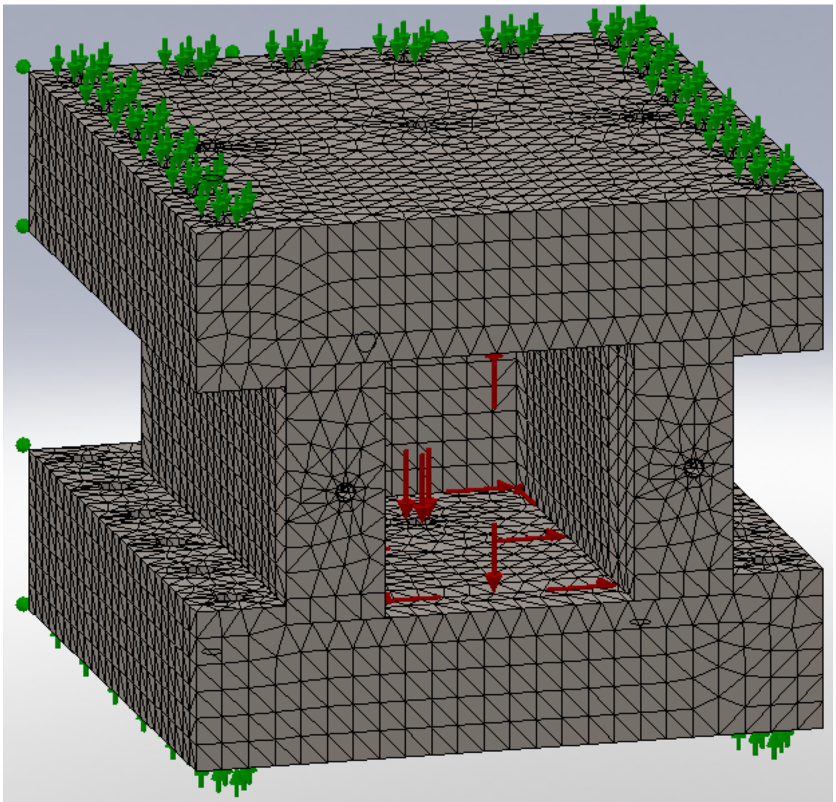


Picture 42: External loads to pressure chamber simulation

5. The chamber model is meshed with fine density, see Picture 43 and Picture 44.



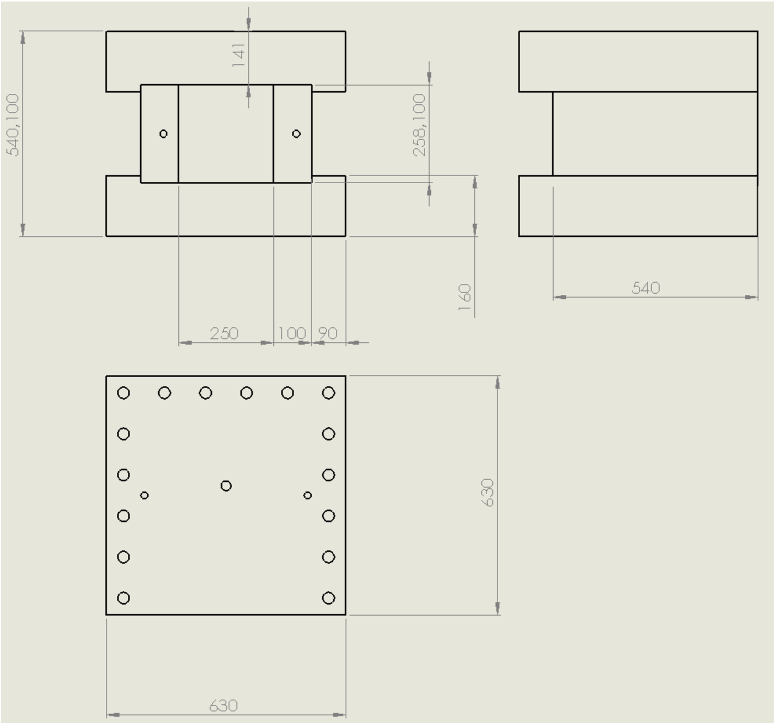
Picture 43: Model mesh



Picture 44: Pressure chamber mesh

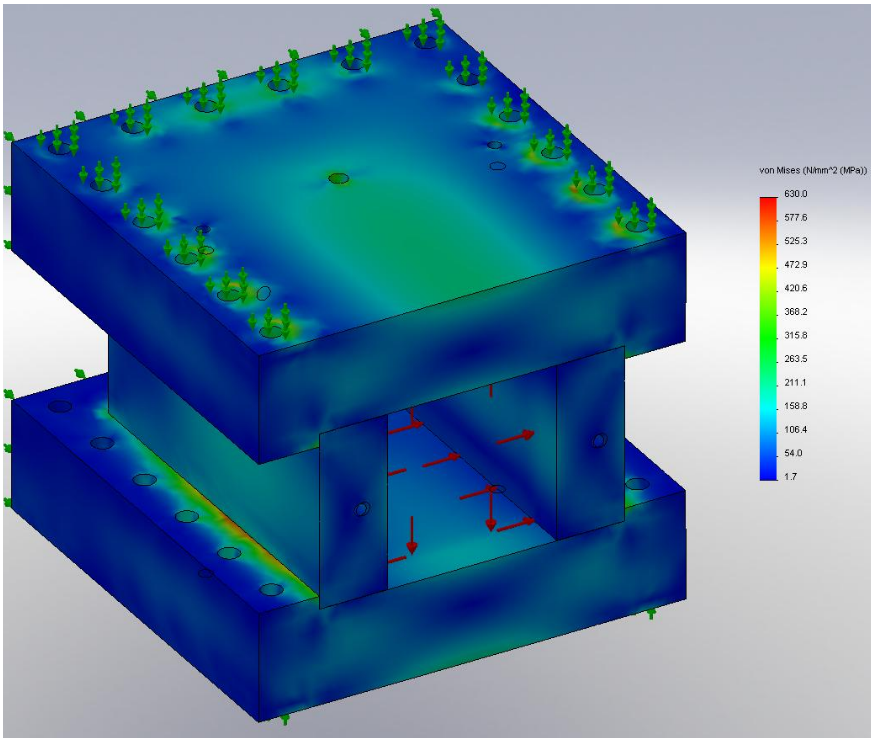
4.3.1.3.3 Simulation results design forces

The final measurements on the pressure chamber is according to Picture 45.

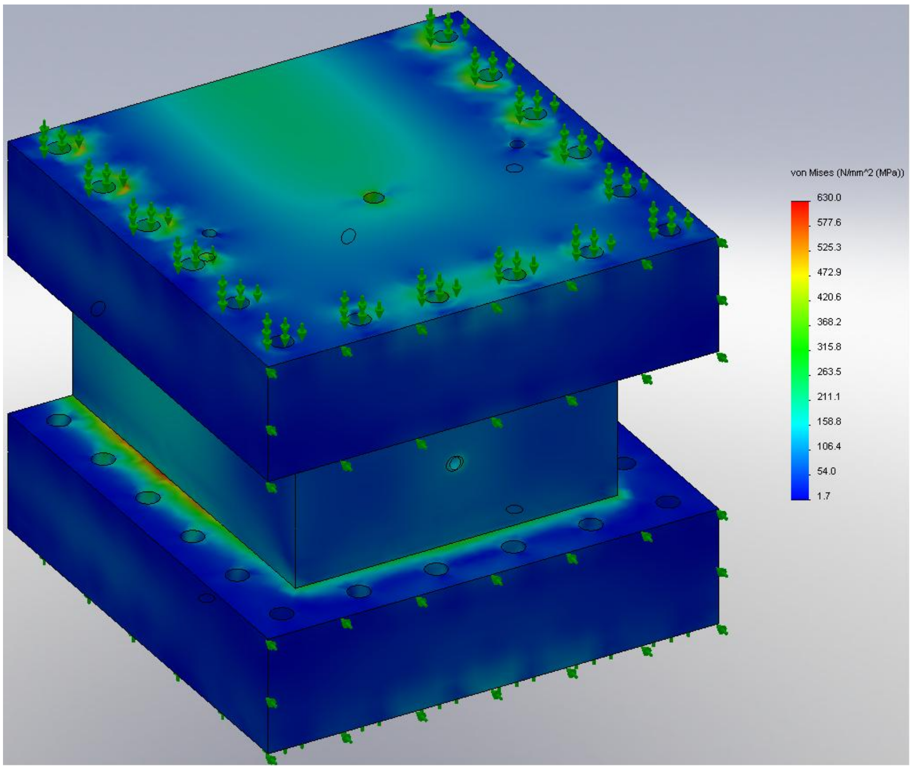


Picture 45: Measurements on pressure chamber

The overview of the simulation results, see Picture 46 and Picture 47, shows why the large wall thickness is needed. It is the bending moments that sets the limitations on this design. Some peak stresses can also be seen along sharp edges and around the washers on the main bolts.

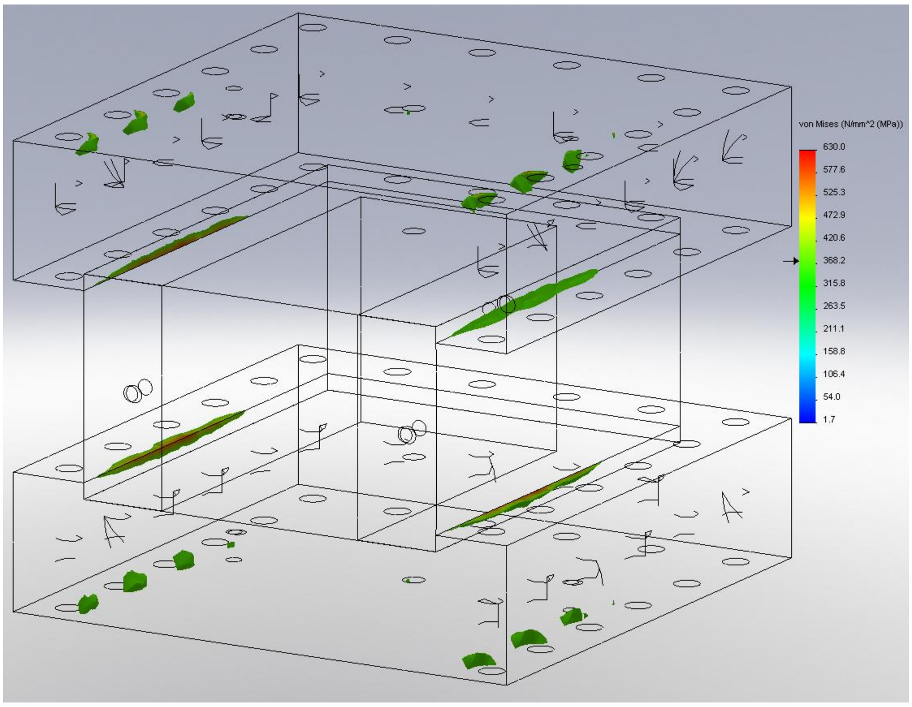


Picture 46: Overview of simulation results pressure chamber



Picture 47: Overview of simulation results pressure chamber

By using ISO-clipping, see Picture 48, the peak points that exceeds the allowable stress limits easily can be shown. The peak points that occurs on the inner edge of the washers, comes from the assumption that the top and bottom is perfectly fixed with no deflection on these points, which in reality is not true, due to some elongation in the bolts. The reason for this to occur on the inner part of the washer is the direction of the bending in the plates.

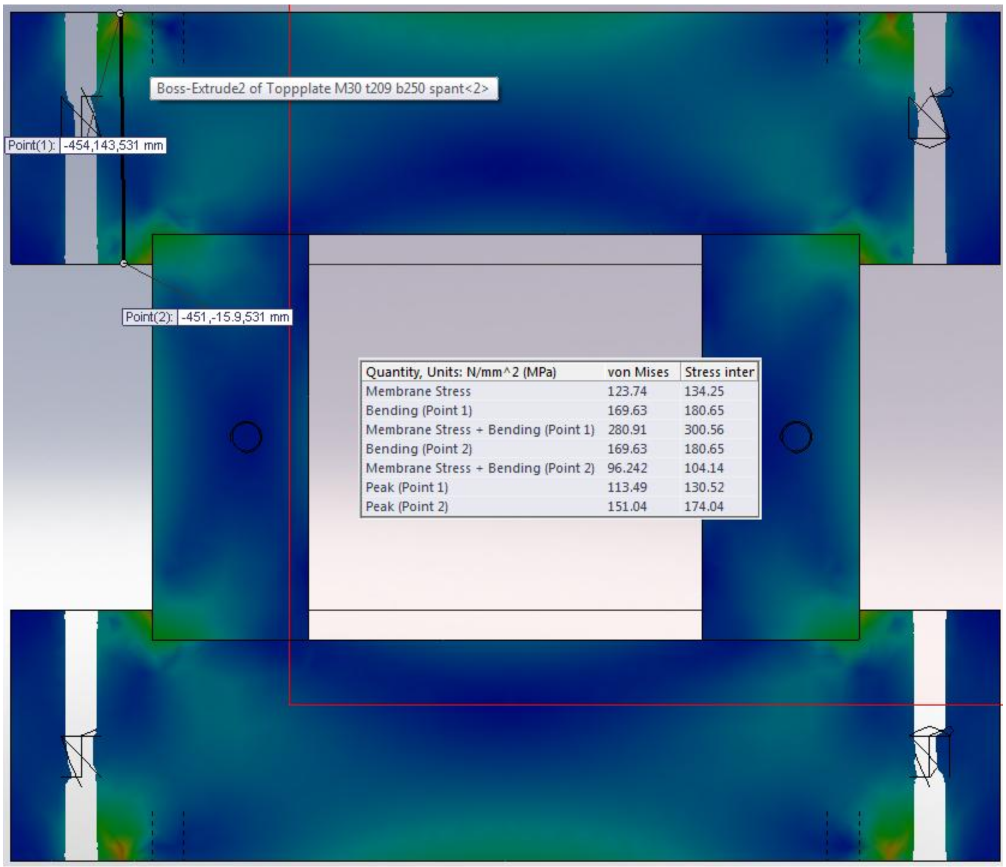


Picture 48: Peak points on pressure chamber that exceeds allowable stress limit

The peak points along the edges where the U-chamber sits against the top and bottom plates comes from the bending in both the top and bottom plate and the U-chamber. There are none points on the pressure chamber that exceeds the yield limit of 630MPa, the highest peak stress is 607MPa.

To verify the peak points according to membrane stress and membrane + bending stress a linearization through the wall thickness for the relevant peak points is done.

Point 1:



Picture 49: Linearization on pressure chamber point 1

None of the linearized von Mises stresses, see Picture 49, is above the allowable stresses calculated according to NS-EN13445-3:2009, see Table 8:

- Membrane stress: 123,74MPa < 375MPa→OK
- Bending stress point 1: 169,63MPa <375MPa→OK
- Membrane + bending stress point 1: 280,91MPa < 562,5MPa→OK
- Bending stress point 2: 169,63MPa <375MPa→OK
- Membrane + bending stress point 2: 96,24MPa < 562,5MPa→OK

Utilization at the relevant points:

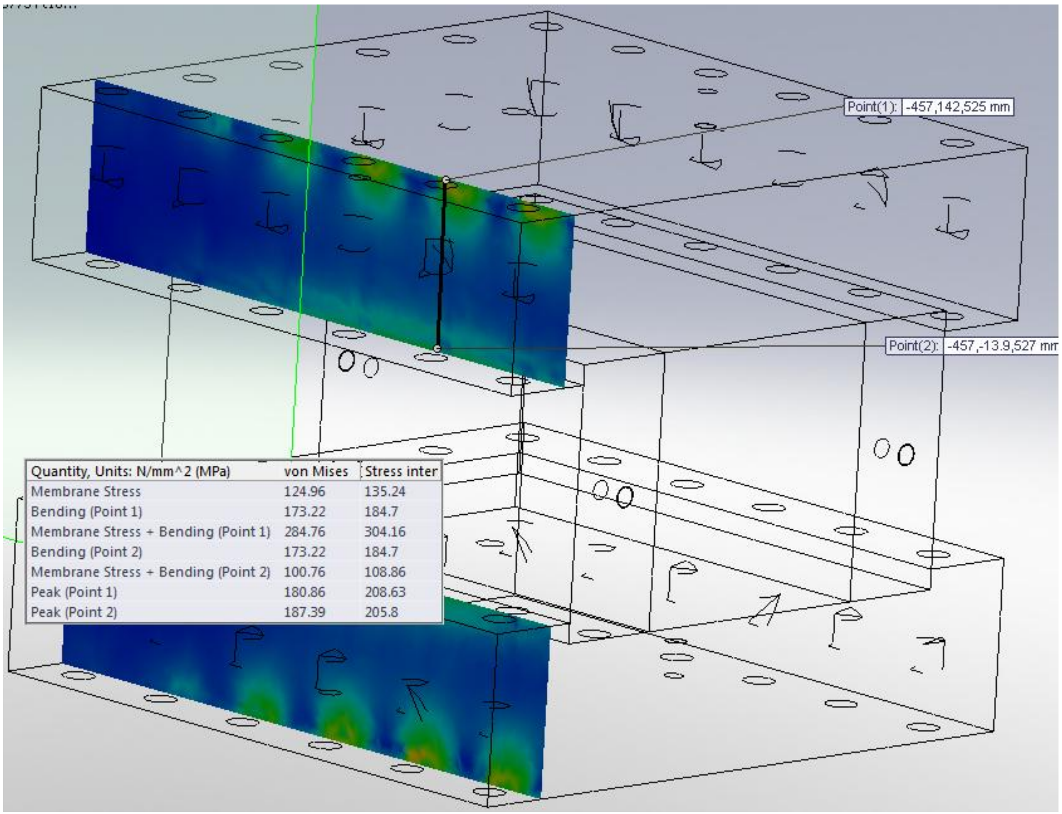
- Membrane stress:

$$U_{M\,C1} = \frac{P_{M\,C1}}{P_{M\,max}} = \frac{123{,}74}{375} = 0{,}33 \qquad \text{(Eq. 20)}$$

- Membrane + bending stress:

$$U_{MB\ C1} = \frac{P_{MB\ C1}}{P_{MB\ max}} = \frac{280{,}91}{562{,}5} = 0{,}50 \qquad \text{(Eq. 21)}$$

Point 2:



Picture 50: Linearization on pressure chamber point 2

None of the linearized von Mises stresses, see Picture 50, is above the allowable stresses calculated according to NS-EN13445-3:2009, see Table 8:

- Membrane stress: 124,96MPa < 375MPa→OK
- Bending stress point 1: 173,22MPa <375MPa→OK
- Membrane + bending stress point 1: 284,76MPa < 562,5MPa→OK
- Bending stress point 2: 173,22MPa <375MPa→OK
- Membrane + bending stress point 2: 100,76MPa < 562,5MPa→OK

Utilization at the relevant points:

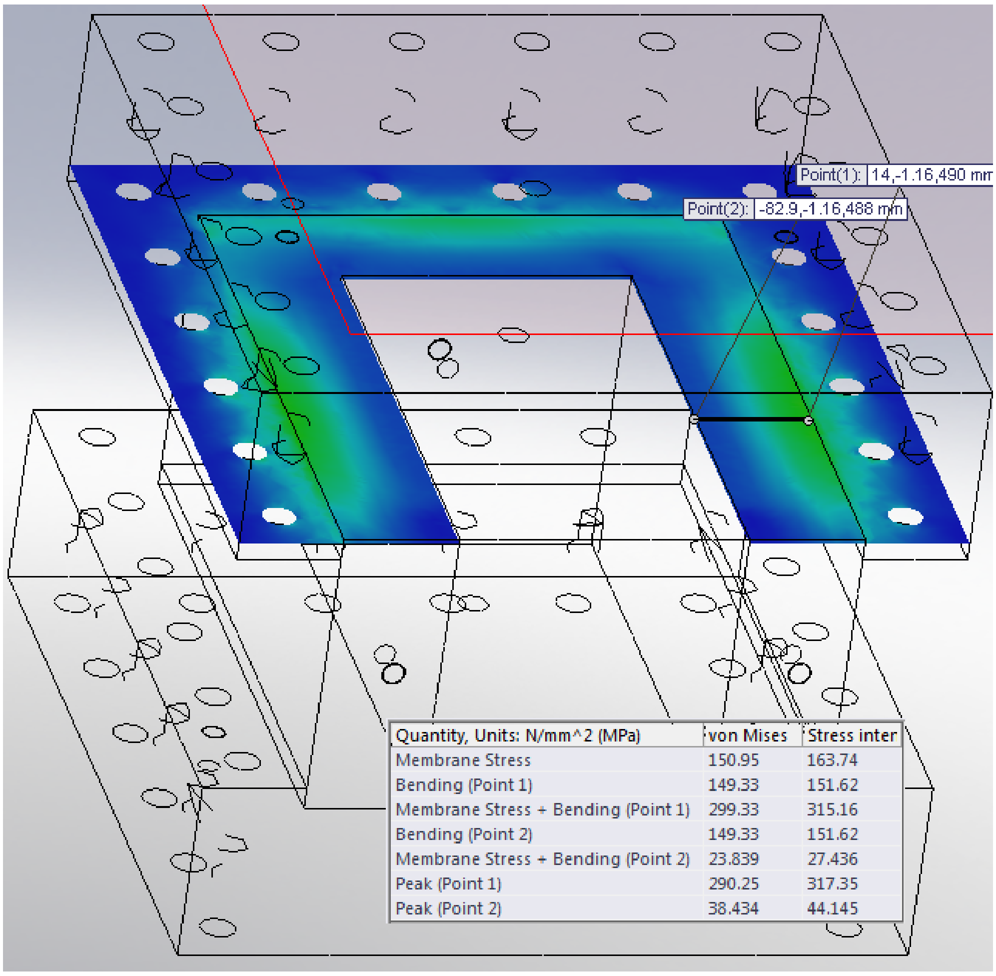
- Membrane stress:

$$U_{M\ C2} = \frac{P_{M\ C1}}{P_{M\ max}} = \frac{124{,}96}{375} = 0{,}33 \qquad \text{(Eq. 22)}$$

- Membrane + bending stress:

$$U_{MB\,C2} = \frac{P_{MB\,C1}}{P_{MB\,max}} = \frac{284{,}76}{562{,}5} = 0{,}51 \qquad \text{(Eq. 23)}$$

Point 3:



Picture 51: Linearization on pressure chamber point 3

None of the linearized von Mises stresses, see Picture 51, is above the allowable stresses calculated according to NS-EN13445-3:2009, see Table 8:

- Membrane stress: 150,95MPa < 375MPa→OK
- Bending stress point 1: 149,33MPa <375MPa→OK
- Membrane + bending stress point 1: 299,33MPa < 562,5MPa→OK
- Bending stress point 2: 149,33MPa <375MPa→OK
- Membrane + bending stress point 2: 23,839MPa < 562,5MPa→OK

Utilization at the relevant points:

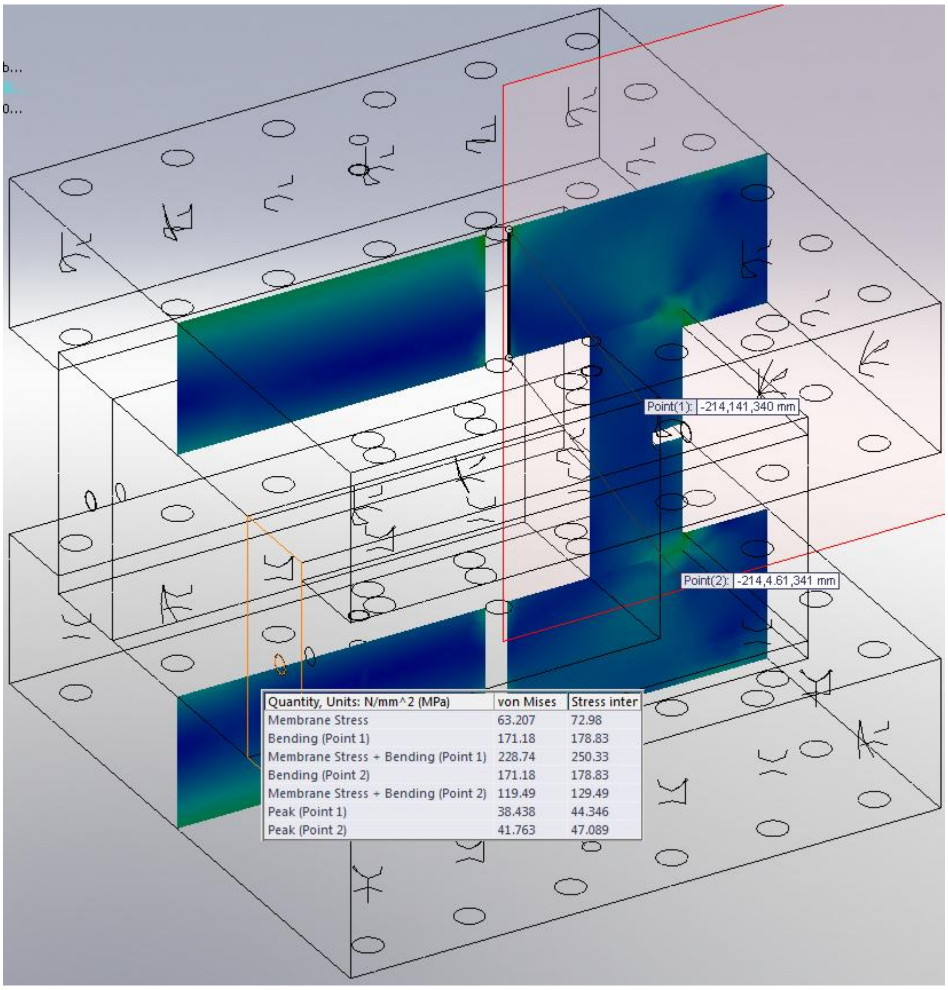
- Membrane stress:

$$U_{M\,C3} = \frac{P_{M\,C1}}{P_{M\,max}} = \frac{150{,}95}{375} = 0{,}40 \qquad \text{(Eq. 24)}$$

- Membrane + bending stress:

$$U_{MB\ C3} = \frac{P_{MB\ C1}}{P_{MB\ max}} = \frac{299{,}33}{562{,}5} = 0{,}53 \qquad \text{(Eq. 25)}$$

Point 4:



Quantity, Units: N/mm^2 (MPa)	von Mises	Stress inter
Membrane Stress	63.207	72.98
Bending (Point 1)	171.18	178.83
Membrane Stress + Bending (Point 1)	228.74	250.33
Bending (Point 2)	171.18	178.83
Membrane Stress + Bending (Point 2)	119.49	129.49
Peak (Point 1)	38.438	44.346
Peak (Point 2)	41.763	47.089

Picture 52: Linearization on pressure chamber point 4

None of the linearized von Mises stresses, see Picture 52, is above the allowable stresses calculated according to NS-EN13445-3:2009, see Table 8:

- Membrane stress: 63,21MPa < 375MPa→OK
- Bending stress point 1: 171,18MPa <375MPa→OK
- Membrane + bending stress point 1: 228,74MPa < 562,5MPa→OK
- Bending stress point 2: 171,18MPa <375MPa→OK
- Membrane + bending stress point 2: 119,49MPa < 562,5MPa→OK

Utilization at the relevant points:

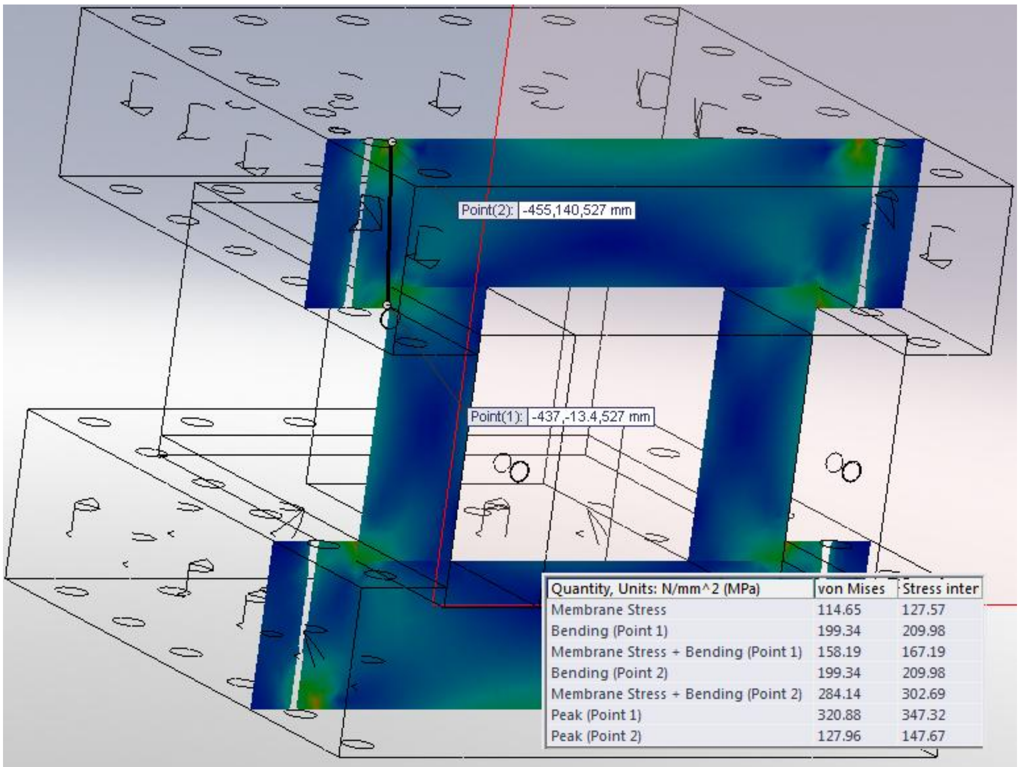
- Membrane stress:

$$U_{M\ C4} = \frac{P_{M\ C1}}{P_{M\ max}} = \frac{63{,}21}{375} = 0{,}17 \qquad \text{(Eq. 26)}$$

- Membrane + bending stress:

$$U_{MB\ C4} = \frac{P_{MB\ C1}}{P_{MB\ max}} = \frac{228,74}{562,5} = 0,41 \qquad \text{(Eq. 27)}$$

Point 5:



Picture 53: Linearization on pressure chamber point 5

None of the linearized von Mises stresses, see Picture 53, is above the allowable stresses calculated according to NS-EN13445-3:2009, see Table 8:

- Membrane stress: 114,65MPa < 375MPa→OK
- Bending stress point 1: 199,34MPa <375MPa→OK
- Membrane + bending stress point 1: 158,19MPa < 562,5MPa→OK
- Bending stress point 2: 199,34MPa <375MPa→OK
- Membrane + bending stress point 2: 284,14MPa < 562,5MPa→OK

Utilization at the relevant points:

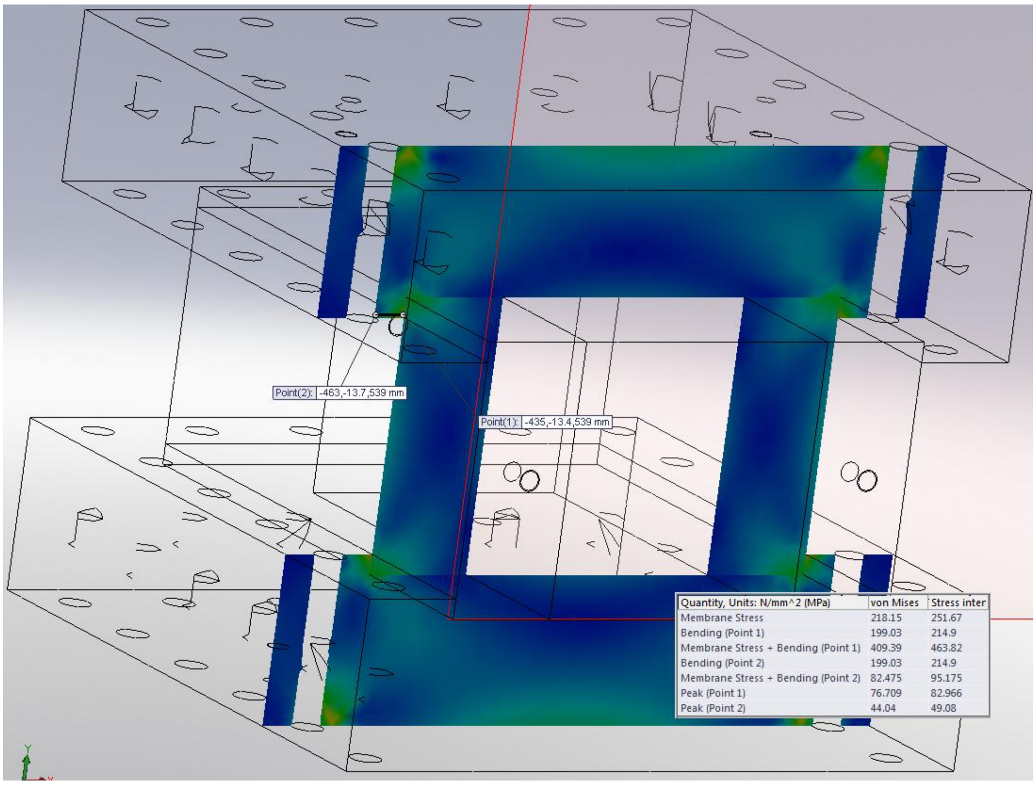
- Membrane stress:

$$U_{M\ C5} = \frac{P_{M\ C1}}{P_{M\ max}} = \frac{114,65}{375} = 0,31 \qquad \text{(Eq. 28)}$$

- Membrane + bending stress:

$$U_{MB\ C5} = \frac{P_{MB\ C1}}{P_{MB\ max}} = \frac{284,14}{562,5} = 0,51 \qquad \text{(Eq. 29)}$$

Point 6:



Picture 54: Linearization on pressure chamber point 6

None of the linearized von Mises stresses, see Picture 54, is above the allowable stresses calculated according to NS-EN13445-3:2009, see Table 8:

- Membrane stress: 218,15MPa < 375MPa→OK
- Bending stress point 1: 199,03MPa <375MPa→OK
- Membrane + bending stress point 1: 409,39MPa < 562,5MPa→OK
- Bending stress point 2: 199,03MPa <375MPa→OK
- Membrane + bending stress point 2: 82,48MPa < 562,5MPa→OK

Utilization at the relevant points:

- Membrane stress:

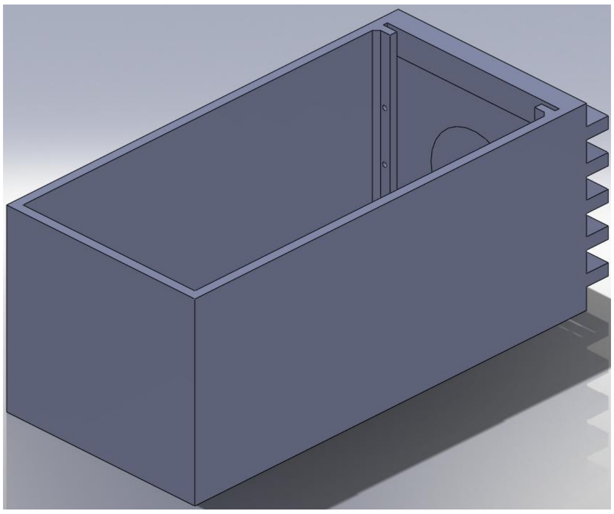
$$U_{M\ C6} = \frac{P_{M\ C1}}{P_{M\ max}} = \frac{218{,}15}{375} = 0{,}58 \qquad \text{(Eq. 30)}$$

- Membrane + bending stress:

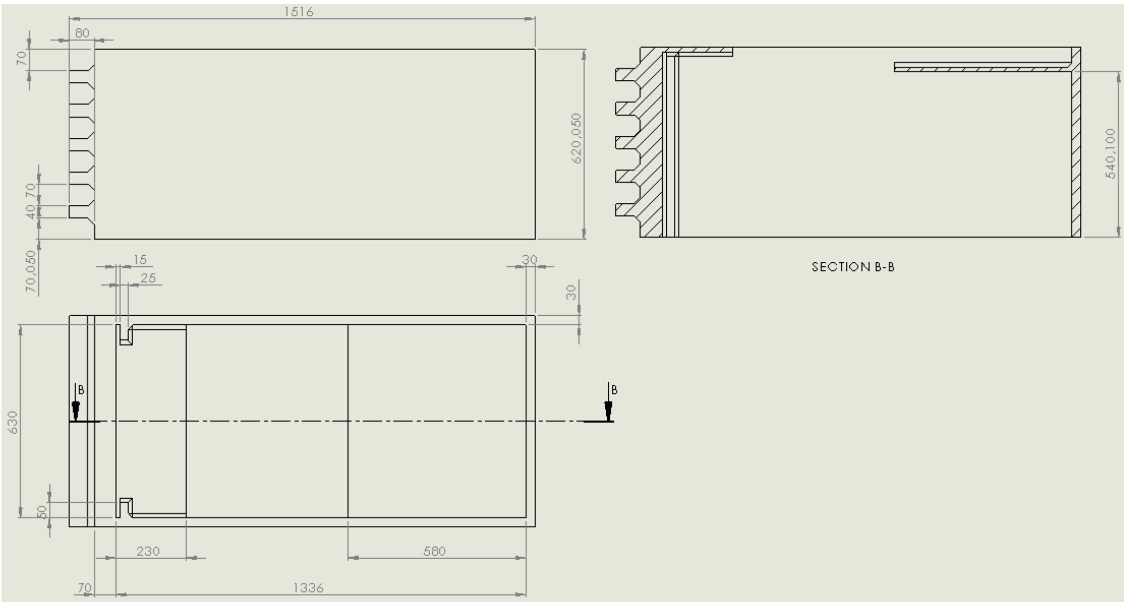
$$U_{MB\ C6} = \frac{P_{MB\ C1}}{P_{MB\ max}} = \frac{409{,}39}{562{,}5} = 0{,}73 \qquad \text{(Eq. 31)}$$

4.3.2 Support frame

The support frame, see Picture 55, supports the pressure chamber and the hydraulic cylinder in the test rig. The frame is the same for all size configurations of the rig see Table 3. The main measurements on the support frame is shown in Picture 56, for all the measurements see Appendix H: SolidWorks drawings for all measurements on the support frame.



Picture 55: Support frame



Picture 56: Measurements on support frame

4.3.2.1 Material selection

The material selected for the support frame is selected with background in:

- The high forces the support frame have to hold.
- Price
- Availability at steel suppliers
- Machining properties
- Weldability
- Pressure containing part

The material with a high yield and tensile strength, that meets the requirements best, and fulfills the requirements in the NS-EN 13445, is the same material as in the pressure chamber. The S690QL, see Appendix E: S690QL is a high yield structural steel grade produced in compliance with EN 10025:6:2004. The material is heat treated using the quench and temper process and has good bending and welding properties.

The material meets the requirements in NS-EN 13445-2:2009 for composition, minimum elongation of 14% after fracture and impact energy measured on a Charpy-V-notch impact test. See Appendix C: Extracts from NS-EN13445:2009

The allowable design stress levels is calculated according to NS-EN 13445-3:2009, see Table 12. Where the input data come from Appendix E: S690QL . Note the higher yield and tensile strength used in this calculation compared to the pressure vessel calculation, the reason is the plate thickness.

Material inndata				
Description	**Notation**	**Unit**	**Value**	**Note**
Yield strength	σy	N/mm^2	650	
Tensile strength	σu	N/mm^2	930	
Allowable stress and safety level				
Description	**Notation**	**Unit**	**Value**	**Note**
z-factor	z		1	1 if no longitudinal or circumferential weld
Allowable design stress level	f	N/mm^2	387,5	NS-EN 13445-3:2009 chapter 6.2
Allowable membrane stress	f * z	N/mm^2	387,5	NS-EN 13445-3:2009 eq: 15.5.3-1
Allowable sum of membran and bending stress	1,5 * f * z	N/mm^2	581,3	NS-EN 13445-3:2009 eq: 15.5.3-2

Table 12: Allowable design stress calculation support frame

4.3.2.2 Hand calculations and fabrication method

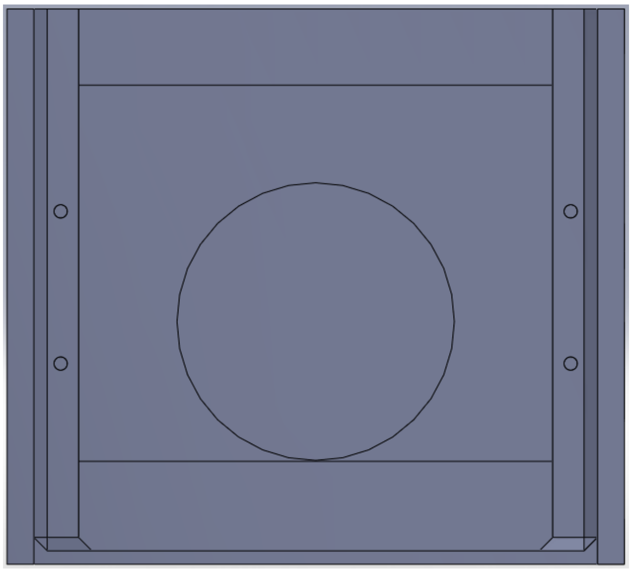
The support frame is going to be welded together with several different welding joints connecting metal plates of different wall thicknesses. In section 4.3.2.2 Hand calculations and fabrication method, everything marked with red shows a weld.

Main walls:

To find the minimum wall thickness of the main walls it is assumed that only membrane stresses and shear stresses occur.

In the right end wall , see Picture 58, where the pressure vessel sits against, this assumption is easy to justify. The reason is that the top and bottom plate in the pressure vessel that sits against this end plate are 100% stiff, which again only will apply membrane stress in the longer side walls, and shear stress in the corners.

In the left end wall, see Picture 57, the hydraulic cylinder sits against, the hydraulic cylinder is 100% stiff but is not as wide as the end wall itself which again will lead to the appearance of bending moments in the end and side walls. To avoid the bending moments the end plate have to be stiffened up. This is done by a thicker end plate with ribs welded on.



Picture 57: End wall of support frame where cylinder sits against

Load:

The maximum load on the support frame comes from the hydraulic cylinder of 350T. No Load factors are used in the calculations, since the nature of the cylinder gives the absolute limit. Further it is assumed that the whole wall thickness can take up the stresses.

$$F = 350000kg * 9{,}81\frac{m}{s^2} = 3433500N \qquad \text{(Eq. 32)}$$

Membrane and shear stress:

The total cross sectional area of the material that has to withstand the membrane stresses from this force can be described as:

$$A = \frac{F}{f} = \frac{3433500N}{387{,}5MPa} = 8860{,}65mm^2 \qquad \text{(Eq. 33)}$$

Where:

- F is the total force acting on the walls
- A is the cross sectional area of the material
- f is the allowable design stress limit ref table

The minimum wall thickness in the support frame due to membrane stresses is then:

$$t = \frac{\frac{A}{2}}{h} = \frac{\frac{8860{,}65mm^2}{2}}{620mm} = 7{,}15mm \qquad \text{(Eq. 34)}$$

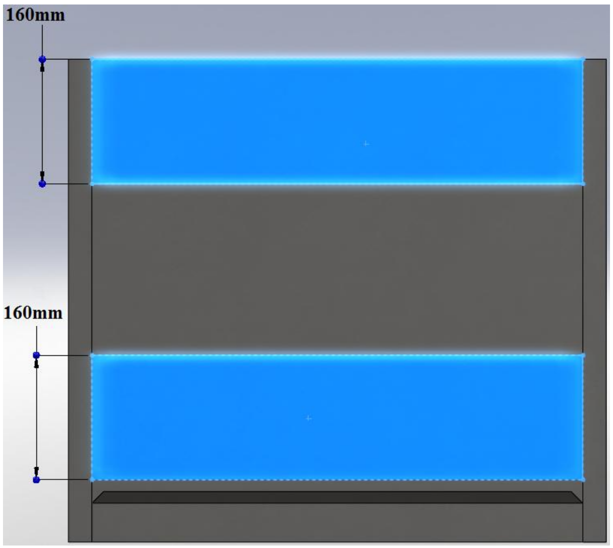
Utilization due to membrane stress:

$$U_{M\,w} = \frac{7{,}15mm}{30mm} = 0{,}25 \rightarrow OK! \qquad \text{(Eq. 35)}$$

Where:

- t is the wall thickness
- h is the height of the support frame
- $U_{M\,w}$ is the membrane utilization in the main walls

The most conservative is to think that it is only the area where the pressure vessel touches the end plate that handle the shear stresses from the load, see Picture 58. The load is divided into half the load on each side.



Picture 58: End wall where the pressure chamber sits against

This give the shear stress in the corner of the end plate:

$$f_{w,M} = \frac{F}{A_{shear}} = \frac{\frac{F}{2}}{2 * t * L_{contact}} = \frac{\frac{3433500N}{2}}{2 * 30mm * 160mm} = 178{,}83MPa \qquad \text{(Eq. 36)}$$

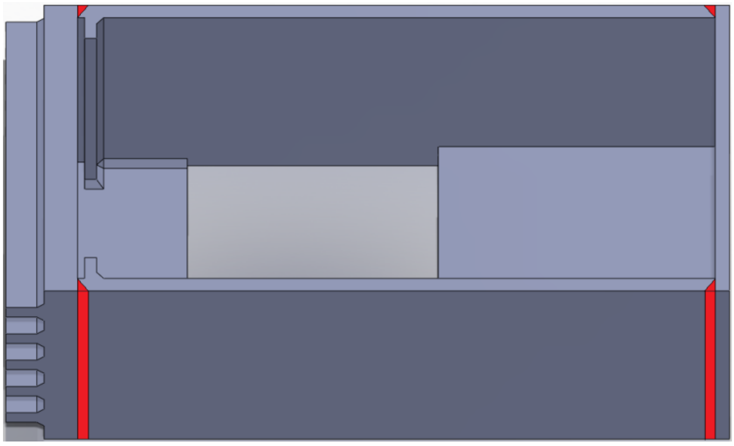
Utilization due to shear stress:

$$U_{S\,w} = \frac{178{,}83MPa}{387{,}5MPa} = 0{,}46 \rightarrow OK! \qquad \text{(Eq. 37)}$$

Where:

- $f_{w,M}$ is the shear stress in the main walls
- A_{shear} is the shear area
- t is the wall thickness
- $L_{contact}$ is the length of the contact surface from the pressure vessel
- $U_{S\,w}$ is the shear utilization in the main walls

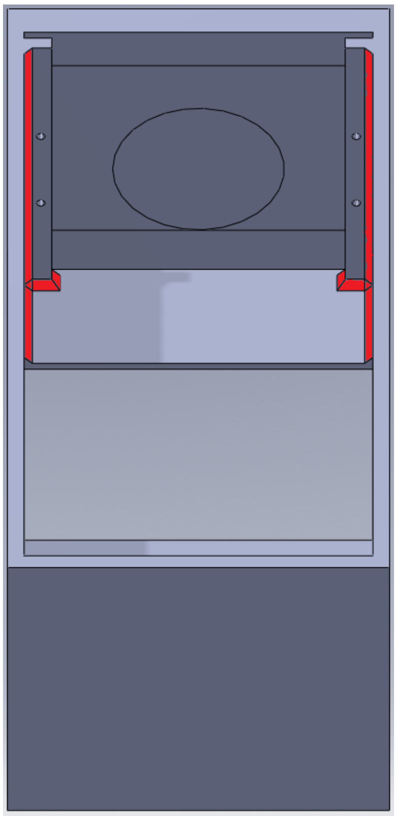
The main walls are welded together with a full penetration butt weld, see Picture 59. According to the NS EN standard, the capacity of a butt weld with full penetration is equal to the capacity of the weakest plates in the joint, assuming that the properties of the welding electrode do not have less quality then the base material. Adding the assumptions that the welds is NDT tested and that the support frame walls can take the load, these welds have sufficient strength. The welds shall be designed as the weld described in NS-EN 13445-3 (E) joint type E15. See Appendix C: Extracts from NS-EN13445:2009.



Picture 59: Full penetration butt weld on support frame main walls

Rails:

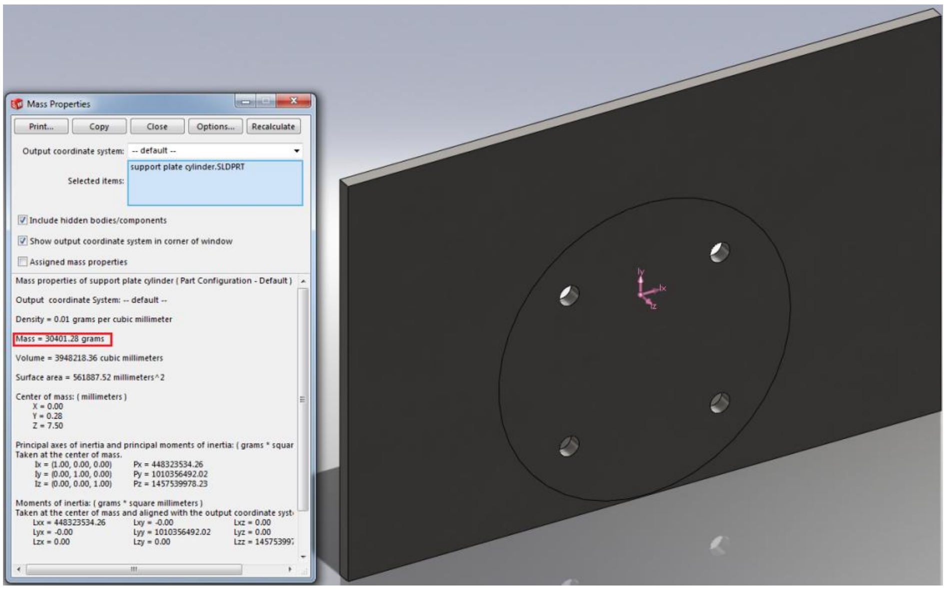
The rails support the plate that holds the cylinder, these welds, see Picture 60, are exposed to shear loading and in a worst case scenario the hole weight of the hydraulic cylinder including the support plate can be applied. According to the NS-EN 13445-3:2009 one sided fillet welds are not allowed for pressure parts. But since these rails only guide the cylinder when it is adjusted into correct height, they bear no load from the pressure, and is therefore designed according to the NS 3472 “Steel structures design rules”



Picture 60: Fillet welds on rails in support frame

Load:

- The weight of the hydraulic cylinder is: 195 kg, see Appendix G: Larzep cylinder
- The weight of the support plate is according to SolidWorks, see Picture 61: 30,4 kg



Picture 61: Weight of hydraulic cylinder support plate

- The factor of weight uncertainties take into account the weight of oil in cylinder, bolts, hoses and other. ω=1,05

The total shear load is then:

$$F_{shear} = (195 + 30{,}4)kg * 9{,}81\frac{m}{s^2} * 1{,}05 = 2321{,}73N \qquad \text{(Eq. 38)}$$

Shear stress and root length:

The shear stress is:

$$f_{w,R} = \frac{F_{shear}}{A} = \frac{F_{shear}}{H_{rails} * \frac{a}{\sin(\theta)}} \qquad \text{(Eq. 39)}$$

Where:

- F_{Shear} is shear force
- A is the cross-sectional area of material with area parallel to the applied force vector.
- H_{rails} is the height of the rails
- Θ is the angle in the fillet weld, normally 45 degrees
- a is the root length of the weld

According to NS 3472:2001 ch. 12.6.2 the design shear stress is:

$$f_{w,d} = \frac{f_u}{\gamma_{M2} * \sqrt{3}} * \frac{1}{\beta_w} \qquad \text{(Eq. 40)}$$

Where:

- $f_{w.d}$ is the design shear stress
- f_u is the tensile strength of the weakest part in the joint
- B_w is the correlation factor, according to NS 3472; $B_w = 1$
- Υ_{M2} is the safety factor for welds, according to NS 3472; $\Upsilon_{M2} = 1{,}25$

Putting Eq. 38,39,40 together:

$$\frac{F_{shear}}{H_{rails} * \frac{a}{\sin(\theta)}} = \frac{f_u}{\gamma_{M2} * \sqrt{3}} * \frac{1}{\beta_w} \qquad \text{(Eq. 41)}$$

$$a = \frac{F_{shear} * \gamma_{M2} * \sqrt{3} * \sin(\theta) * \beta_w}{f_u * H_{rails}} \qquad \text{(Eq. 42)}$$

$$a = \frac{\frac{2321{,}73N}{2} * 1{,}25 * \sqrt{3} * \sin(45) * 1}{930MPa * 605{,}05mm} = 0{,}0032mm \qquad \text{(Eq. 43)}$$

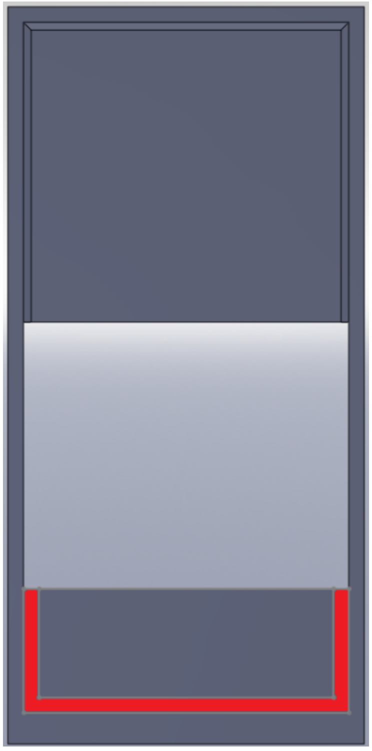
Utilization of fillet welds in rails:

$$U_{W\,R} = \frac{0{,}0032mm}{10mm} = 0{,}00032 \rightarrow OK! \qquad \text{(Eq. 44)}$$

The fillet weld holding the rails needs a root length of minimum 0,0032mm, but is chosen to be 10mm.

Bottom plate 1:

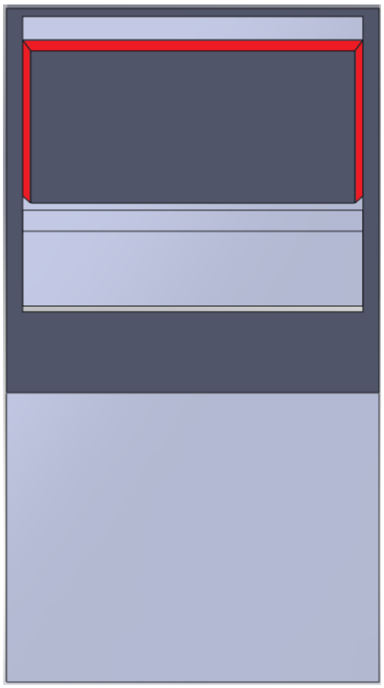
The bottom plate below the rails adding stiffness to the structure and is welded with a full penetration butt weld from the bottom, see Picture 62. According to the NS EN standard, the capacity of a butt weld with full penetration is equal to the capacity of the weakest plates in the joint, assuming that the properties of the welding electrode do not have less quality then the base material. Adding the assumptions that the welds is NDT tested and that the support frame walls can take the load, these welds have sufficient strength. The welds shall be designed as the weld described in NS-EN 13445-3 (E) joint type E15. See Appendix C: Extracts from NS-EN13445:2009.



Picture 62: Full penetration butt weld on bottom plate 1 in support frame

Bottom plate 2:

Bottom plate 2, see Picture 63, is the plate that the pressure vessel sits on. This plate carry the weight of the pressure chamber, coupon, pressure head and bowl. The plate is welded with fillet welds along the edges underneath the plate, the welds are exposed to shear loading. According to the NS-EN 13445-3:2009 one sided fillet welds are not allowed for pressure parts. But since this plate and its welds only support the weight of the pressure chamber, assuming the end plate is 100% stiffened, it take no load from the pressure, and is therefore designed according to the NS 3472 “Steel structures design rules”



Picture 63: Fillet weld on bottom plate 2 in support frame

Load:

- The weight of the pressure vessel is according to SolidWorks: 1141,77 kg
- The weight of the bowl part is according to SolidWorks: 22,72 kg
- The weight of the pressure head part is according to SolidWorks: 32,16 kg
- The weight of the packer coupon is according to SolidWorks: 9,94 kg
- The factor of weight uncertainties take into account the weight of bolts, water in the test rig and equipment for measuring and inspection. ω=1,05

The total shear load is then:

$$F_{shear} = (1141{,}77 + 22{,}72 + 32{,}16 + 9{,}94)kg * 9{,}81\frac{m}{s^2} * 1{,}05 = 12428{,}48\ N \quad \text{(Eq. 45)}$$

Shear stress and root length:

The shear stress is:

$$f_{w,SP} = \frac{F_{shear}}{A} = \frac{F_{shear}}{L_{welds} * \frac{a}{\sin(\theta)}} \quad \text{(Eq. 46)}$$

Where:

- F_{Shear} is shear force
- A is the cross-sectional area of material with area parallel to the applied force vector.
- L_{welds} is the length of the welds bearing the pressure vessel
- Θ is the angle in the fillet weld, normally 45 degrees
- a is the root length of the weld

According to NS 3472:2001 ch. 12.6.2 the design shear stress is:

$$f_{w,d} = \frac{f_u}{\gamma_{M2} * \sqrt{3}} * \frac{1}{\beta_w} \qquad \text{(Eq. 47)}$$

Where:

- $F_{w.d}$ is the design shear stress
- F_u is the tensile strength of the weakest part in the joint
- B_w is the correlation factor, according to NS 3472; $B_w = 1$
- Υ_{M2} is the safety factor for welds, according to NS 3472; $\Upsilon_{M2} = 1{,}25$

Putting eq 45,46,47 together and taking the conservative assumption that only the welds on the longitudinal ends carry the whole weight:

$$\frac{F_{shear}}{L_{welds} * \frac{a}{\sin(\theta)}} = \frac{f_u}{\gamma_{M2} * \sqrt{3}} * \frac{1}{\beta_w} \qquad \text{(Eq. 48)}$$

$$a = \frac{F_{shear} * \gamma_{M2} * \sqrt{3} * \sin(\theta) * \beta_w}{f_u * L_{welds}} \qquad \text{(Eq. 49)}$$

$$a = \frac{\frac{12428{,}48N}{2} * 1{,}25 * \sqrt{3} * \sin(45) * 1}{760MPa * 580mm} = 0{,}0216\,mm \qquad \text{(Eq. 50)}$$

Utilization of fillet weld holding the support plate for the pressure chamber:

$$U_{W\,SP2} = \frac{0{,}0216mm}{10mm} = 0{,}00216 \rightarrow OK! \qquad \text{(Eq. 51)}$$

Where:

- $U_{W\,SP2}$ is the utilization of the fillet weld on the support plate for the pressure vessel

The fillet weld holding the support plate for the pressure chamber needs a root length of minimum 0,0216 mm, but is chosen to be 10mm.

Ribs:

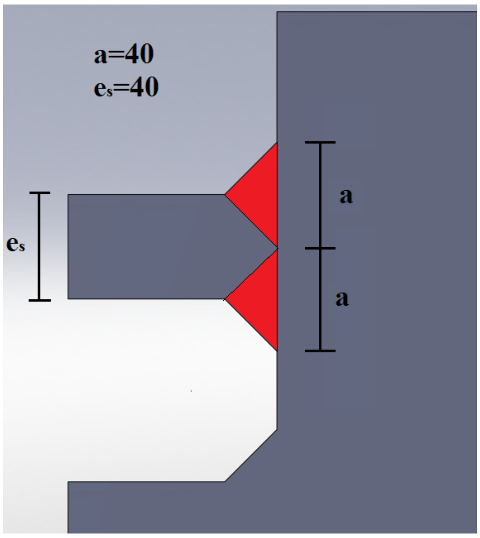
The ribs stiffen the end plate in the support frame where the hydraulic cylinder is supported under loading, see Picture 64. Each rib is welded with a full penetration K-weld, see Picture 65. Designed as the weld described in NS-EN 13445-3 (E) joint type E10. See Appendix C: Extracts from NS-EN13445:2009.

Picture 64: Full penetration K-welds on ribs in support frame

According to the standard the requirements for this specific weld type is:

- a must be more or equal to the thickness of the thinnest plate
- the properties of the welding electrode have same or better quality as the base material
- the welds is NDT tested

To fulfill these requirements the a-measurements have to be 40mm, see Picture 65.



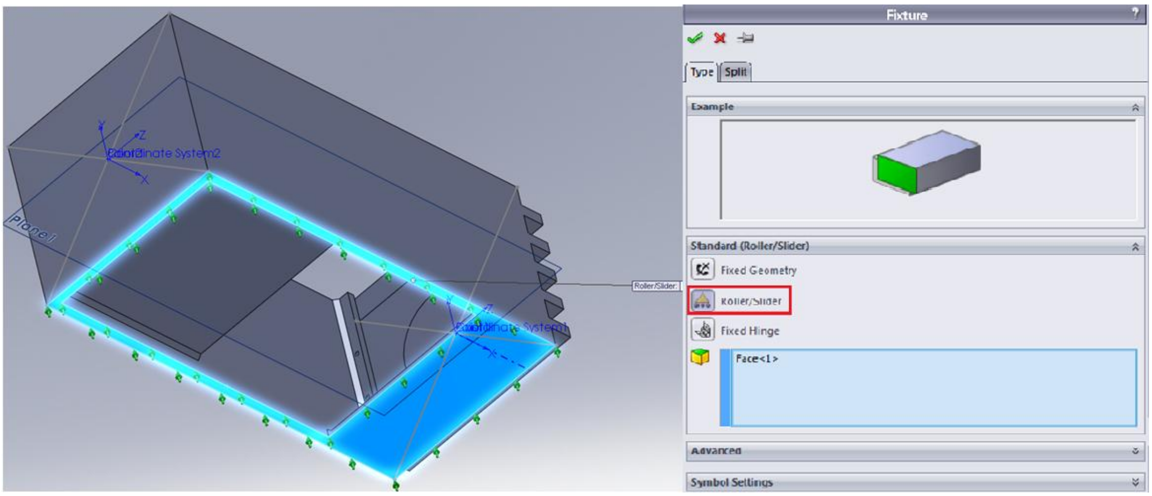
Picture 65: Full penetration K-weld

4.3.2.4 SolidWorks simulations

This chapter describes the process of the SolidWorks simulation on the support frame. In the iteration process several simulations have been run, however it is only the final design simulation that is described in this chapter.

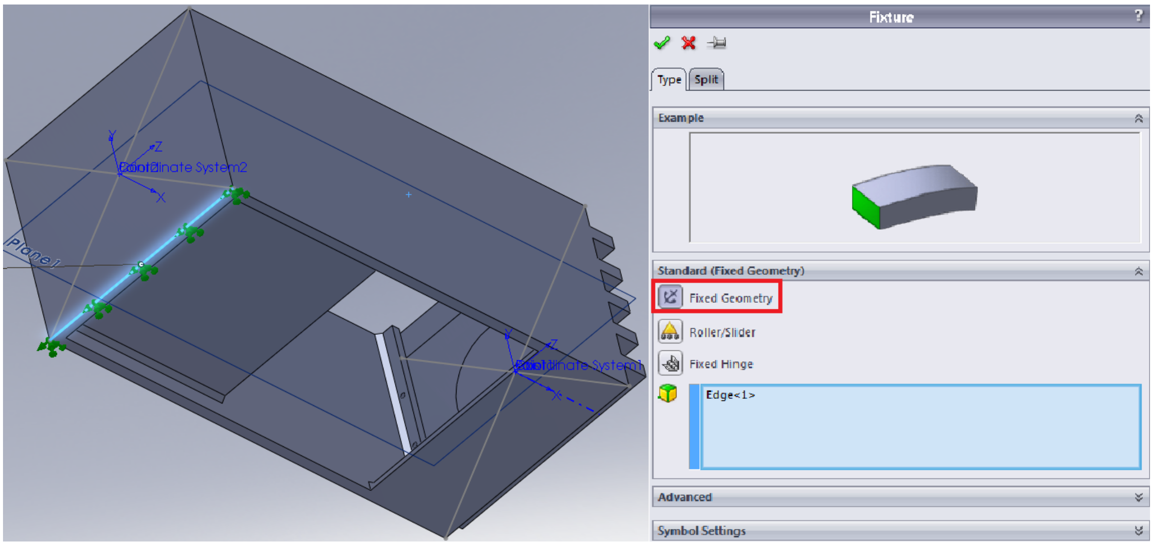
4.3.2.4.1 Linear static study set up

1. Linear static analysis, see Picture 38, is used in the simulation of the support frame, see 3.2.4 Linear static analysis for an explanation of the method and the assumption made.
2. Since this is a simulation of a part not an assembly there will be no contact points between parts.
3. The fixtures of the pressure chamber is:
    a. Soft springs is used to stabilize the model, see Picture 40.
    b. The support frame sits against the floor/table, this is simulated by supporting it on a roller bearing at the bottom, see Picture 66.



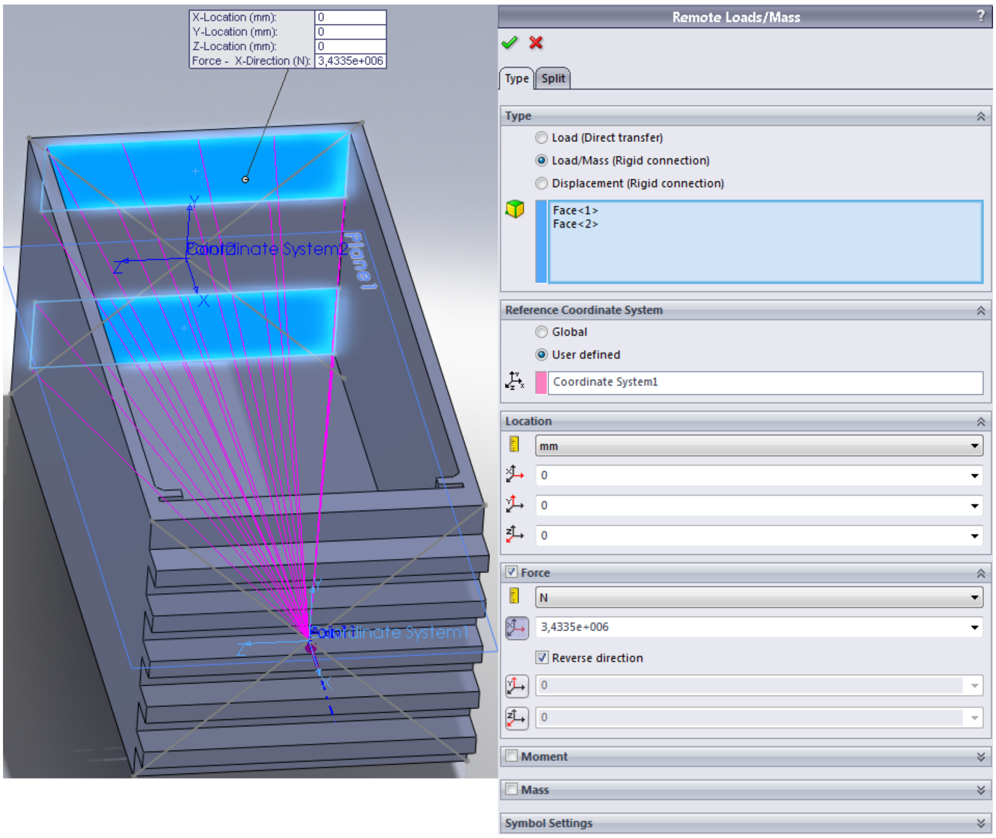
Picture 66: Fixture of support frame

c. To get the model sufficient fixed, one edge is set as a fixed geometry, see Picture 67.

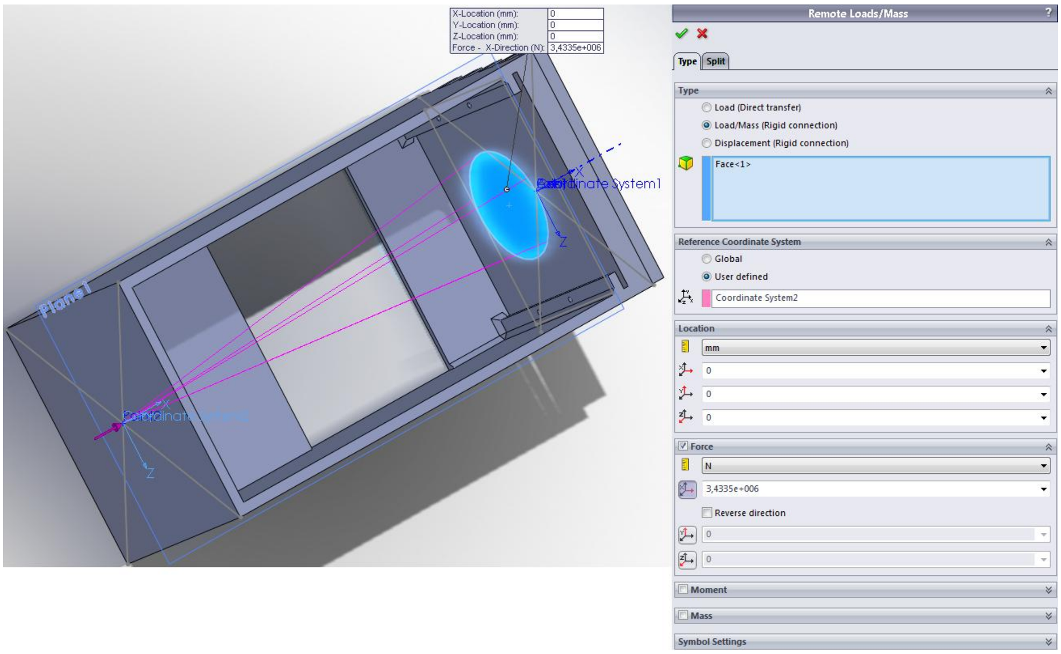


Picture 67: Fixture of support frame

4. The external load applied to the support frame is 350T force from the hydraulic cylinder. The force is applied as a remote load to simulate the added stiffness from the rigid cylinder and the rigid top and bottom plate of the pressure chamber that touches the end plates in the frame, see Picture 68 and Picture 69.



Picture 68: External load on the support frame



Picture 69: External load on the support frame

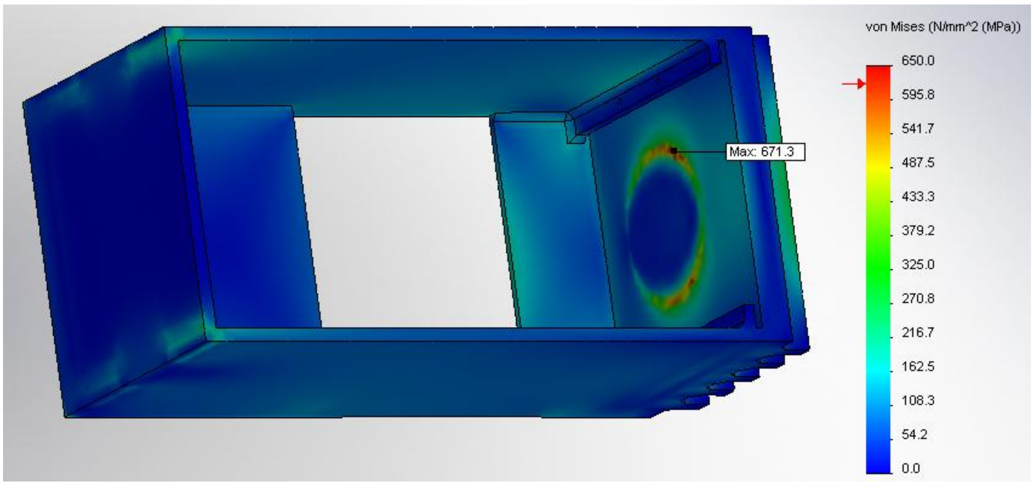
5. The frame model is meshed with fine density, see Picture 43.

4.3.2.4.2 Simulation results design forces

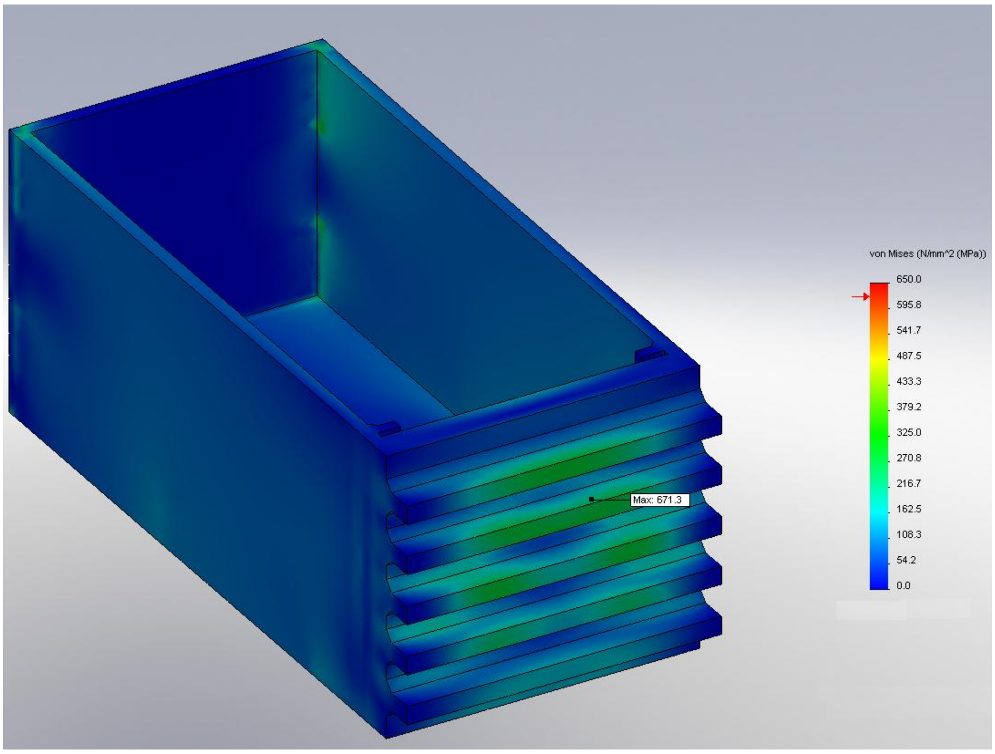
The final measurements on the pressure chamber is according to Picture 56.

The overview of the simulation results, see Picture 70 and Picture 71 ,shows why the end wall supporting the hydraulic cylinder need a large wall thickness and ribs to stiffen it up. The bending moment is the critical component on this wall. Some peak stresses occurs along the edge where the cylinder is supported/pushing against.

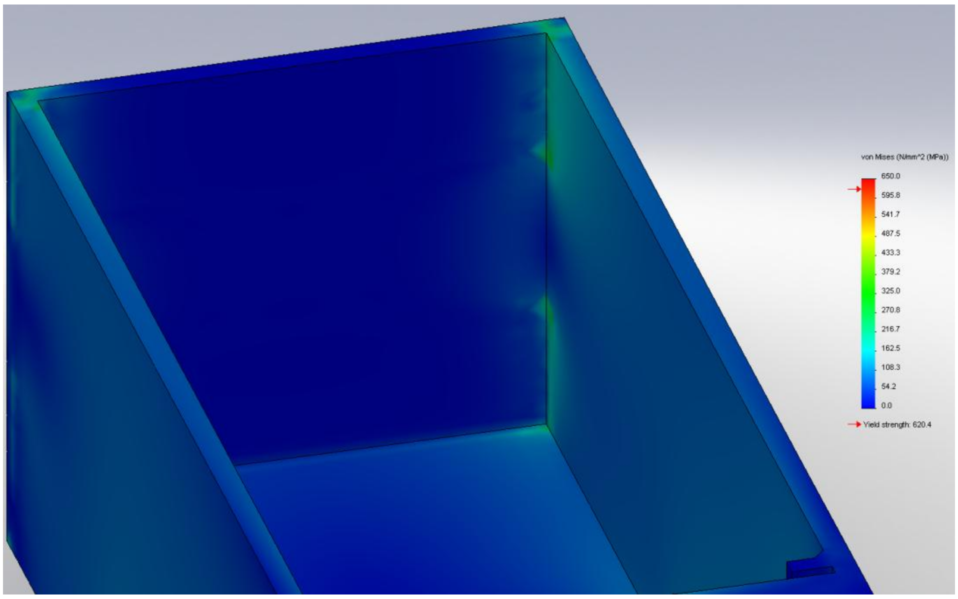
The critical component on the end wall where the pressure vessel is supported/pushing against, is the shear stresses in the corners, see Picture 72.



Picture 70: Overview of simulations results support frame

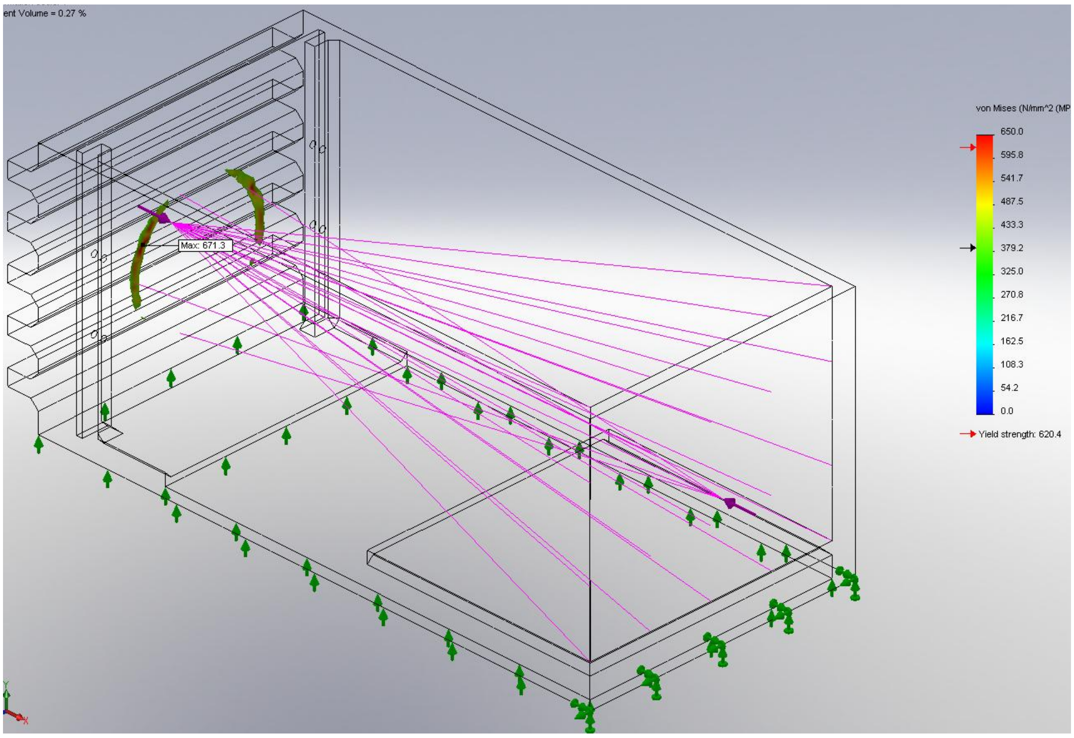


Picture 71: Overview of simulations results support frame



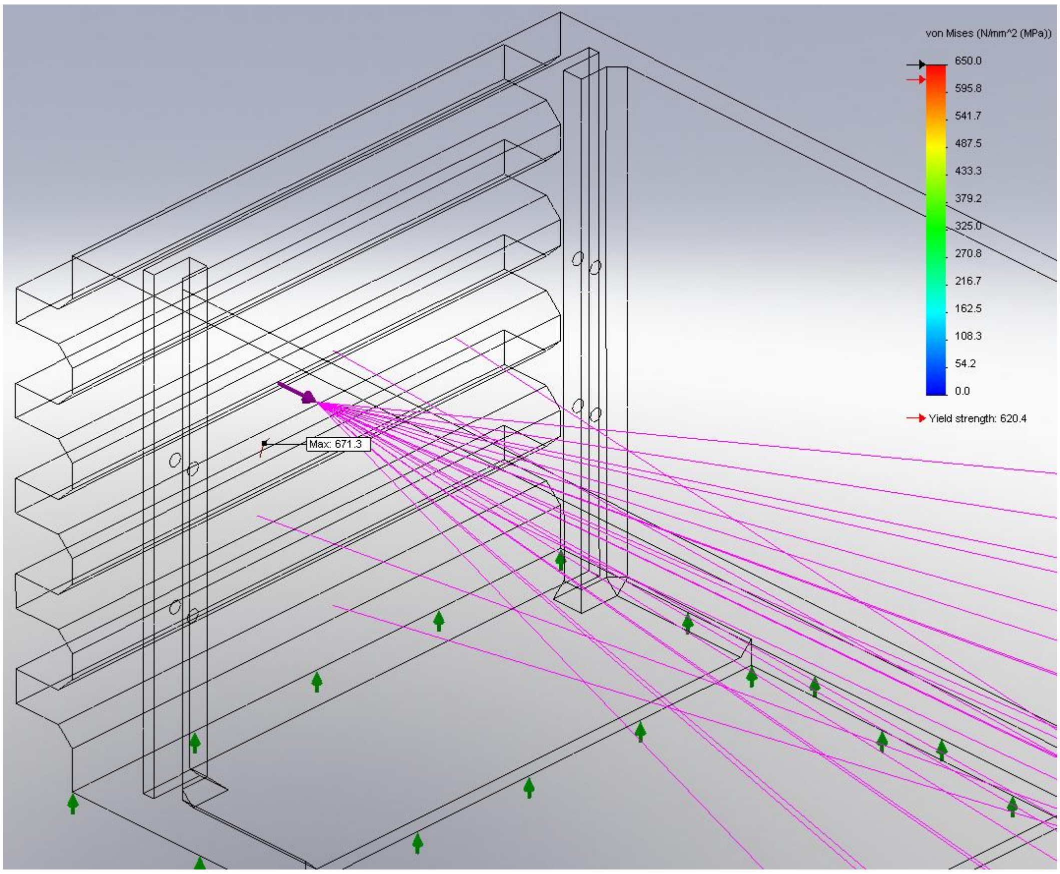
Picture 72: Shear stresses in corners

By using ISO-clipping, see Picture 73, the peak points that exceeds the allowable stress limits easily can be shown.



Picture 73: Peak points that exceeds the allowable stress limit on support frame

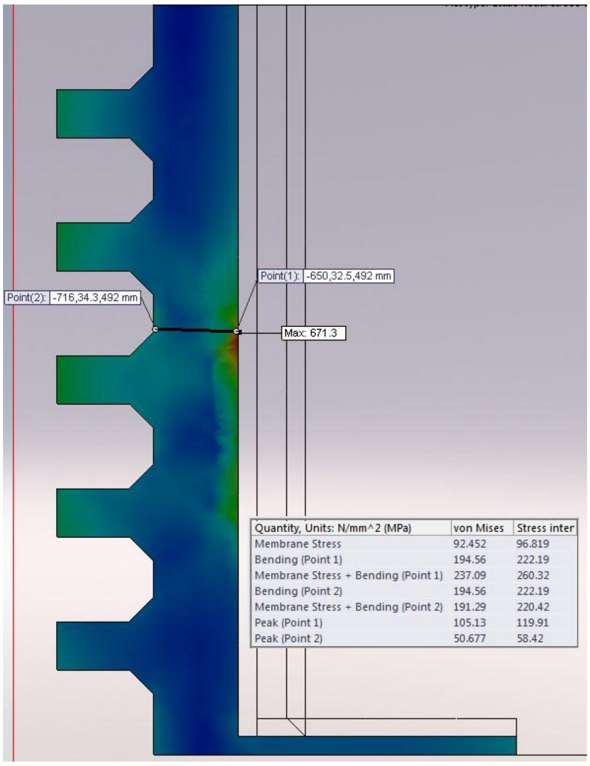
The peak point that exceeds the yield strength of 650MPa can be seen in Picture 74.



Picture 74: Peak points that exceeds the yield strength on suppport frame

To verify the peak points according to membrane stress and membrane + bending stress a linearization through the wall thickness for the relevant peak points is done.

Point 1:



Picture 75: Linearization on support frame point 1

This linearization is done to verify the wall thickness where the maximum peak point is located. None of the linearized von Mises stresses, see Picture 75, is above the allowable stresses calculated according to NS-EN13445-3:2009, see Table 12:

- Membrane stress: 92,45MPa < 387,5MPa→OK
- Bending stress point 1: 194,56MPa <387,5MPa→OK
- Membrane + bending stress point 1: 237,09MPa < 581,3MPa→OK
- Bending stress point 2: 194,56MPa <387,5MPa→OK
- Membrane + bending stress point 2: 191,29MPa < 581,3MPa→OK

Utilization:

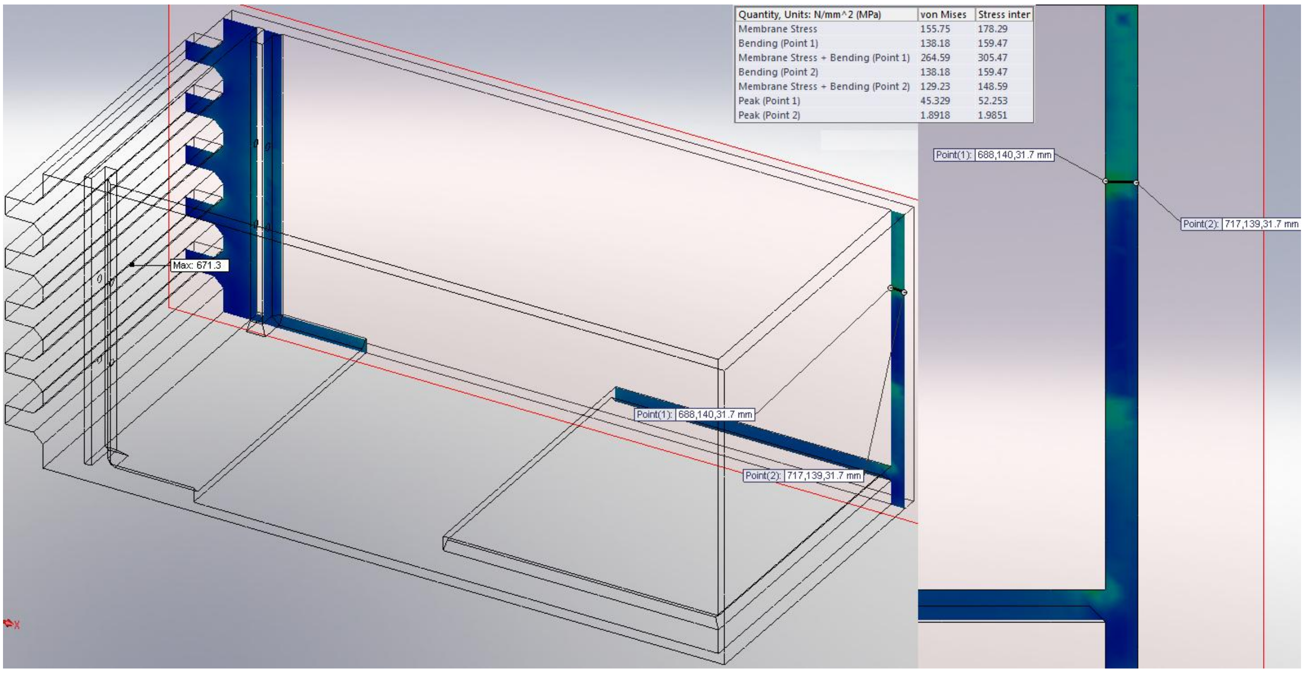
- Membrane stress:

$$U_{M\ SF1} = \frac{P_{M\ C1}}{P_{M\ max}} = \frac{92,45}{387,5} = 0,24 \qquad \text{(Eq. 52)}$$

- Membrane + bending stress:

$$U_{MB\ SF1} = \frac{P_{MB\ C1}}{P_{MB\ max}} = \frac{237,09}{581,3} = 0,41 \qquad \text{(Eq. 53)}$$

Point 2:



Picture 76: Linearization on support frame point 2

The linearization on this point is done to verify the wall thickness in and around the corners that is subjected to shear stress. None of the linearized von Mises stresses, see Picture 76, is above the allowable stresses calculated according to NS-EN13445-3:2009, see Table 12:

- Membrane stress: 155,75MPa < 387,5MPa→OK
- Bending stress point 1: 138,18MPa <387,5MPa→OK
- Membrane + bending stress point 1: 264,59MPa < 581,3MPa→OK
- Bending stress point 2: 138,18MPa <387,5MPa→OK
- Membrane + bending stress point 2: 129,23MPa < 581,3MPa→OK

Utilization:

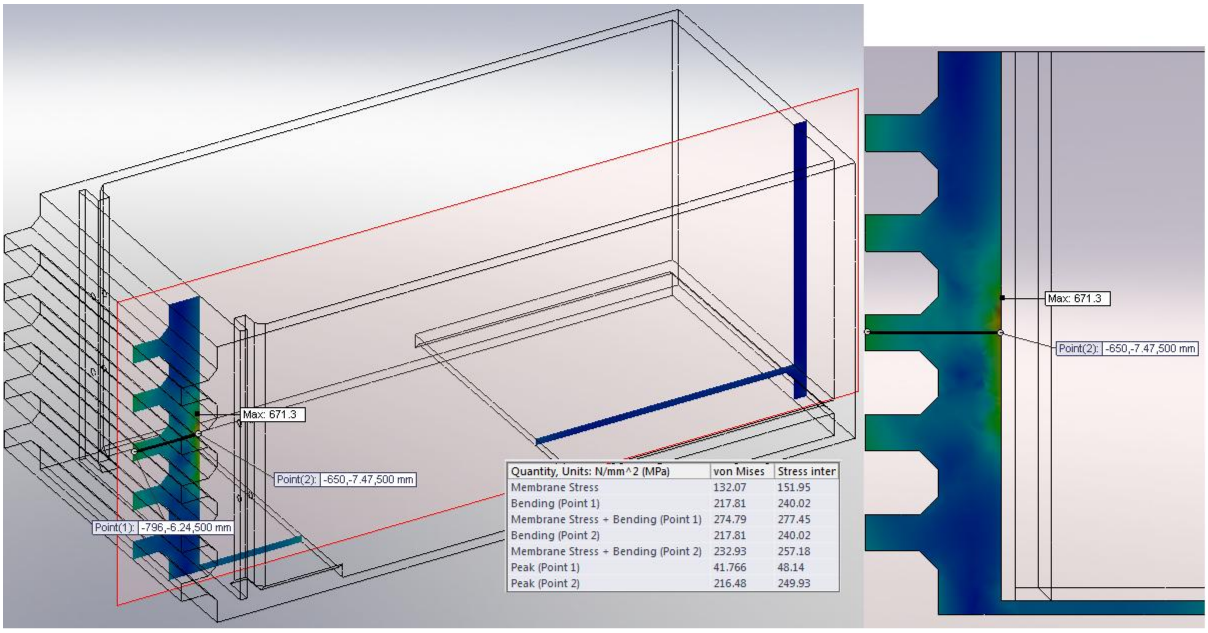
- Membrane stress:

$$U_{M\ SF2} = \frac{P_{M\ C1}}{P_{M\ max}} = \frac{155{,}75}{387{,}5} = 0{,}40 \qquad \text{(Eq. 54)}$$

- Membrane + bending stress:

$$U_{MB\ SF2} = \frac{P_{MB\ C1}}{P_{MB\ max}} = \frac{264{,}59}{581{,}3} = 0{,}46 \qquad \text{(Eq. 55)}$$

Point 3:



Picture 77: Linearization on support frame point 3

This linearization is done to verify the utilization of the wall thickness including the ribs. None of the linearized von Mises stresses, see Picture 77, is above the allowable stresses calculated according to NS-EN13445-3:2009, see Table 12:

- Membrane stress: 132,07MPa < 387,5MPa→OK
- Bending stress point 1: 217,81MPa <387,5MPa→OK
- Membrane + bending stress point 1: 274,79MPa < 581,3MPa→OK
- Bending stress point 2: 217,81MPa <387,5MPa→OK
- Membrane + bending stress point 2: 232,93MPa < 581,3MPa→OK

Utilization:

- Membrane stress:

$$U_{M\ SF3} = \frac{P_{M\ C1}}{P_{M\ max}} = \frac{132{,}07}{387{,}5} = 0{,}34 \qquad \text{(Eq. 56)}$$

- Membrane + bending stress:

$$U_{MB\ SF3} = \frac{P_{MB\ C1}}{P_{MB\ max}} = \frac{274{,}79}{581{,}3} = 0{,}47 \qquad \text{(Eq. 57)}$$

4.3.3 Hydraulic cylinder

TDW's requirements regarding forces to the test rig is:

- Able to simulate pipeline pressure of 350bar on a 48" packer.
- Able to use same cylinder on all dimensions
- Able to set packers with large extrusion gaps

4.3.3.1 Force

From the calculations done in chapter 4.3.1.3.1 Recalculation of packer coupon forces, it is showed that the maximum pipeline pressure that can be simulated with the 350T force from the Larzep DDR35020 cylinder is 357,5bar>350→OK. The potential for simulating 357,5 bar pipeline pressure on the 48" packer is 2,14% higher than the minimum requirement of 350bar.

The next cylinder with lower force capacity is the Powerteam RD30013 cylinder, see Appendix J: Cylinder price offers, with capacity of 300T. If we run a calculation on this cylinder, it can be seen that this cylinder are able to simulate a pipeline pressure of 304,3bar<350bar→ Not OK.

4.3.3.2 Piston stroke

From the calculations in chapter 4.3.1.3.1 Recalculation of packer coupon forces, it can be seen that the change in the packer width from unset to set position with maximum extrusion gap on the existing 48" packer coupon is 103,8mm. To make the rig flexible to future packer geometries and dimension some margin of extension in deformation have to be added. See from Appendix J: Cylinder price offers that the Larzep DDR35020 cylinder have 200mm piston stroke. This is 92,7% margin of extension in packer deformation, due to the depth of the chamber of 440mm a total of 200mm deformation seems ok. Alternatively it can be upgraded to the Larzep DDR35025 cylinder with 250mm cylinder stroke.

4.3.4 Bowl

4.3.4.1 Material selection

The material selected for the bowl is selected with background in:

- The high forces it have to withstand
- Price
- Availability at steel suppliers
- Machining properties

The material that fulfill the requirements best is the S355J2. S355 structural steel plate is a high-strength low-alloy European standard structural steel covering four of the six "Parts" within the EN 10025 – 2004 standard. With minimum yield of 50,000 KSI, it meets requirements in chemistry and physical properties similar to ASTM A572 / 709. See Appendix F: S355 EN 10025:2004 for the full overview of the material.

The material meets the requirements in NS-EN 13445-2:2009 for composition, minimum elongation of 14% after fracture and impact energy measured on a Charpy-V-notch impact test. See Appendix C: Extracts from NS-EN13445:2009

The allowable design stress levels is calculated according to NS-EN 13445-3:2009, see Table 13. Where the input data come from Appendix F: S355 EN 10025:2004

Material inndata				
Description	**Notation**	**Unit**	**Value**	**Note**
Yield strength	σy	N/mm^2	315	
Tensile strength	σu	N/mm^2	630	
Allowable stress and safety level				
Description	**Notation**	**Unit**	**Value**	**Note**
z-factor	z		1	1 if no longitudinal or circumferential weld
Allowable design stress level	f	N/mm^2	210,0	NS-EN 13445-3:2009 chapter 6.2
Allowable membrane stress	f * z	N/mm^2	210,0	NS-EN 13445-3:2009 eq: 15.5.3-1
Allowable sum of membran and bending stress	1,5 * f * z	N/mm^2	315,0	NS-EN 13445-3:2009 eq: 15.5.3-2

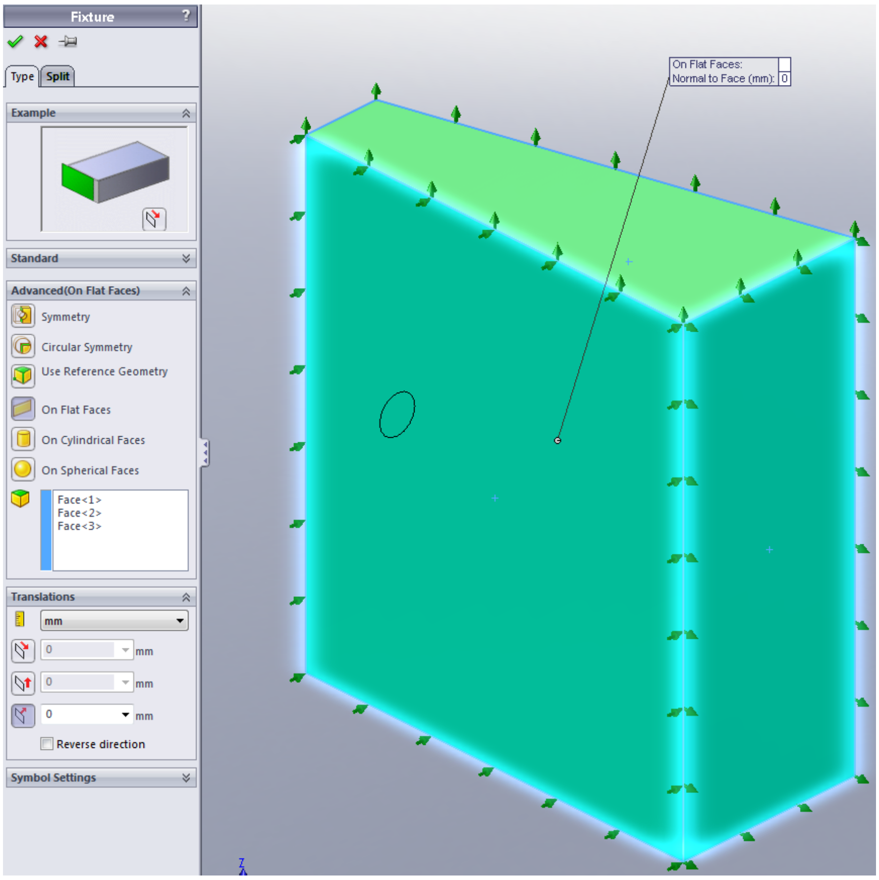
Table 13: Allowable design stress calculation bowl

4.3.4.2 Fabrication method

The bowl will be machined out of a solid block of steel or steel plate.

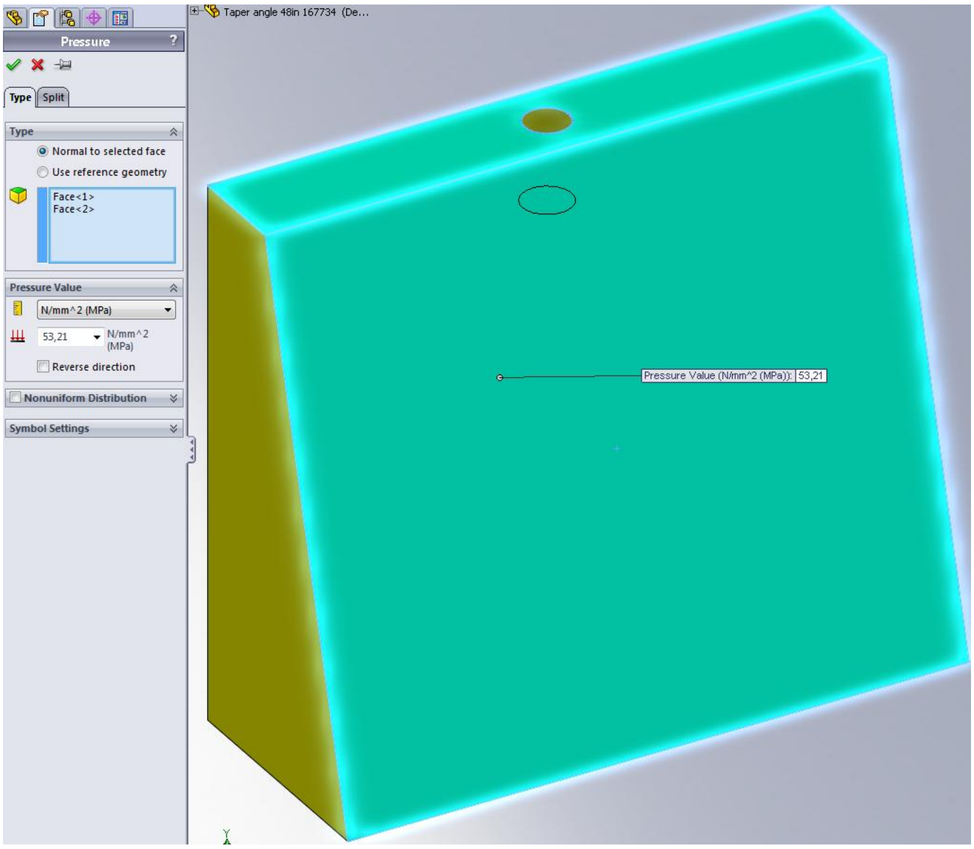
4.3.4.3 Linear static study set up

1. Linear static analysis, see Picture 38, is used in the simulation of the bowl part, see 3.2.4 Linear static analysis for an explanation of the method and the assumption made.
2. Since this is a simulation of a part not an assembly there will be no contact points between parts.
3. The fixtures of the bowl is:
    a. Soft springs is used to stabilize the model, see Picture 40.
    b. The bowl sits against the bottom, end and side walls in the pressure chamber. This is simulated by supporting it on flat faces on the relevant faces, see Picture 78.



Picture 78: Fixture of bowl

4. The external load applied to the bowl is the pressure from the packer on the front face, and pressure from the water on the top. The pressure is 53,21MPa on both faces, see Picture 79.



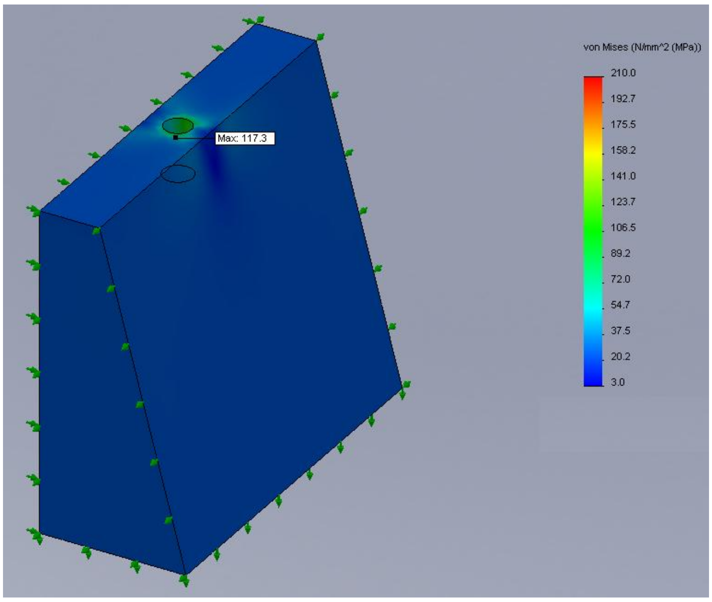
Picture 79: External load on bowl

5. The bowl is meshed with fine density, see Picture 43.

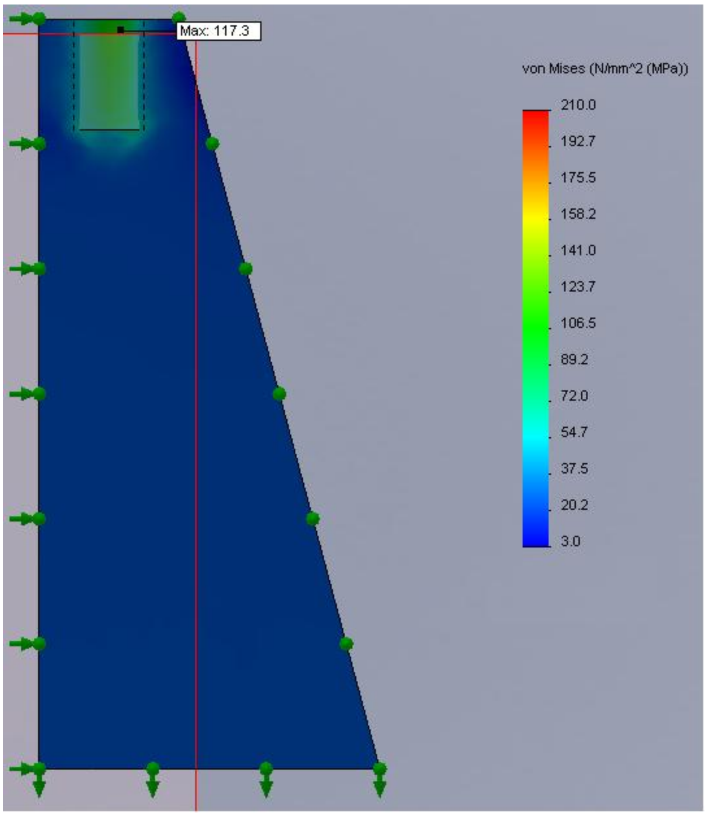
4.3.5.4 Simulation results design forces

The final measurements on the bowl is according to Appendix H: SolidWorks drawings

Picture 80 and Picture 81 shows the results from the simulation.



Picture 80: Overview of simulation results bowl



Picture 81: Overview of simulation results bowl

No points on the bowl exceeds the allowable stress limit of 210MPa → The bowl is ok dimensioned.

4.3.5 Pressure head

4.3.5.1 Material selection

The material selected for the pressure head is selected with background in:

- The high forces
- Price
- Availability at steel suppliers
- Machining properties
- Weldability

The material with a high yield and tensile strength that meet the requirements best and fulfill the requirements in the NS-EN 13445 is the same material as in the pressure chamber and support frame. The S690QL, see Appendix E: S690QL is a high yield structural steel grade produced in compliance with EN 10025:6:2004. The material is heat treated using the quench and temper process and has good bending and welding properties.

The material meets the requirements in NS-EN 13445-2:2009 for composition, minimum elongation of 14% after fracture and impact energy measured on a Charpy-V-notch impact test. See Appendix C: Extracts from NS-EN13445:2009

The allowable design stress levels is calculated according to NS-EN 13445-3:2009, see Table 14. Where the input data come from Appendix E: S690QL . Note the higher yield and tensile strength used in this calculation compared to the pressure vessel calculation, the reason is the plate thickness.

Material inndata				
Description	**Notation**	**Unit**	**Value**	**Note**
Yield strength	σy	N/mm^2	650	
Tensile strength	σu	N/mm^2	930	
Allowable stress and safety level				
Description	**Notation**	**Unit**	**Value**	**Note**
z-factor	z		1	1 if no longitudinal or circumferential weld
Allowable design stress level	f	N/mm^2	387,5	NS-EN 13445-3:2009 chapter 6.2
Allowable membrane stress	f * z	N/mm^2	387,5	NS-EN 13445-3:2009 eq: 15.5.3-1
Allowable sum of membran and bending stress	1,5 * f * z	N/mm^2	581,3	NS-EN 13445-3:2009 eq: 15.5.3-2

Table 14: Allowable design stress calculation pressure head

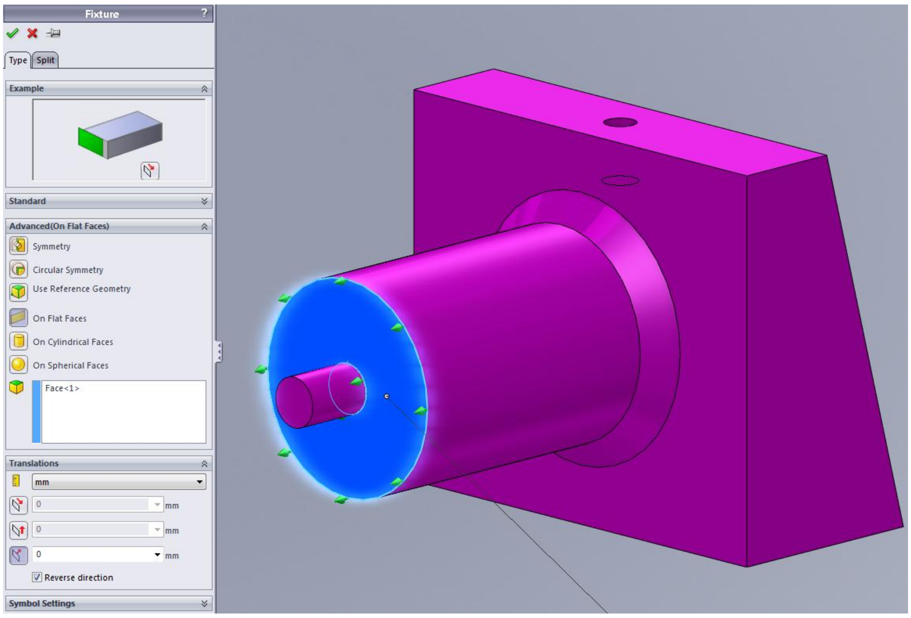
4.3.4.2 Fabrication method

The pressure head rectangular part will be machined out of a solid block of steel or steel plate. And the bolt/rod will be welded on to it. See Appendix H: SolidWorks drawings for complete measurements.

4.3.5.3 Linear static study set up

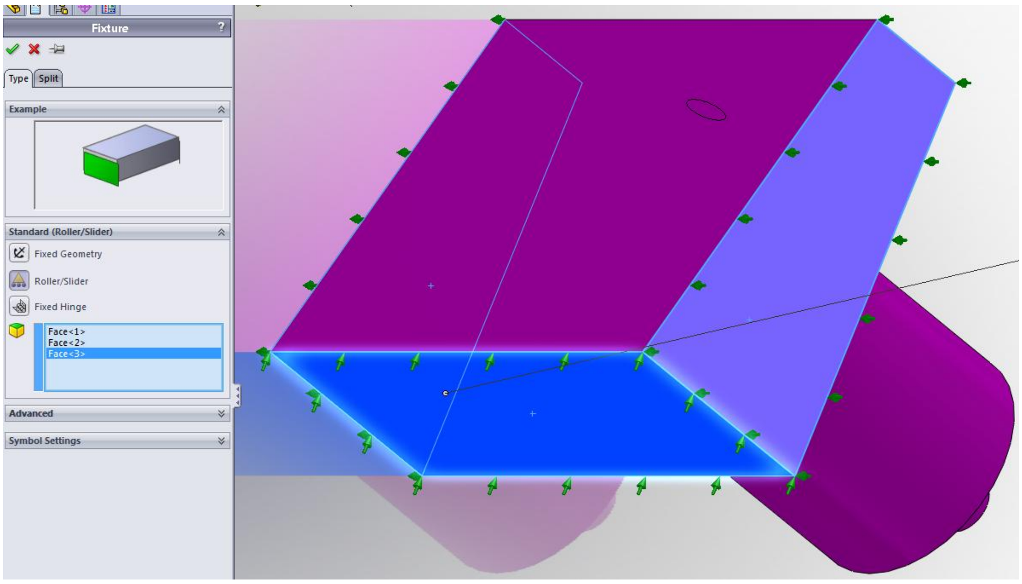
1. Linear static analysis, see Picture 38, is used in the simulation of the pressure head, see 3.2.4 Linear static analysis for an explanation of the method and the assumption made.
2. Since this is a simulation of a part not an assembly there will be no contact points between parts.
3. The fixtures of the pressure head is:
    a. Soft springs is used to stabilize the model, see Picture 40.

b. The applied force to the system all goes through the pressure head. The hydraulic cylinder pushes against the rod on the pressure head to apply pressure to the packer. There are two ways to simulate this, one is to fix the end of the rod where the cylinder pushes and apply pressure to the side facing the packer coupon. Another is to fix the side facing the packer and apply force from the cylinder to the rod. In this case it is the first method that is used, see Picture 82.



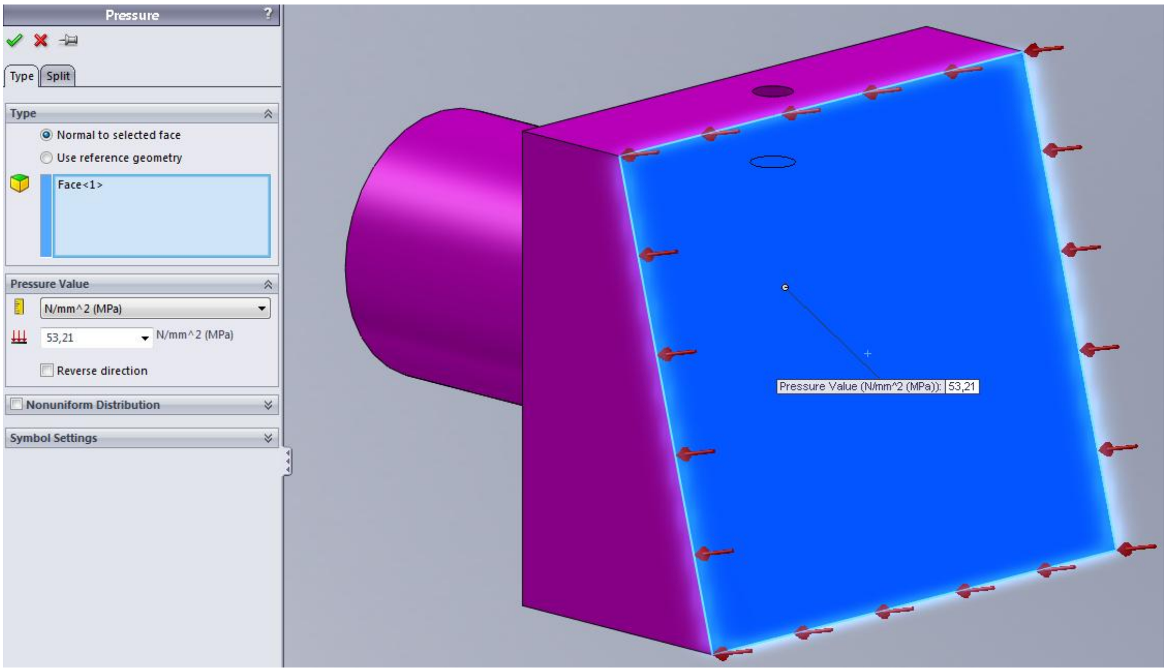
Picture 82: Fixture of pressure head

c. The pressure head slides along the bottom and left and right sides in the pressure vessel. To simulate this the model is fixed with roller bearing on these three faces in the relevant direction, see Picture 83.



Picture 83: Fixture of pressure head

4. The external load applied to the pressure head is the pressure of 53,21MPa the packer apply to the front face, see Picture 84.



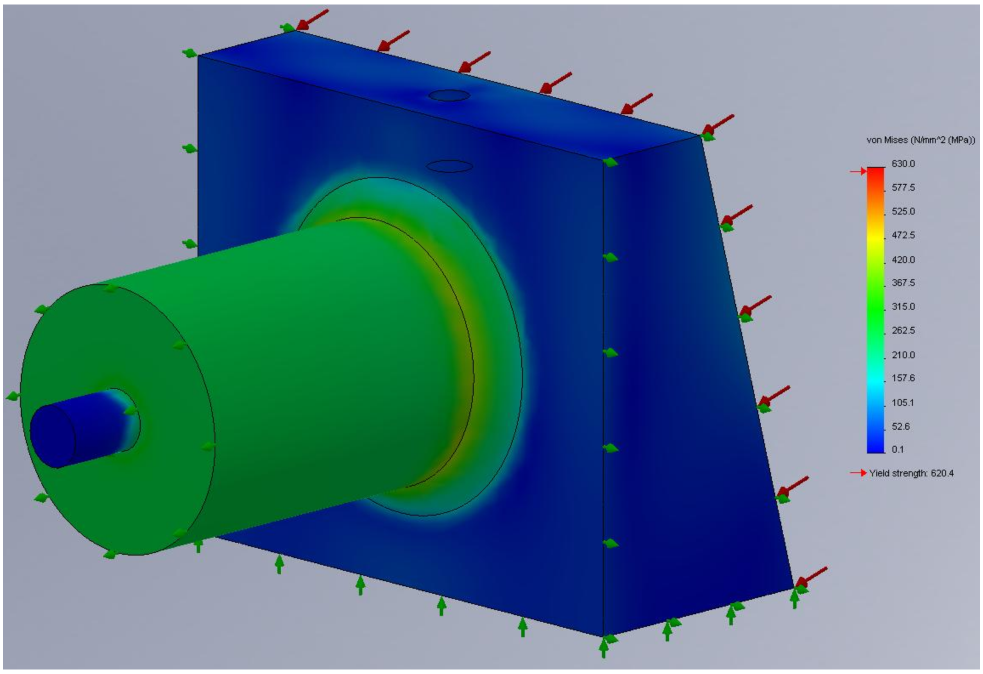
Picture 84: External load on pressure head

5. The model is meshed with fine density, see Picture 43.

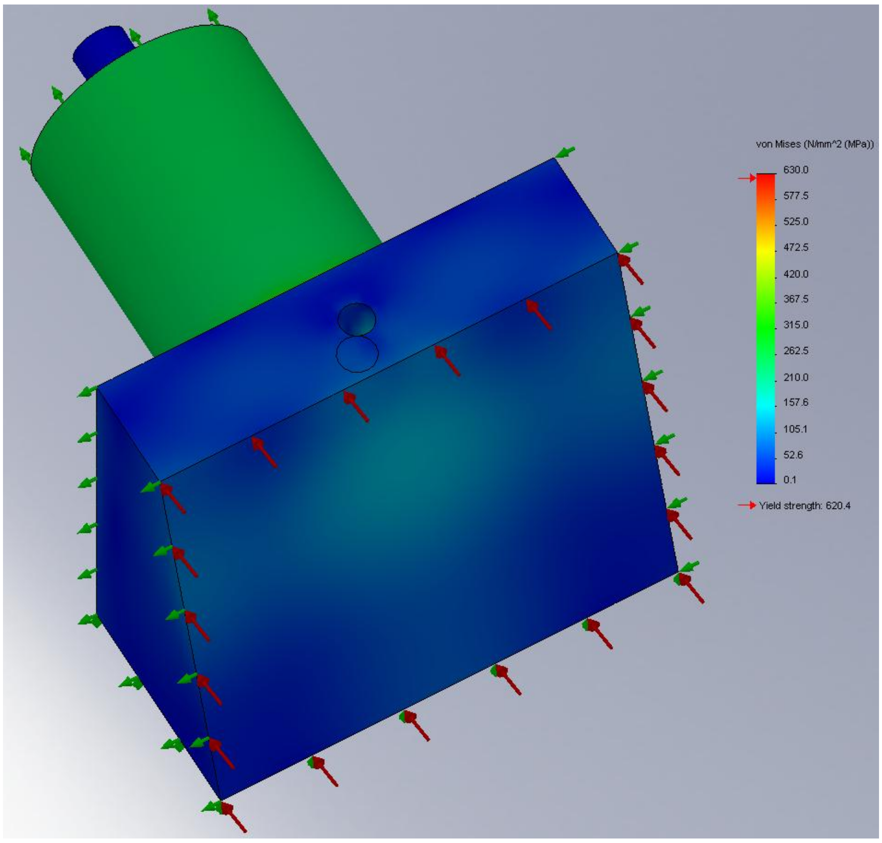
4.3.5.4 Simulation results design forces

The final measurements on the pressure head is according to Appendix H: SolidWorks drawings

Picture 85 and Picture 86 shows the results from the simulation.

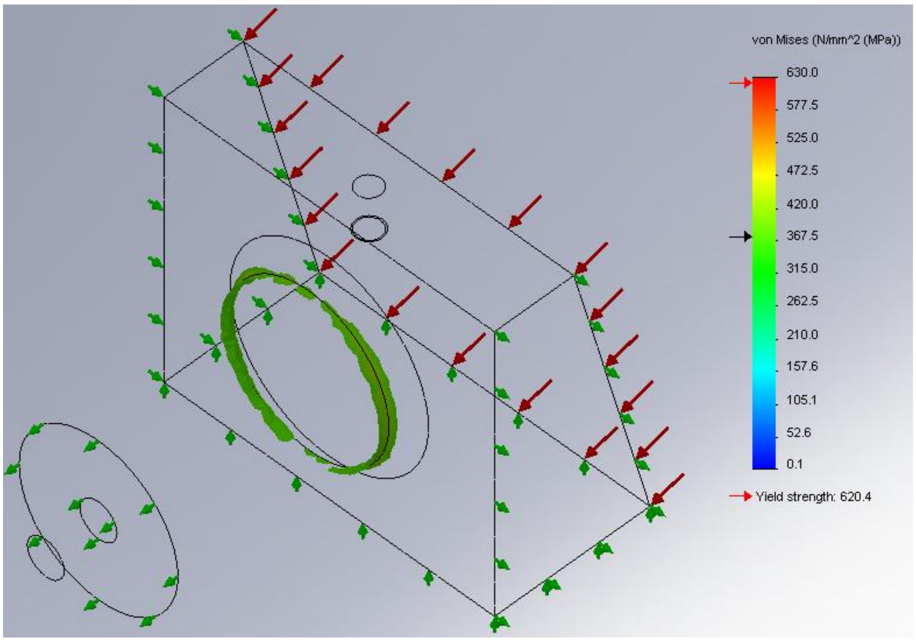


Picture 85: Overview of simulation results on pressure head



Picture 86: Overview of simulation results on pressure head

By using ISO-clipping, see Picture 87, the peak points that exceeds the allowable stress limits of 375MPa easily can be shown.



Picture 87: Peak points that exceeds the allowable stress limit on the pressure head

Only peak points exceeds the allowable stress limit of 375MPa and none points on the part exceeds the yield strength limit → The pressure head is OK dimensioned.

4.3.6 Interface between hydraulic cylinder and pressure head

4.3.5.1 Material selection

The material selected for the interface is selected with background in:

- The high forces
- Price
- Availability at steel suppliers
- Machining properties

The material with a high yield and tensile strength that meet the requirements best and fulfill the requirements in the NS-EN 13445 is the same material as in the pressure chamber, support frame and Pressure head. The S690QL, see Appendix E: S690QL is a high yield structural steel grade produced in compliance with EN 10025:6:2004. The material is heat treated using the quench and temper process and has good bending and welding properties.

The material meets the requirements in NS-EN 13445-2:2009 for composition, minimum elongation of 14% after fracture and impact energy measured on a Charpy-V-notch impact test. See Appendix C: Extracts from NS-EN13445:2009

The allowable design stress levels is calculated according to NS-EN 13445-3:2009, see Table 15. Where the input data come from Appendix E: S690QL . Note the higher yield and tensile strength used in this calculation compared to the pressure vessel calculation, the reason is the plate thickness.

Material inndata				
Description	**Notation**	**Unit**	**Value**	**Note**
Yield strength	σy	N/mm^2	650	
Tensile strength	σu	N/mm^2	930	
Allowable stress and safety level				
Description	**Notation**	**Unit**	**Value**	**Note**
z-factor	z		1	1 if no longitudinal or circumferential weld
Allowable design stress level	f	N/mm^2	387,5	NS-EN 13445-3:2009 chapter 6.2
Allowable membrane stress	f * z	N/mm^2	387,5	NS-EN 13445-3:2009 eq: 15.5.3-1
Allowable sum of membran and bending stress	1,5 * f * z	N/mm^2	581,3	NS-EN 13445-3:2009 eq: 15.5.3-2

Table 15: Allowable stress limit calculation on interface

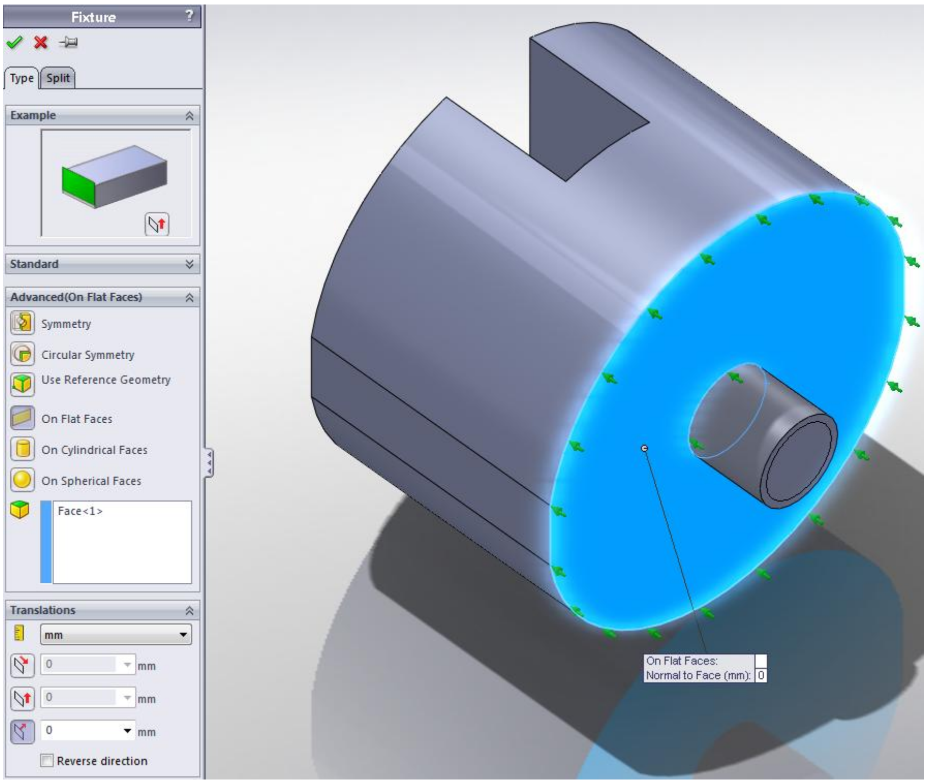
4.3.4.2 Fabrication method

The interface will be machined out of a solid bolt or block of steel alternatively a steel plate.

4.3.5.3 Linear static study set up

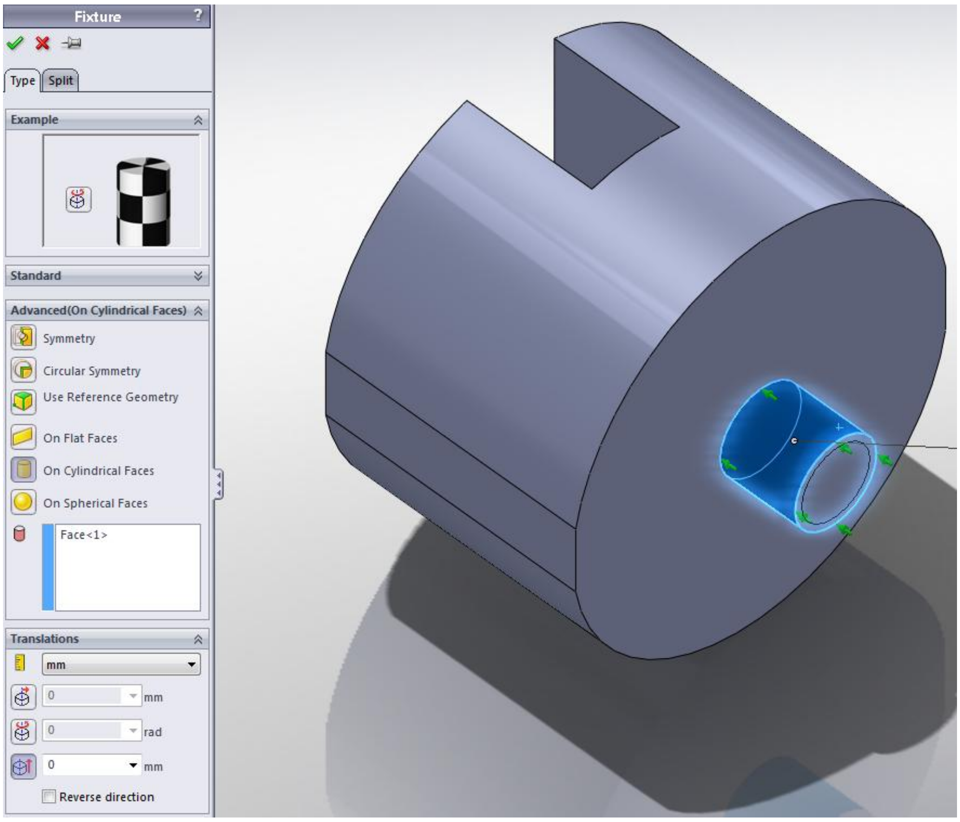
1. Linear static analysis, see Picture 38, is used in the simulation of the interface, see 3.2.4 Linear static analysis for an explanation of the method and the assumption made.
2. Since this is a simulation of a part not an assembly there will be no contact points between parts.
3. The fixtures of the interface is:
   a. Soft springs is used to stabilize the model, see Picture 40.
   b. The interface is simply explained a steel block that sits between the hydraulic cylinder piston and the rod on the bowl. The purpose of the interface is to help supporting the interface between these two components. The force from the

hydraulic cylinder go straight through the interface. To simulate this, one of the sides in the interface is fixed on flat face, see Picture 88.



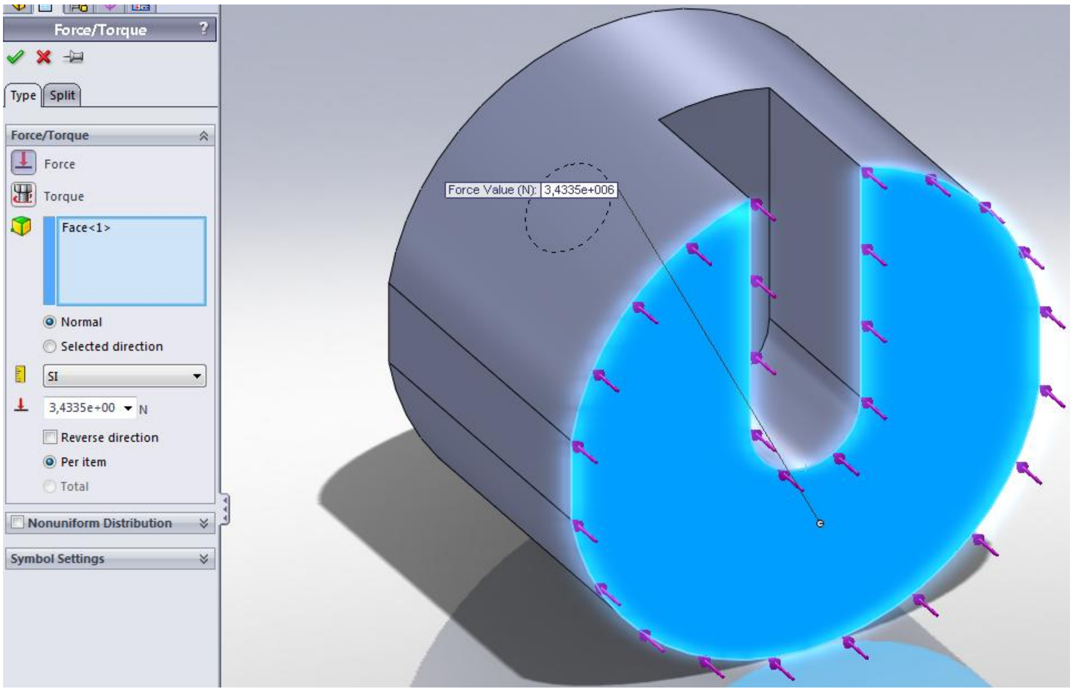
Picture 88: Fixture of interface

c. The rod sticking out of the interface sits inside a hole in the piston rod on the cylinder. It is assumed that this rod do not take any forces and is therefore supported on cylindrical faces with opportunity to move in the force direction, see Picture 89.



Picture 89: Fixture of interface

4. The external load applied to the interface is the maximum force from the hydraulic cylinder of 350T, see Picture 90.



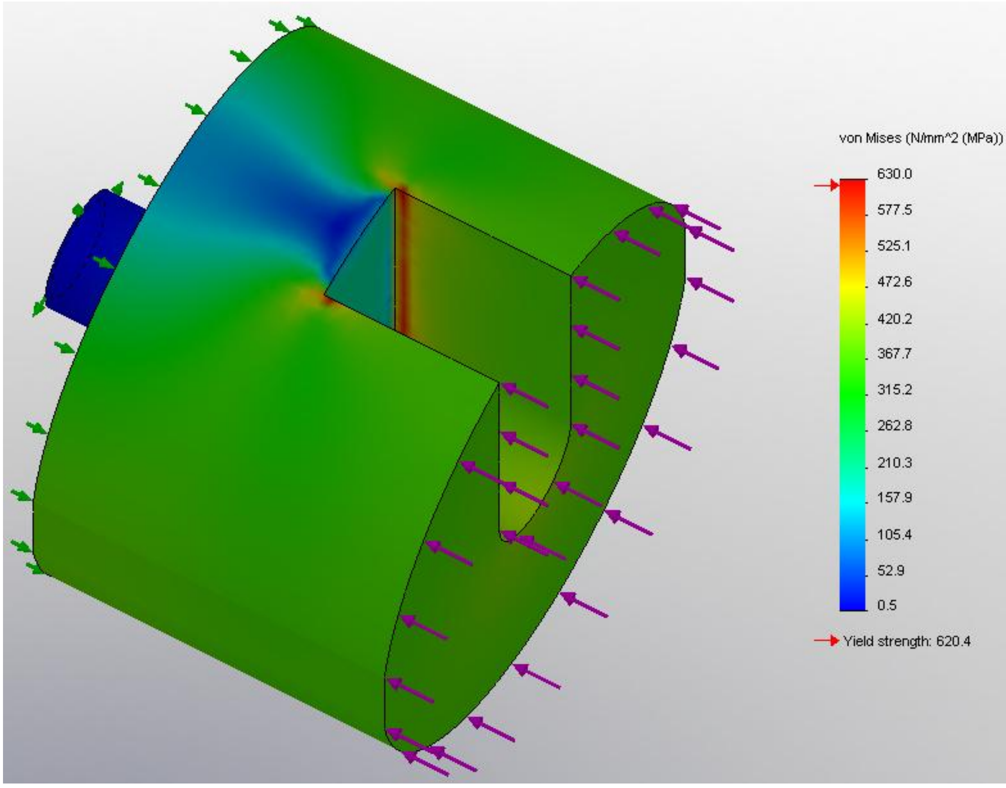
Picture 90: External load on interface

5. The model is meshed with fine density, see Picture 43.

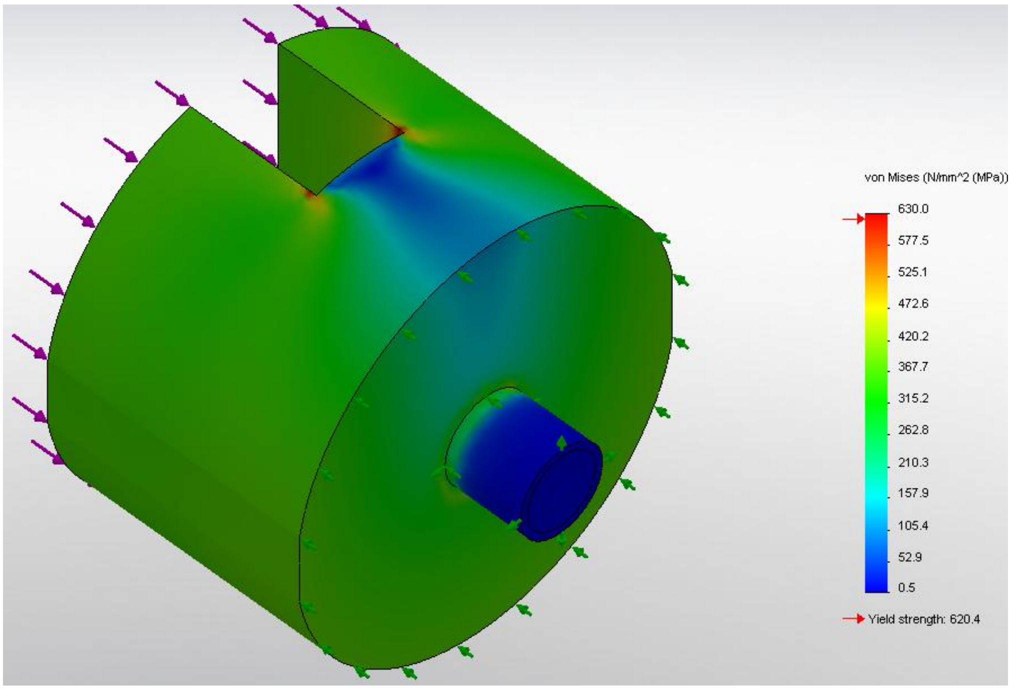
4.3.5.4 Simulation results design forces

The final measurements on the interface is according to Appendix H: SolidWorks drawings

Picture 91 and Picture 92 shows the results from the simulation.

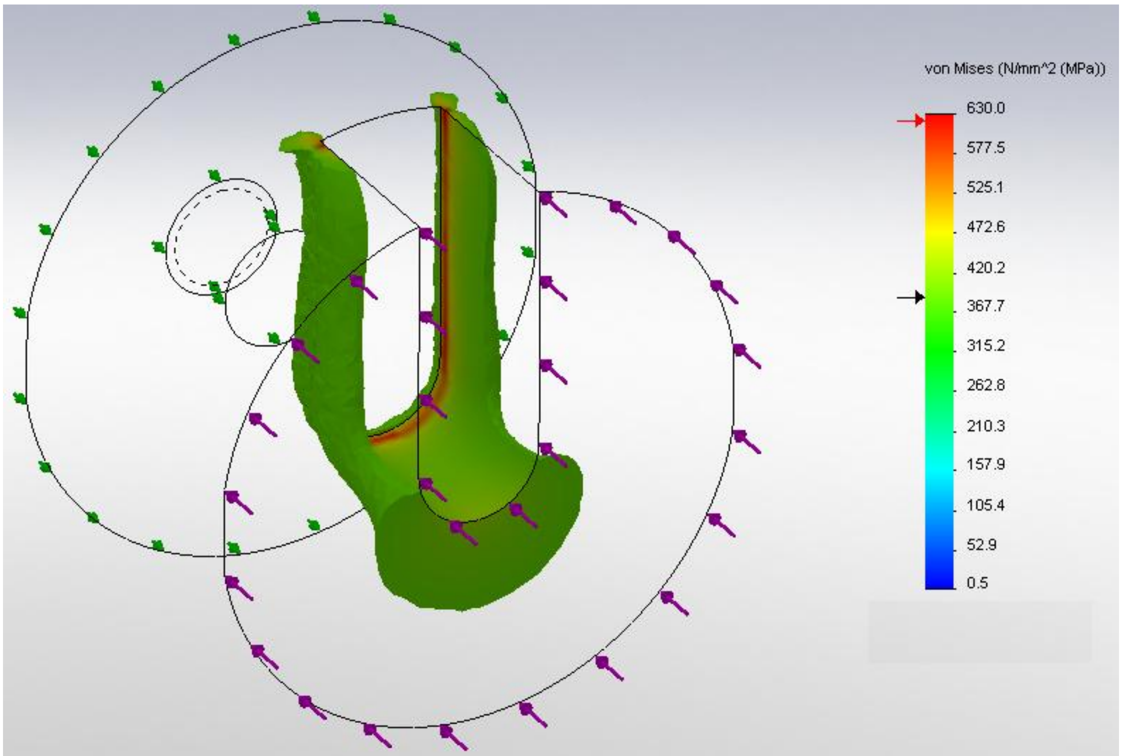


Picture 91: Overview of simulation results on interface



Picture 92: Overview of simulation results on interface

By using ISO-clipping, see Picture 93, the volume that exceeds the allowable stress limits of 375MPa easily can be shown.



Picture 93: Volume of interface that exceeds the allowable stress limit

Picture 93 shows that some volume of the model is over the allowable stress limit. But the stress that exceeds the limit do not go through the whole part, and there is no points on the model that exceeds the yield strength limit of 650MPa → The interface is OK dimensioned.

4.3.7 Bolt calculation

The bolt calculation is done according to NS-EN 13445-3:2009. The friction coefficient data, see Table 16 and Table 17, is collected from (Markserv 2012).

Friction coefficient data threads					
		External threads			
		Self finish	Zinc plated	Cast iron	Aluminium
Internal threads	Steel	0,10-0,16	0,12-0,18	0,10-0,16	0,10-0,20
	Dry	0,08-0,16	0,10-0,18	0,08-0,18	0,10-0,18
	Self finish or phosphate treated oiled	0,12-0,20	0,12-0,22	0,10-0,17	0,12-0,20
	Zinc dry	0,10-0,18	0,10-0,18	0,10-0,16	0,10-0,18
	Plated oiled	0,18-0,24	0,18-0,24	0,18-0,24	0,18-0,24
	Thread adhesive	-	-	-	-

Table 16: Friction coefficient data on threads (Markserv 2012)

Friction coefficient data underhead					
		Bolt head or nut			
		Zinc plated	Self finish	Cast iron	Aluminium
Flange or plate	Zinc plated dry	0,16-0,22	0,12-0,20	0,10-0,20	-
	Finish slight oil applied	0,10-0,18	0,10-0,18	0,10-0,18	-
	Self finish or dry	0,10-0,18	0,10-0,18	0,08-0,16	-
	Phosphate or black oxide finish slight oil applied	0,10-0,18	0,10-0,18	0,12-0,20	0,08-0,20

Table 17: Friction coefficient data on bolt head (Markserv 2012)

The typical K-values, see Table 18 is collected from(Loctite 2012). However the K-value used in the bolt calculations is calculated following the method described by (Euler, Bolt Preload Calculation 2002).

Typical "K" values				
	Lightly oiled	Lightly oiled + Loctite 243	Degreased	Degreased + Loctite 243
Steel fastener	0,15	0,14	0,2	0,2
Phosphated steel	0,13	0,11	0,24	0,14
Cadium plated steel	0,14	0,13	-	-
Stainless steel 404	0,22	0,17	-	-
Zinc plated steel	0,18	0,16	-	0,15

Table 18: Typical K-values (Loctite 2012)

The results from the bolt hand calculations is shown in Table 19. The key results from the calculations to note is:

- Amount of bolts: 16
- Dimensions: M36x2
- Bolt grading: 10.9
- Utilization of the bolts: 99%
- Minimum length of threaded holes: 41,65mm

Pressure vessel inndata				
Description	Notation	Unit	Value	Note
Internal depth of chamber	Dc	mm	440	
Internal width chamber	Wc	mm	250	
Distance between gasket grooves outer edges		mm	346,15	
Distance from outer edge to prssure subjected area		mm	100	Subtrackts from the internal depth
Wanted packer pressure	Ppacker wall	N/mm^2	53,21	
Yield strength pressure vessel walls		N/mm^2	650	
Bolt inndata				
Description	Notation	Unit	Value	Note
Number of bolts	Nbolts	Quantity	16	
Bolt grading		x.x.	10.9.	
Yield strength bolt	σybolt	N/mm^2	940	
Tensile strength bolt	σubolt	N/mm^2	1040	
Bolt diameter	D	mm	36	
Thread pitch (bolt longitudinal distance per thread)	p	mm	2	
Thread profile angle (60 degrees for M,MJ,UN,UNR,UNJ)	αth	degrees	60	
Thread coefficient of friction	myt		0,15	
Collar coefficient of friction	myc		0,15	
Tensile stress area	As	mm^2	914,53	To be read out of table
Output bolted conection				
Description	Notation	Unit	Value	Note
Minimum engagement length of screws in threaded holes		mm	41,65	NS-EN 13445-3:2009 chapter 11.4.3.3
Allowable design stress level	f	N/mm^2	433,3	NS-EN 13445-3:2009 chapter 6.2
Total tensile force the bolts will see	Ftensile all	kN	6262,3	
Tensile force each bolt will see	Ftensile	kN	391,40	
Tensile stress each bolt will see (Without pretension)		N/mm^2	427,98	
Tensile capacity of each bolt (including material safety factor)	Fd,t	kN	396,30	
Tensile capacity of all bolts (including material safety factor)		kN	6340,7	
Bolt utilization	Um Bolt		0,99	
Torque coefficient	K		0,186	http://euler9.tripod.com/fasteners/preload.html
Maximum pretension force per bolt		kN	601,76	NS-EN 1090-2: 2008
Maximum pretension force all bolts		kN	9628,2	
Maximum bolt installation torque	T	Nm	4030,9	NS-EN 1090-2: 2008

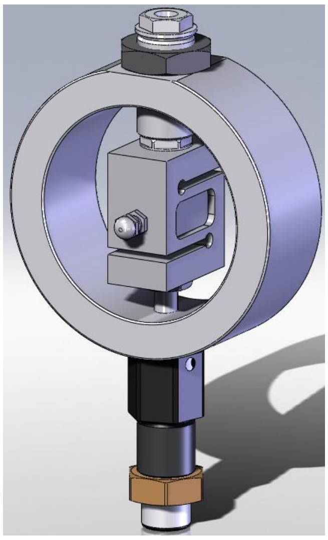
Table 19: Bolt hand calculation

4.4 Test result recordings

4.4.1 Qualitative results

The parameters the qualitative results can be recorded from is:

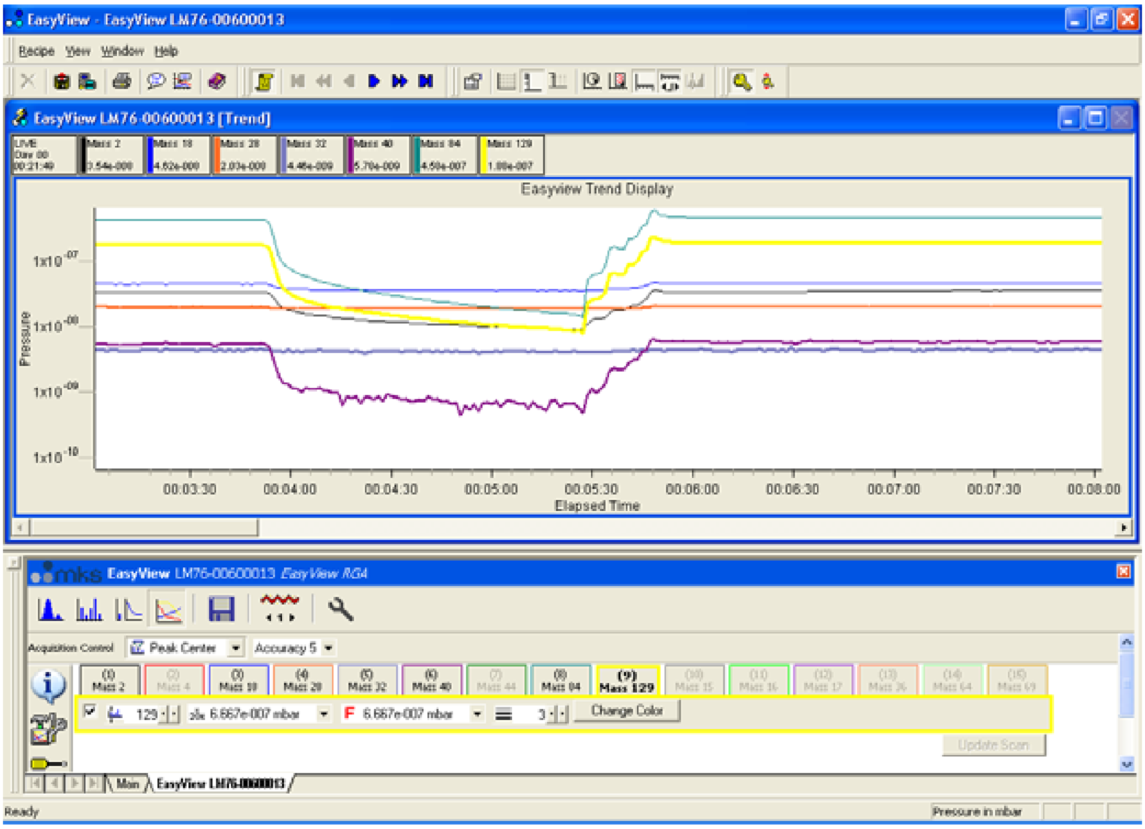
- Hydraulic cylinder pressure is recorded by:
  - Visual manometer
  - Pressure transmitter connected to Easyview, see Picture 95.
- Annulus water pressure is recorded by:
  - Visual manometer
  - Pressure transmitter connected to Easyview, see Picture 95.
- Temperature is recorded by:
  - Temperature transmitter connected to Easyview, see Picture 95.
- Packer pressure is recorded by:
  - Load cell, see Picture 94, placed in center of packer coupon in set position, the load cell is connected to Easyview, see Picture 95.



Picture 94: Packer pressure load cell

  - Alternatively the packer pressure can be recorded by an piezoelectric film put in between the test rig wall and the packer coupon, see Appendix L: I-Scan piezoelectric pressure recording. This film would give a more accurate pressure reading, and have the opportunity to tell the pressure distributions over the whole packer surface. This can give a better understanding on how the packer and anti extrusion springs behaves under pressure, and also how it behaves in a failure mode. The piezoelectric film can also be put on other surfaces of the packer in a test to record other unknown pressure distributions.

    Since this packer pressure recording method is relatively expensive, see Appendix M: Price offer on I-Scan system from CA Mätsystem AB, it is decided to produce the prototype of the test rig with the load cell, see Picture 94. The piezoelectric system can in a later stage be applied with no or minor adjustments to the rig.



Picture 95: Example of EasyView RGA Software platform (MKS Instruments UK Ltd 2010)

The recording methods for the relevant results is presented in Table 20.

Result	***Recording method***
Measured force needed to deform the packer from unset to set position	• Recorded by hydraulic cylinder pressure
The packer pressure which creates the stress in the pipe wall	• Recorded by load cell on top of test rig • Alternatively by piezoelectric film between wall and packer
Stress in the plug module	• Can be recorded by piezoelectric film between bowl/pressure head and packer coupon
The maximum differential pressure the packer can withstand before it leaks or total failure	• Recording hydraulic and water pressure drop • Visual recording by camera during test
Maximum pipe diameter the packer can operate in i.e. the biggest possible OD of the packer in set position without failure of anti extrusion springs	• Trial and error recordings • Visual recording by camera during test • When failure occur → pressure drop in hydraulic cylinder and annulus water
The minimum differential pressure the packer can withstand before it leaks (also called; “self lock pressure”)	• Visual inspection by camera during test • Pressure drop in annulus water • NB! This parameter can be hard to record, because of the design of the rig.

Table 20: Qualitative test result recordings

4.4.2 Quantitative results

The parameters the quantitative results can be recorded from is:

- Visual inspection after test is recorded by a snapshot camera and explanatory text
- Visual inspection during test is recorded by a film camera

The recording methods for the relevant results is presented in Table 21

Result	***Recording method***
Total packer failure i.e. the pressure in the rubber is too big for the anti extrusion springs to hold it→rubber squeezes out of the initially closed area	• Visual recording by camera during test • Recording of loss of water pressure in annulus area • Recording of loss of hydraulic cylinder pressure • Visual inspection after test, with coupon out of test rig
The rubber has cracks	• Visual inspection after test, with coupon out of test rig • Visual recording by camera during test
The springs have plastic deformation	• Visual inspection after test, with coupon out of test rig • Visual recording by camera during test • Packer fails→recording by loss of hydraulic and water pressure
The packer starts to leak	• Recording by loss of water pressure in annulus area • Visual recording by camera during test
The packer has folds and an unnatural shape after setting	• Recording of unnatural packer pressure • Visual inspection after test, with coupon out of test rig • Packer leaks→ Visual recording by camera and recording of water pressure
Spring behavior	• Visual inspection after test, with coupon out of test rig • Visual recording by camera during test
Rubber behavior	• Visual inspection after test, with coupon out of test rig • Visual recording by camera during test
What fluid properties it can isolate against	• Material properties check • Visual inspection on failure after test, with coupon out of test rig

Table 21: Quantitative test result recordings

4.5 Economics

The cost of the packer test rig is divided into two groups as shown in Table 22:

Fixed costs:	**Size specific costs:**
Hydraulic cylinder	Main bolts
Top plate	U-chamber
Bottom plate	Bowl
Support frame	Pressure head
Sensors	Packer coupon

Table 22: Cost groups

The fixed costs is a one-time expense that accrue when the test rig is made. The fixed cost parts are reused parts that are going to be used independent of the:

- Geometry
- Size
- Rubber configuration
- Spring configuration
- Test pressure

The size specific costs are costs related to parts that have to be made for the specific packer geometry and size. These parts are also reusable parts, but only for a specific geometry and size. The packer coupon listed under the size specific costs, see Table 22, is not a 100% reusable part, this depends on the test. If the coupon is going to be tested to failure point, it will break and cannot be used again, if not it may be used multiple times before it needs to be changed.

The price estimates comes from these suppliers:

- Maskinering & Sveiseservice AS – All the steel parts. See Appendix H: SolidWorks drawings
- Hytorc AS – Powerteam hydraulic cylinders. See Appendix J: Cylinder price offers
- K. Lund Offshore AS – Larzep hydraulic cylinders. See Appendix J: Cylinder price offers
- Rubberstyle AS – Packer coupon. See Appendix K: Packer coupon price offer

Estimates were collected early in the conceptual design phase. Because of this the parts are dimensioned to a 200mm wide coupon size. Since it was decided to widen the coupon size to 250mm the parts have to be dimensioned for the added stresses due to this modification. If we assume a linear stress rise and a linear price estimate for the components, we can estimate the cost of the parts after the modification. The price estimates would rise according to Eq. 58:

$$200 + \frac{200 * x}{100} = 250 \rightarrow x = 0,25 = 25\% \qquad \text{(Eq. 58)}$$

The price estimates for the fixed components can be seen in Table 23. Important factors to note are:

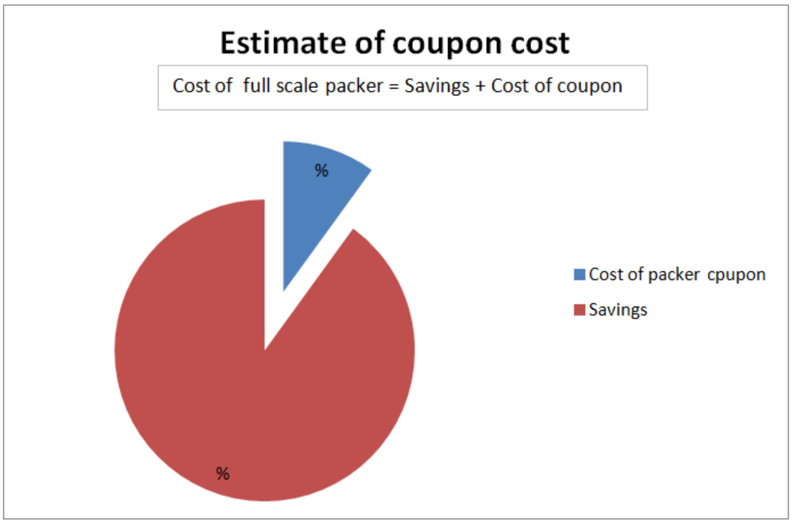
- The price for the hydraulic cylinder is fixed and will not increase linearly with the increase of the width of the packer coupon.
- Lifting eyes are workshop material, and will not be considered in this thesis
- Pressure censors are workshop material and a load cell to measure packer pressure exists in the TDW system and will therefore not be considered in this thesis.
- The hydraulic cylinder is 50% of the total “fixed components” price.

Fixed components		
Component	***Estimated price 200mm width***	***Estimated price 250mm width***
Larzep 350mT 200mm stroke	56228	56228
Support frame	10500	13125
Top plate	15500	19375
Bottom plate	16900	21125
Interface hydr. cylinder	3500	4375
Lifting eyes	-	-
Censors	-	-
Sum	**102628**	**114228**

Table 23: Price estimates for fixed components

The price estimates for the size dependent components can be seen in Table 24. Important factors to note is:

- The packer coupon estimate. The estimate from Rubberstyle for producing the packer coupons is:
  - Mold: 30 000 NOK
  - Rubber: 100 NOK per kg
  - Production: Casted on hourly basis;
    - 800 NOK per hour for press
    - 600 NOK per man-hour
    - two press hours per man hour
  - The estimates from Rubberstyle does not provide a good overview of the packer coupon cost. A better way to visualize the estimate of the possible savings earned by using a coupon instead of a full scale packer is simply by assuming that producing a certain percentage out of the full-scale packer, will cost that percentage of the cost of the full scale packer that we know a more accurate cost estimate on, see Picture 96.



Picture 96: Packer coupon cost estimate

- Price estimates for the main bolts and seals are not collected.

Size dependent components						
Component	***Estimated price 200mm width***			***Estimated price 250mm width***		
	48in	**24in**	**8in**	**48in**	**24in**	**8in**
U-chamber	22200	6200	4500	27750	7750	5625
Pressure head	4900	6500	5500	6125	8125	6875
Piston (Bowl)	5900	4800	4500	7375	6000	5625
Main bolts	-	-	-	-	-	-
Seals	-	-	-	-	-	-
Packer coupon	See other estimate			See other estimate		
Sum	**33000**	**17500**	**14500**	**41250**	**21875**	**18125**

Table 24: Price estimates for size dependent components

Chapter 5: Discussion

5.1 Final design

The drawings on the final design of the packer test rig can be seen in Appendix H: SolidWorks drawings. The overall dimensions of the test rig prepared for a 48 inch test are shown in Picture 97.

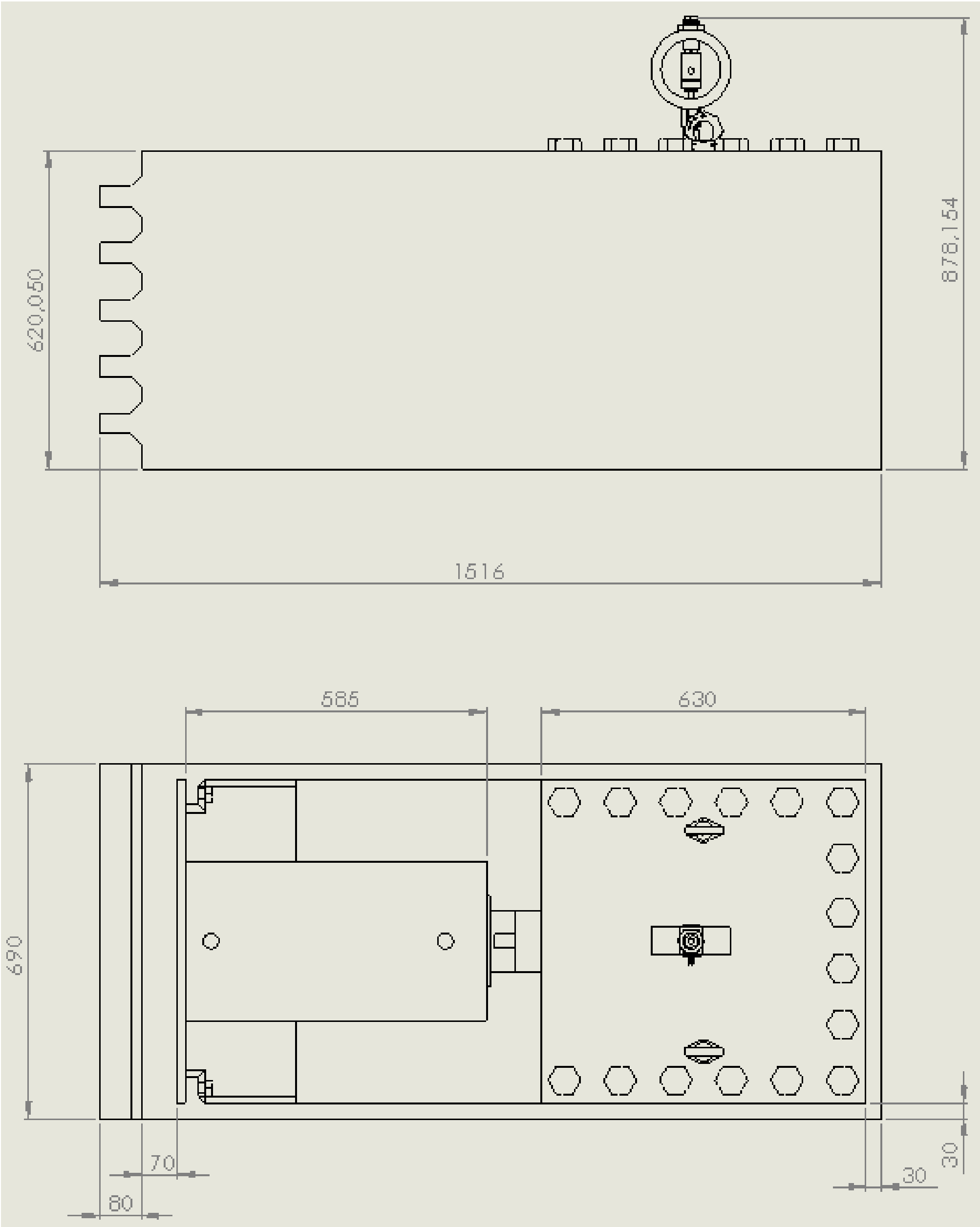
The limitations on the packer tester are relative to packer pressure, coupon size and geometries, but the absolute limitations will be:

- Maximum force available: 350T
- Maximum packer pressure that can be simulated on the 48 inch packer coupon: 532,1 bar.
- Maximum internal chamber size with maximum internal pressure (eqvivalent to maximum packer pressure) of 532,1bar:
    - Width: 250mm
    - Height: 258,1mm
    - Depth: 440mm
- Maximum pipeline pressure to be simulated on the 48 inch packer coupon with the available force: 357,5bar
- Test rig is flexible to packer coupon sizes with normal geometries between 8inch and 48inch (this is relative to the geometries of the packer cross section).
- The simulations in this thesis are done on the biggest packer cross section in the TDW system. On another smaller packer coupon the simulated packer pressure and eqvivalend pipeline pressure could be much higher. Because of this, a simulations on the specific configuration of the test rig should be done before the test is conducted.

A mass estimate, see Table 25, of the packer test rig is done with mass information from SolidWorks.

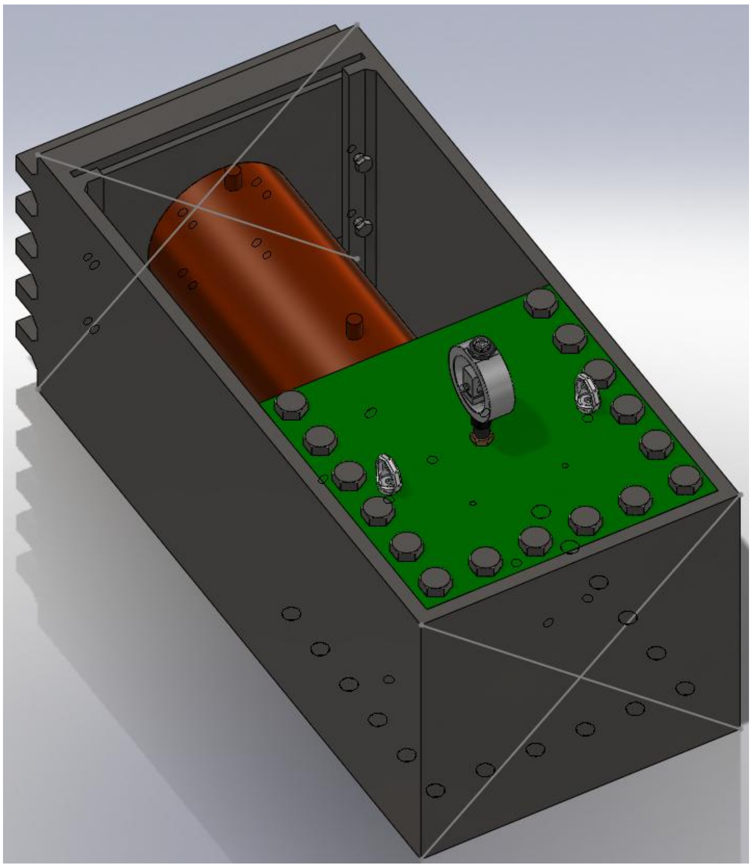
Mass calculation on packer test rig prepared for 48 inch test			
Part	Amount	Weight per piece [kg]	Total [kg]
U-chamber	1	260,3	260,3
Top plate	1	430,4	430,4
Bottom plate	1	435,4	435,4
Support plate hydr.cyl	1	30,2	30,2
Support frame	1	881,2	881,2
Hydraulic cylinder	1	195	195
Interface	1	3,8	3,8
Countersunk M20x2 L40	4	0,16	0,62
M36x2 L540	16	4,69	74,98
M20x2 L40	4	0,17	0,69
Packer coupon	1	12,42	12,42
Total			**2325,01**

Table 25: Mass calculation on packer test rig

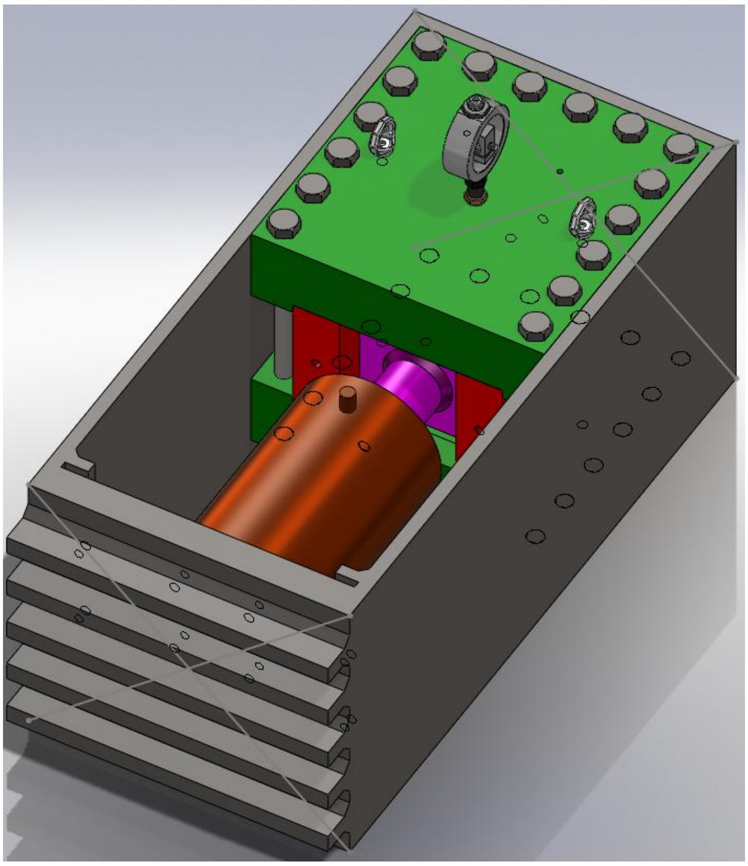


Picture 97: Overall dimensions of test rig prepared for 48 inch test

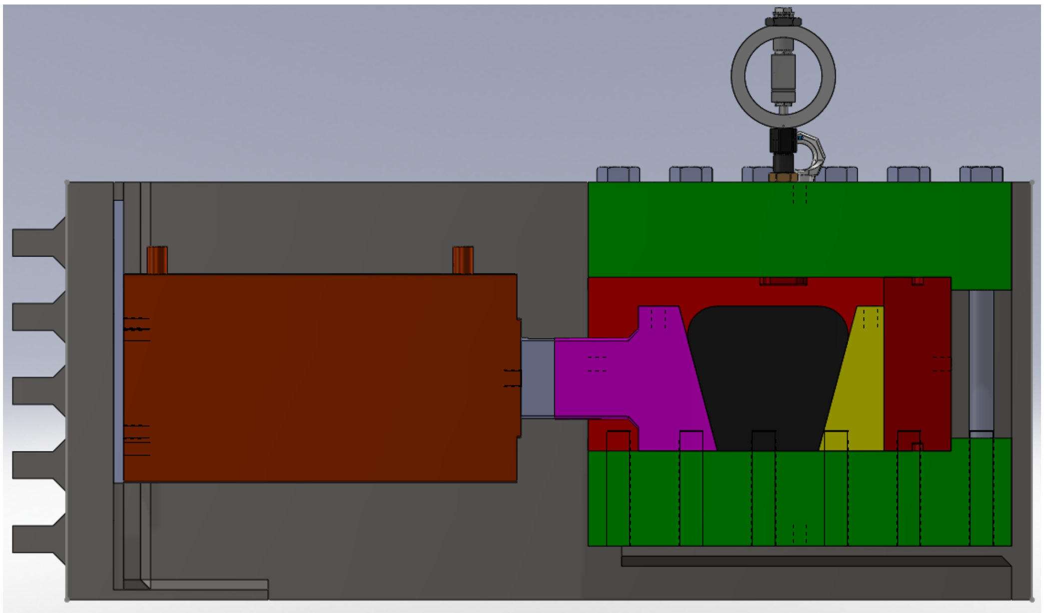
Picture 97, Picture 98 and Picture 99 give an overview of the rig. The cross sectional view seen in Picture 100 shows the rig configured for the 48 inch packer coupon. In the configuration shown in Picture 100 and Picture 101 the coupon is in the unset position. In Picture 101 the top plate is removed from the pressure chamber and the seal groves in the U-chamber (red part) shows. On one point the seal groves are in contact with packer coupon, the reason for this is to seal against the annulus water pressure. The load cell measure the packer pressure in the middle of the coupon when it is in set position.



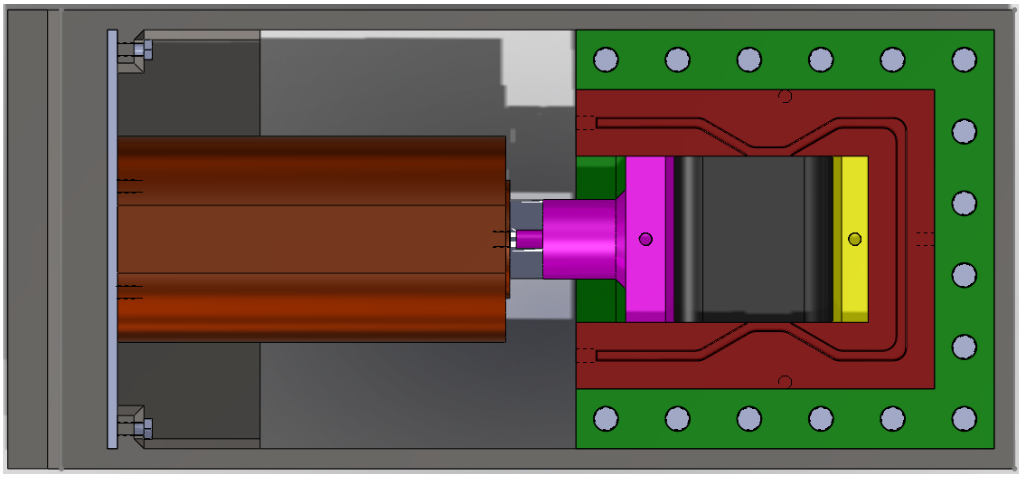
Picture 98: Final design of packer test rig



Picture 99: Final design of packer test rig



Picture 100: Cross section of the final design of the packer test rig



Picture 101: Overview of the final design of the packer test rig with top plate removed

5.2 Discussion and Conclusion

The background for the thesis was the economical issue with the existing packer development method TDW used. The thesis scope of work was:

- To address the problems and challenges with TDW's existing packer test methods.
- To look into a totally new concept for testing the packers and finding its attributes.
- To develop the new testing concept.

The benefits of the proposed testing system are:

- An opportunity for a higher test frequency.
- A faster and easier execution for the workshop personnel.
- A more economical development method .

- Only one packer test rig that covers all the packer dimensions and geometries with little configurations. Which saves storage and workshop space.
- An inexpensive to produce and operate test rig.
- A test rig that gives consistent and accurate results.
- A test system that limits the loss when a packer fails or does not fulfill the requirements.
- A test rig that can test packers with 30% expansion, and is robust enough to test a packer to failure (Very high pressure; 400-600 bar packer pressure).

To address the problems and challenges with today's packer development a new packer test method has been developed to minimize the losses when a packer does not fulfill the desired requirements. The biggest economical losses derive from the expensive production of the test packer. To prevent this economical loss the new method, tests only a small segment out of a full-scale packer. The segment would be only a fraction of the size and cost of a full-scale packer, resulting in less economical loss upon failure.

The desired results from the new packer test method are:

- Qualitative results:
    - Spring configurations
    - Force needed to deform the packer from unset to set position
    - The packer pressure
    - Stress transferred to pipe wall from packer
    - Stress transferred to the plug module
    - The maximum differential pressure the packer can withstand before it leaks or total failure
    - What fluid properties it can isolate against
    - Maximum pipe diameter the packer can operate in i.e. the biggest possible OD of the packer in set position without failure of anti extrusion springs
    - The minimum differential pressure the packer can withstand before it leaks (also called; "self lock pressure")
- Quantitative results:
    - Total packer failure i.e. the pressure in the rubber is too big for the anti extrusion springs to hold it, therefore the rubber squeezes out of the initially closed area
    - The rubber has cracks
    - The springs have plastic deformation
    - The packer starts to leak
    - The packer has folds and an unnatural shape after setting
    - Spring behavior
    - Rubber behavior

All the desirable results from the developed test rig can be collected and recorded apart from the self lock pressure. Due to the design of the test rig, results on self lock pressure that corresponds to a real life situation seems unlikely to be able to collect.

The results from the new packer test method have some expected errors, it is believed that the errors can be estimated by comparing test results from the test rig to earlier recorded full scale data.

The expected errors are a result of these differences from a full scale situation:

- The change in volume
- The tensile force from the anti extrusion springs
- The hoop stress from rubber
- The curved surface
- The force calculation that have to be done to find the hydraulic force that is equivalent to a real life situation pipeline pressure.

Although testing in the new test rig is not equivalent to testing a full scale packer, by using the new method, many packer geometries and configurations inexpensively can be tested to find the one that perform best, before the full-scale packer is made and tested. This removes a lot of uncertainties when the full-scale packer is made.

The design of the packer test rig, increasing the probability of success in full scale testing, and have the opportunity for applying a mechanism that simulate the differences from a full scale test a later date.

Early in the master thesis program it was planned to produce the new packer test rig while the work on the master thesis was carried out in the spring. When the rig was produced, tests could be performed and the results could be presented in the report. Unfortunately there was not enough time to achieve this.

To validate if the test method developed in this thesis fulfill all the desirable goals, tests with the test rig have to be run. Assumptions indicate that the packer test method can deal with the problems and fulfill all the goals. The goal about having a test rig that give consistent and accurate results can only be proven through testing. The test method is expected to give consistent and accurate results, but not results equal to full scale results, these errors are expected and can be compensated for.

5.4 Further work

Further work that has to be conducted on the packer test rig:

- Make production drawings.
- Produce the rig for a packer geometry with good data from previous full scale testing.
- Run tests on a packer coupon with same geometries as the geometry on the full-scale packer which the earlier data comes from.
- Compare results from test rig to the results from the full scale situation.
- Estimate errors.
- Make a calculation model for calculating the hydraulic force that corresponds to the full scale pipeline pressure with the estimated errors compensated for.
- Make a procedure on how to use the calculation model and the test rig.
- Validate if the test rig performs as planned .
- If the test rig performs as planned, start planning a mechanism to take care of the:
    - The change in volume
    - The hoop stress from springs
    - The hoop stress from rubber

References

Azom. "Engineering Seals - Sealing of Static Joints." *Azom.* 05 15, 2012. http://www.azom.com/article.aspx?ArticleID=503 (accessed 05 15, 2012).

—. "Rubbers and Elastomers - An Introduction." *Azom.* 05 15, 2012. http://www.azom.com/article.aspx?ArticleID=317 (accessed 05 15, 2012).

Brydson, J. A. *Rubbery Materials and Their Compounds.* Elsevier Science Publishers Ltd., 1988.

Ciesielski, Andrew. *An Introduction to Rubber Technology.* Shawnbury, Shrewsbury, Shropshire, UK: Rapra Technology, 1999.

Euler, Garrett D. *Bolt Preload Calculation.* 01 01, 2002. http://euler9.tripod.com/fasteners/preload.html (accessed 06 08, 2012).

—. *Standard bolt dimensions.* 09 10, 2002. http://euler9.tripod.com/bolt-database/23.html (accessed 06 11, 2012).

Gent, Alan N. *Engineering with Rubber.* München: Hanser Verlag, 2001.

International Institute of Synthetic Rubber Producers, Inc. "ACRYLONITRILE-BUTADIENE RUBBER (NBR)." *International Institute of Synthetic Rubber Producers, Inc.* 05 15, 2012. http://www.iisrp.com/WebPolymers/07NBR-18Feb2002.pdf (accessed 05 15, 2012).

Jostein Aleksandersen & Edd Tveit. "The SmartPlug: A Remotely Controlled Pipeline Isolation System." *e-book.lib.* 06 17, 2001. http://e-book.lib.sjtu.edu.cn/isope2001/pdffiles/papers/152.pdf (accessed 05 15, 2012).

K. Lund Offshore. *K. Lund Offshore.* 06 05, 2012. http://www.kl-offshore.no/no/products/larzep-hydrauliske-verkty/#country=no&store=sale&cat=6&anc= (accessed 06 05, 2012).

Leeco steel. *S355 European Standard Steel.* 06 09, 2012. http://www.leecosteel.com/products/en-standards-steel/s355.html (accessed 06 09, 2012).

Loctite. *The forgotten factor.* 06 08, 2012. http://www.loctite.com.au/cps/rde/xchg/henkel_aue/hs.xsl/3089_3280_AUS_HTML.htm (accessed 06 08, 2012).

Markserv. *Markserv.* 06 08, 2012. http://www.markserv.co.th/home.php (accessed 06 08, 2012).

Masteel. *Masteel.* 05 23, 2012. http://www.masteel.co.uk/s690-ql.htm (accessed 05 23, 2012).

MKS Instruments UK Ltd. "Easy View User Manual." *MKS technology for productivity.* 01 01, 2010. http://mmrc.caltech.edu/Kratos%20XPS/MKS%20RGA/Easy%20View%20V5.7%20User%20Manual%20-%20SP104022.101.pdf (accessed 06 12, 2012).

Oilinfo. *Oilinfo.* 05 15, 2012. http://www.google.no/imgres?q=tdw+smartplug&start=96&hl=nn&tbm=isch&tbnid=KcH4hX0SZmTLMM:&imgrefurl=http://www.oilinfo.no/index.cfm%3Fevent%3DdoLink%26famId%3D38860&docid=k

KcWCQLA3P1ZkM&itg=1&imgurl=http://www.oilinfo.no/_files/TDW_North%252520Sea%252520t (accessed 05 15, 2012).

Piping Guide. *Piping guide.* 05 15, 2012. http://www.pipingguide.net/2008/11/introduction-to-pipeline-pigging-2.html (accessed 05 15, 2012).

R. K. Flitney, B. S. Nau, D. Reddy. *The Seal Users Handbook.* California: BHRA The fluid engineering centre, 1984.

Simrit. "THE PHYSICS OF RUBBER." *Simrit.* 05 15, 2012. http://www.simritna.com/catalog/o-ring/98physics_rubber.htm (accessed 05 15, 2012).

SkyTruth's Photostream. *flickr from yahoo.* 06 29, 2009. http://www.flickr.com/photos/skytruth/3768390779 (accessed 05 15, 2012).

SolidWorks. "Backround on meshing." *SolidWorks Help.* 05 24, 2012. http://help.solidworks.com/2011/English/SolidWorks/cworks/LegacyHelp/Simulation/Meshing_topics/step5_mesh_the_part.html?id=b340d9fa1ab64eb389656f8145045e36#Pg0 (accessed 05 24, 2012).

—. "Basic consepts of analysis." *SolidWorks Help.* 05 24, 2012. http://help.solidworks.com/2011/English/SolidWorks/cworks/LegacyHelp/Simulation/Fundamentals/Basic_Concepts_of_Analysis.htm?id=23b9c5d061ed4309b16548da1cc72781#Pg0 (accessed 05 24, 2012).

—. "Defenition of basic quantities." *SolidWorks Help.* 05 24, 2012. http://help.solidworks.com/2011/English/SolidWorks/cworks/LegacyHelp/Simulation/AnalysisBackground/StaticAnalysis/IDH_Analysis_Background_What_is_Stress.html?id=7509e3f67f754e55bddbf6db67ad7ec8#Pg0 (accessed 05 24, 2012).

—. "Linear Static Analysis." *SolidWorks Help.* 05 24, 2012. http://help.solidworks.com/2011/English/SolidWorks/cworks/LegacyHelp/Simulation/AnalysisBackground/StaticAnalysis/IDH_Analysis_Background_Linear_Static_Analysis.html?id=b8d858e16d3247b5a28bcb49140fa162#Pg0 (accessed 05 24, 2012).

—. "Solid Mesh." *SolidWorks Help.* 05 24, 2012. http://help.solidworks.com/2011/English/SolidWorks/cworks/LegacyHelp/Simulation/Meshing_topics/ID_Solid_Mesh.html?id=563b04e5012944d3970adc8d8c298d3c#Pg0 (accessed 05 24, 2012).

Stevenson, A. *Rubber in Offshore Engineering.* Bristol, UK: Taylor & Francis; 1 edition, 1984.

Strømsmo, Per Olav. *Design analysis of packer.* Master Thesis, Stavanger: Høgskolen i Stavanger, 2004.

Strømsmo, Per Olav. *Design Analysis of Packer.* Master Thesis, Stavanger: Stavanger University College, 2004.

T.D Williamson. *T.D Williamson homepage, Pipeline Services.* 11 18, 2011. http://www.tdwilliamson.com/en/Services/Pages/Home.aspx (accessed 11 18, 2011).

William D. Callister, Jr. *Materials Science and Engineering.* John Wiley Sons, Inc, 2007.

Appendices

Appendix A: Conceptual design review meeting summary

Meeting summary

Meeting information	
Date	23.03.2012
Prepared	Sven Tore Jakobsen
Facilitator	Jeff Wilson
Place	Conf room, STA SmartTrack - 2nd floor

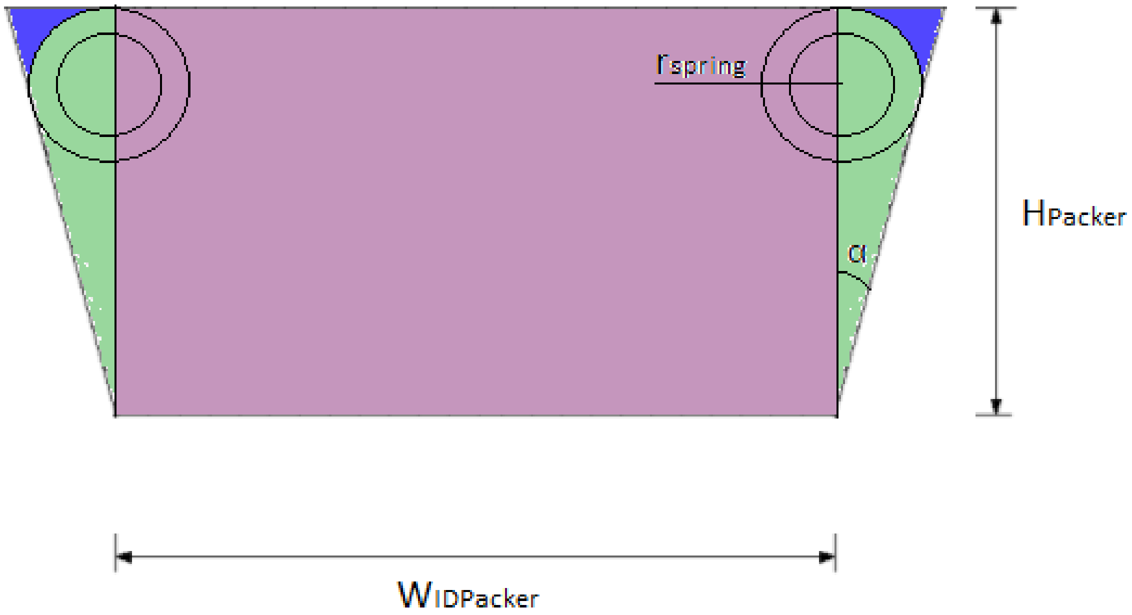
Members	
Members invited	**Members attending**
Sven Tore Jakobsen	x
Jeff Wilson	x
Vigmund Bjørsvik	x
Harald Wittersø	x
Erik Herredsvela	x
Robert Hendrics	x
Henning Bø	

Summary	The purpose of the meeting was to present Sven Tore's master thesis concerning the design of the new packer test rig. And after the presentation there was planned a brainstorm with questions and suggestions to improvement.
Decisions and agreements	The initial first draft of the rig, does not take into account the volume change and hoop stress in the rubber, and the strain in the springs that occurs in a real life situation. The main concern from the other engineers was how to deal with this hoop stress and strain. After a discussion round there was an agreement on that the initial simplifyed idea for the test rig with a rektangular packer specimen without curvature would give the results that we are looking for in this first round, and allso that Sven Tore's time schedule did not have room for making a mekanisme that take the hoop stress and strain in the packer in to account. Further it was decided that only small edits to the initial design would make an opening for this mekanisme to be made on a later stage, and fittet to the allready existing rig.

Actions to be taken	By whoom?	When?
Make production drawings with revisions	Sven Tore	27.04.2012
Master thesis report writing	Sven Tore	15.06.2012
Do tests with the test rig (if there is time)	Sven Tore	15.06.2012

Appendix B: Accurate Volume Calculation of packer

Area cross section Packer:



- ● + ● = Area of packer cross section
- ● = Area of rectangular
- ● + ● = Area of triangular
- ● = Area of corner

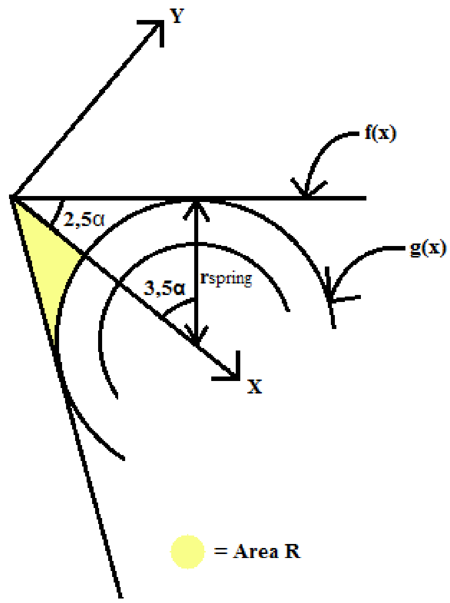
Area of packer cross section:

$$A_{Packer} = A_{Rectangular} + (2 * A_{Triangular}) - (2 * A_{Corner}) \qquad \text{(Eq.B 1)}$$

$$A_{Rectangular} = W_{IDPacker} * H_{Packer} \qquad \text{(Eq.B 2)}$$

$$2 * A_{Triangular} = {H_{Packer}}^2 * Tan(\alpha) \qquad \text{(Eq.B 3)}$$

Area of cross section corners:



$$A = \iint_R dA \qquad \text{(Eq.B 4)}$$

$$A_R = \int_{y_0}^{y_1}\int_{x_0}^{x_1} dxdy \qquad \text{(Eq.B 5)}$$

Equation for f(x):

$$f(x) = tang(2{,}5\alpha) * x \qquad \text{(Eq.B 6)}$$

Distance from origo to center of spring:

$$x_{center} = \frac{r_{spring} * \sin(3{,}5\alpha)}{\tan(2{,}5\alpha)} + r_{spring} * \cos(3{,}5\alpha) \qquad \text{(Eq.B 7)}$$

$$x_{center} = r_{spring}\left(\frac{\sin(3{,}5\alpha)}{\tan(2{,}5\alpha)} + \cos(3{,}5\alpha)\right) \qquad \text{(Eq.B 8)}$$

Equation for the circle:

$$(x - x_0)^2 + (y - y_0)^2 = r^2 \qquad \text{(Eq.B 9)}$$

$$\left(x - r_{spring}\left(\frac{\sin(3{,}5\alpha)}{\tan(2{,}5\alpha)} + \cos(3{,}5\alpha)\right)\right)^2 + y^2 = r_{spring}^2 \qquad \text{(Eq.B 10)}$$

Integration limits in x direction is found by solving equation (A3) and (A5) with respect to x:

$$x_0 = \frac{y}{\tan(2{,}5\alpha)} \qquad \text{(Eq.B 11)}$$

$$x_1 = -\sqrt{r_{spring}^2 - y^2} + r_{spring} * \left(\frac{\sin(3{,}5\alpha)}{\tan(2{,}5\alpha)} + \cos(3{,}5\alpha)\right) \qquad \text{(Eq.B 12)}$$

Integration limits in y direction:

$$y_0 = 0 \qquad \text{(Eq.B 13)}$$

$$y_1 = r_{spring} * \sin(3{,}5\alpha) \qquad \text{(Eq.B 14)}$$

Put eq.11.12.13.14 into eq.4 and get area of half corner cross section area R (yellow area) ref fig:

$$A_R = \int_0^{r_{spring}*\sin(3{,}5\alpha)} \int_{\frac{y}{\tan(2{,}5\alpha)}}^{-\sqrt{r_{spring}^2 - y^2} + r_{spring}*\left(\frac{\sin(3{,}5\alpha)}{\tan(2{,}5\alpha)} + \cos(3{,}5\alpha)\right)} dxdy \qquad \text{(Eq.B 15)}$$

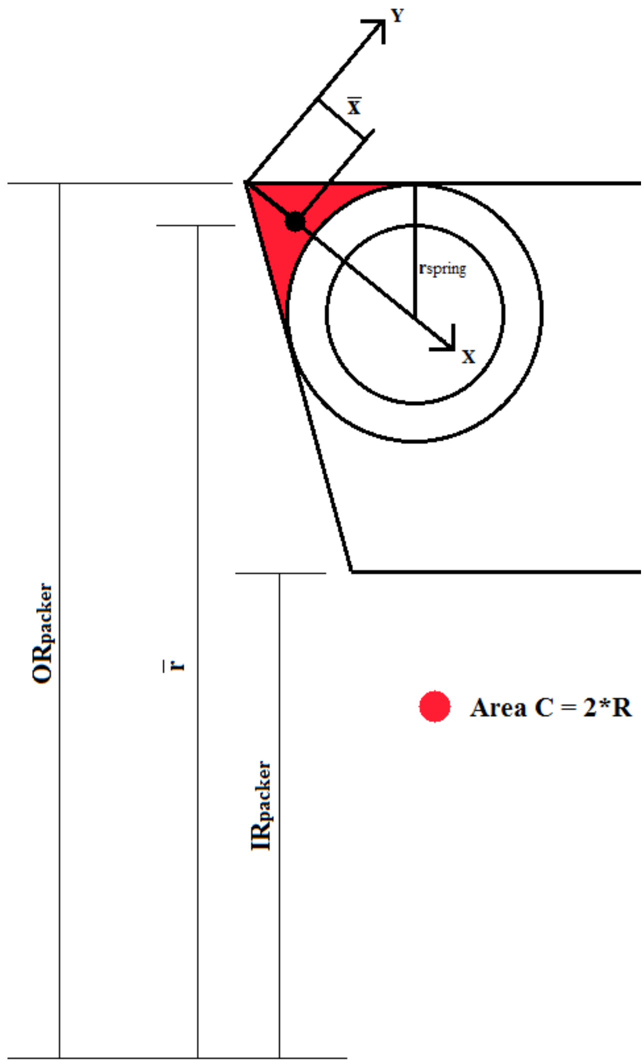
Solving the integral eq.15 and find the area of half the corner:

Volume of corners:

Pappu's revolving theorem:

$$V = 2\pi\bar{r}A \qquad \text{(Eq.B 16)}$$

$$\bar{r} = distance\ from\ revolving\ axis\ to\ area\ of\ centre$$



Distance r̄ ref pic:

$$\bar{r} = OR_{packer} - \bar{x} * \sin(2{,}5\alpha) \qquad \text{(Eq.B 17)}$$

Finding distance x̄:

$$\bar{x} = \frac{1}{A_C}\iint_C x dA = \frac{1}{A_C}\iint_C xdxdy \qquad \text{(Eq.B 18)}$$

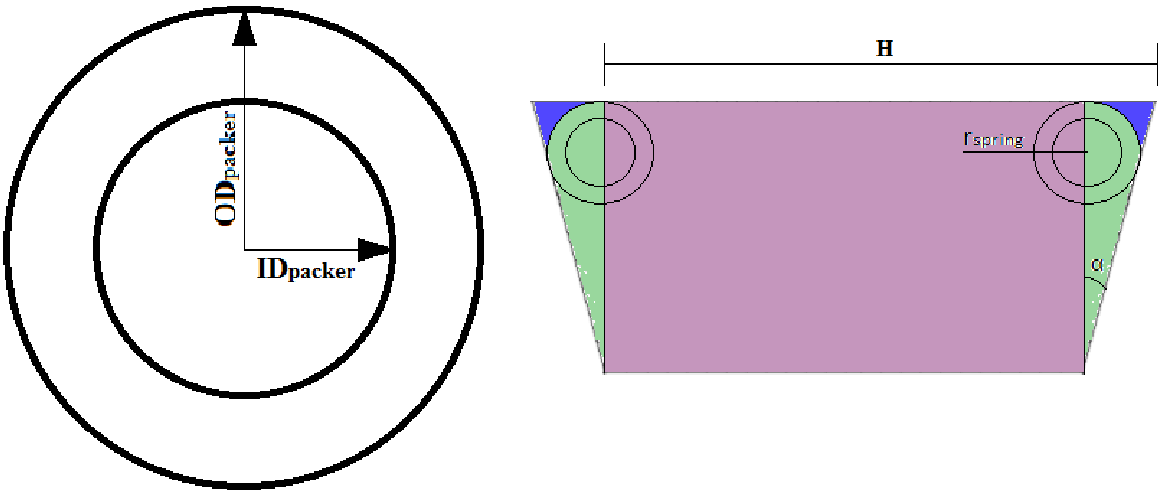
$$\bar{x} = \frac{1}{A_C}\int_0^{r_{spring}*\sin(3{,}5\alpha)} \int_{\frac{y}{\tan(2{,}5\alpha)}}^{-\sqrt{r_{spring}^2 - y^2} + r_{spring}*\left(\frac{\sin(3{,}5\alpha)}{\tan(2{,}5\alpha)} + \cos(3{,}5\alpha)\right)} xdxdy \qquad \text{(Eq.B 19)}$$

Solving eq.20:

Volume of one corner:

$$V_C = 2\pi * \left(OR_{packer} - \bar{x} * \sin(2{,}5\alpha)\right) * A_C \qquad \text{(Eq.B 20)}$$

Volume of the whole packer:



$$V_{packer} = \left(\frac{\pi}{4} * \left(OD_{packer}^2 - ID_{packer}^2\right) * H\right) - 2 * V_C \qquad \text{(Eq.B 21)}$$

Appendix C: Extracts from NS-EN13445:2009

C1: Allowable design stress

6 Maximum allowed values of the nominal design stress for pressure parts

6.1 General

6.1.1 This clause specifies maximum allowed values of the nominal design stress for pressure parts other than bolts and physical properties of steels.

The values to be used within the creep range are given in Clause 19.

NOTE Nominal design stresses for bolting materials are given in clauses 11 and 12.

6.1.2 For a specific component of a vessel, i.e. specific material, specific thickness, there are different values of the nominal design stress for the normal operating, testing, and exceptional load cases.

For exceptional load cases, a higher nominal design stress may be used (see 6.1.3). The manufacturer shall prescribe, in the instructions for use, an inspection of the vessel before returning it to service after occurrence of such an exceptional case.

In assessing testing or exceptional load cases, progressive deformation and fatigue requirements need not be taken into consideration.

6.1.3 The maximum values of the nominal design stress for normal operating and testing load cases shall be determined from the material properties as specified in 6.1.5 and the safety factors given in 6.2 to 6.5. The formulae for deriving the maximum values of nominal design stresses are given in Table 6-1.

For testing group 4 vessels, the maximum value of the nominal design stress for the normal operating load cases shall be multiplied by 0,9.

The nominal safety factor for exceptional load cases shall not be less than that for the testing load cases.

6.1.4 Special considerations may require lower values of the nominal design stress, e.g. risk of stress corrosion cracking, special hazard situations, etc.

6.1.5 For the tensile strength and the yield strength the values shall be those which apply to the materials in the final fabricated condition and shall conform to the minimum values of the technical documentation prepared in accordance with EN 13445-5:2009, clause 5.

NOTE These values will generally be achieved when the heat treatment procedures conform to EN 13445-4:2009.

The minimum values, specified for the delivery condition, can be used for design purposes unless the heat treatment is known to lead to lower values, in which case these lower values shall be used. If the weld metal gives lower strength values after fabrication, these shall be used.

6.1.6 For the determination of the tensile strength and the yield strength above 20 °C procedure of EN 13445-2:2009, 4.2 shall be used.

6.1.7 For the definition of rupture elongation see EN 13445-2:2009, Clause 4.

6.2 Steels (except castings), other than austenitic steels covered by 6.4 and 6.5, with a minimum rupture elongation, as given in the relevant technical specification for the material, below 30 %

6.2.1 Normal operating load cases

The nominal design stress for normal operating load cases f shall not exceed f_d, the smaller of the two following values:

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— the minimum yield strength or 0,2 % proof strength at calculation temperature, as given in the technical specification for the material, divided by the safety factor 1,5; and $\frac{f_y}{1.5}$

— the minimum tensile strength at 20 °C, as given in the technical specification for the material, divided by the safety factor 2,4. $\frac{f_u}{2,4}$

6.2.2 Testing load cases

The nominal design stress for testing conditions f shall not exceed f_{test}, the minimum yield strength or 0,2 % proof strength at test temperature, as given in the technical specification for the material, divided by the safety factor 1,05. $\frac{f_y}{1.05}$

6.3 Alternative route for steels (except castings), other than austenitic steels covered by 6.4 and 6.5, with a minimum rupture elongation, as given in the relevant technical specification for the material, below 30 %

6.3.1 General

Alternative route allows the use of higher nominal design stress with an equivalent overall level of safety if all of the following conditions are met:

a) Material requirements as specified in EN 13445-2:2009 for Design by Analysis – Direct Route.

b) Restriction in construction and welded joints as specified in Clause 5 and in Annex A for Design by Analysis – Direct Route.

c) All welds which must be tested by non-destructive testing (NDT) according to the requirements of EN 13445-5:2009 shall be accessible to NDT during manufacture and also for in-service inspection.

d) Fatigue analysis according to Clause 17 or 18 in all cases.

e) Fabrication requirements as specified in EN 13445-4:2009 for Design by Analysis – Direct Route.

f) NDT as specified in EN 13445-5:2009 for Design by Analysis – Direct Route.

g) Appropriate detailed instructions for in-service inspections are provided in the operating instructions of the manufacturer.

NOTE Until sufficient in-house experience can be demonstrated, the involvement of an independent body, appropriately qualified, is recommended for the assessment of the design (calculations) and for assurance that all requirements are met in materials, fabrication and NDT.

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6.3.2 Normal operating load cases

The nominal design stress for normal operating load cases f shall not exceed f_d, the smaller of the two following values:

— the minimum yield strength or 0,2 % proof strength at calculation temperature, as given in the technical specification for the material, divided by the safety factor 1,5; and

— the minimum tensile strength at 20 °C, as given in the technical specification for the material, divided by the safety factor 1,875.

6.3.3 Testing load cases

The nominal design stress for testing conditions f shall not exceed f_{test}, the minimum yield strength or 0,2 % proof strength at test temperature, as given in the technical specification for the material, divided by the safety factor 1,05.

6.4 Austenitic steels (except castings) with a minimum elongation after rupture, as given in the relevant technical specification for the material, from 30 % to 35 %.

6.4.1 Normal operating load cases

The nominal design stress for normal operating load cases f shall not exceed f_d, the minimum 1 % proof strength at calculation temperature, as given in the technical specification for the material, divided by the safety factor 1,5.

6.4.2 Testing load cases

The nominal design stress for testing load cases f shall not exceed f_{test}, the minimum 1 % proof strength at test temperature, as given in the technical specification for the material, divided by the safety factor 1,05.

6.5 Austenitic steels (except castings) with a minimum rupture elongation, as given in the relevant technical specification for the material, from 35 %.

6.5.1 Normal operating load cases

The nominal design stress for normal operating load cases f shall not exceed f_d the greater of the two values:

a) that derived from 6.4.1; or

b) if a value of $R_{m/T}$ is available, the smaller of two values:

   — the minimum tensile strength at calculation temperature, as given in the technical specification for the material, divided by the safety factor 3,0; and

   — the minimum 1 % proof strength at calculation temperature, as given in the technical specification for the material divided by the safety factor 1,2.

6.5.2 Testing load cases

The nominal design stress for testing load cases f shall not exceed f_{test}, the greater of the two values:

a) the value derived from 6.4.2; and

b) the minimum tensile strength at test temperature, as given in the technical specification for the material, divided by the safety factor 2.

6.6 Cast steels

6.6.1 Normal operating load cases

The nominal design stress for normal operating load cases f shall not exceed f_d, the smaller of the following two values:

- the minimum yield strength or 0,2 % proof strength at calculation temperature, as given in the technical specification for the material divided by the safety factor 1,9;
- the minimum tensile strength at 20 °C, as given in the technical specification for the material, divided by the safety factor 3,0.

6.6.2 Testing load cases

The nominal design stress for testing load cases f shall not exceed f_{test}, the minimum yield strength or 0,2 % proof strength at test temperature, as given in the technical specification for the material, divided by the safety factor 1,33.

NOTE Physical properties of steels are given in Annex O.

Table 6-1 — Maximum allowed values of the nominal design stress for pressure parts other than bolts

Steel designation	Normal operating load cases[a b]	Testing and exceptional load cases[b c]
Steels other than austenitic, as per 6.2 $A < 30\ \%$[d]	$f_d = \min\left(\frac{R_{p0,2/T}}{1{,}5};\frac{R_{m/20}}{2{,}4}\right)$	$f_{test} = \left(\frac{R_{p0,2/T_{test}}}{1{,}05}\right)$
Steels other than austenitic, as per 6.3: Alternative route $A < 30\ \%$[d]	$f_d = \min\left(\frac{R_{p0,2/T}}{1{,}5};\frac{R_{m/20}}{1{,}875}\right)$	$f_{test} = \left(\frac{R_{p0,2/T_{test}}}{1{,}05}\right)$
Austenitic steels as per 6.4 $30\ \% \le A < 35\ \%$[d]	$f_d = \left(\frac{R_{p1,0/T}}{1{,}5}\right)$	$f_{test} = \left(\frac{R_{p1,0/T_{test}}}{1{,}05}\right)$
Austenitic steels as per 6.5 $A \ge 35\ \%$[d]	$f_d = \max\left[\left(\frac{R_{p1,0/T}}{1{,}5}\right);\min\left(\frac{R_{p1,0/T}}{1{,}2};\frac{R_{m/T}}{3}\right)\right]$	$f_{test} = \max\left[\left(\frac{R_{p1,0/T_{test}}}{1{,}05}\right);\left(\frac{R_{m/T_{test}}}{2}\right)\right]$
Cast steels as per 6.6	$f_d = \min\left(\frac{R_{p0,2/T}}{1{,}9};\frac{R_{m/20}}{3}\right)$	$f_{test} = \left(\frac{R_{p0,2/T_{test}}}{1{,}33}\right)$

a For testing group 4 the nominal design stress shall be multiplied by 0,9.

b Yield strength R_{eH} may be used in lieu of $R_{p0,2}$ if the latter is not available from the material standard.

c See 5.3.2 and 6.1.2

d For definition of rupture elongation, see EN 13445-2:2009, Clause 4.

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15 Pressure vessels of rectangular section

15.1 Purpose

This clause specifies requirements for the design of unreinforced and reinforced pressure vessels of rectangular cross-section. For fatigue, designs shall be checked against either clause 17 or clause 18.

15.2 Specific definitions

The following terms and definitions apply in addition to those in clause 3.

15.2.1
membrane stress
equivalent uniform stress through the wall of the vessel, see also C.4.4.2

15.2.2
bending stress
equivalent linear distributed stress through the wall of the vessel, see also C.4.4.3

15.3 Specific symbols and abbreviations

The following symbols and abbreviations apply in addition to those in clause 4:

a	is the inside corner radius;
A_1	is the cross-sectional area of a reinforcing member which is attached to the short side of a vessel;
A_2	is the cross-sectional area of a reinforcing member which is attached to the long side of the vessel;
b	is the unsupported width of a flat plate between reinforcing elements, see Figure 15.6-1;
b_e	is the effective width of a plate in combination with a reinforcing member, see Figure 15.6-1;
b_R	is the pitch between centrelines of reinforcing members on a vessel;
c	is the distance from the neutral axis of a section to the outer fibre of a section and is positive when inwards;
C	is a shape factor determined from the long and short sides of an unsupported plate between stiffeners, see Table 15.6-2;
d	is either the diameter of an opening or the inside diameter of a welded connection if attached by a full penetration weld;
g	is the length of an unsupported span;
h	is the inside length of the long side;
h_1	is the distance between the neutral axes of reinforcing members on the long side;
H	is the inside length of the short side;
H_1	is the distance between the neutral axes of reinforcing members on the short side;
I_1, I_2	is the second moment of area per unit width of a strip of thickness e;
I_{11}	is the second moment of area of the combined reinforcing member and plate of on the short side of the vessel;

I_{21} is the second moment of area of the combined reinforcing member and plate on the long side of the vessel;

k is a factor, see equation (15.5.2-4);

K_3 is a factor for unreinforced vessel to Figure 15.5-1;

l_1, l_x, L, L_y are the dimensions of the vessel;

M_A is the bending moment at the middle of the long side, it is positive when the outside of the vessel is put into compression. It is expressed as bending moment per unit length (in N·mm/mm);

p is the hole pitch along the plate length, see Figure 15.5-2;

p_s is the diagonal hole pitch, see Figure 15.5-2;

α is a factor, see equation (15.5.2-5);

α_1 is a factor, see equation (15.5.1.2-13);

α_3 is a factor, see equation (15.5.1.2-14);

β is the angle between the line of the holes and the long axis, see Figure 15.5-2.

θ is an angle indicating position at the corner of a vessel, see Figure 15.5-2;

μ is the ligament efficiency;

σ_b is the bending stress;

σ_m is the membrane stress;

ϕ is a factor, see equation (15.5.1.2-15).

15.4 General

The equations given in this subclause shall be used for calculation of the membrane and bending stresses in unreinforced and reinforced rectangular pressure vessels. The maximum stress at a given location shall be taken as the sum of the membrane stress and the bending stress at that location.

For vessels operating with extensive fatigue loads (for example sterilizers) the longitudinal corners of the vessel shall be provided with an inside radius not less than three times the wall thickness.

For pressure vessels provided with doors a special analysis shall be performed to detect any deformation in the door and the edge of the vessel.

NOTE Special care should be taken in the choice of gasket for the door.

15.5 Unreinforced vessels

15.5.1 Unreinforced vessels without a stay

15.5.1.1 General

This method applies to vessels of the type shown in Figure 15.5-1.

It is assumed that the thicknesses of the short and long sides are equal. When they are not, the method in 15.6 shall be used.

15.5.1.2 Unperforated plates

Where the thickness of the smaller side is not the same as the thickness of the longer side, the calculation method for reinforced vessels in 15.6 shall be used.

For unreinforced vessels conforming to Figure 15.5-1, the membrane stresses are determined from the following equations:

at C,

$$(\sigma_m)_C = \frac{P(a+L)}{e} \qquad (15.5.1.2\text{-}1)$$

at D,

$$(\sigma_m)_D = (\sigma_m)_C$$

at B,

$$(\sigma_m)_B = \frac{P(a+l_1)}{e} \qquad (15.5.1.2\text{-}2)$$

at A,

$$(\sigma_m)_A = (\sigma_m)_B$$

at a corner, e.g. between B and C, it is given by:

$$(\sigma_m)_{B-C} = \frac{P}{e}\left\{a + \sqrt{\left(L^2 + l_1^2\right)}\right\} \qquad (15.5.1.2\text{-}3)$$

The second moment of area is given by:

$$I_1 = I_2 = e^3/12 \qquad (15.5.1.2\text{-}4)$$

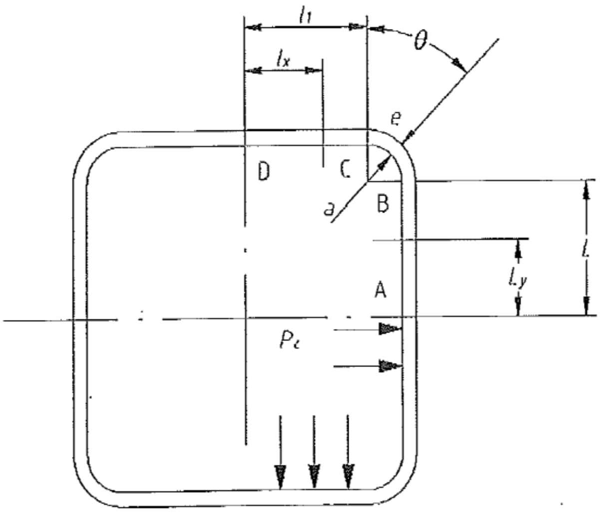


Figure 15.5-1 — Unreinforced vessels

The bending stresses shall be determined from the following equations:

at C,

$$(\sigma_b)_C = \pm \frac{e}{4I_1}\left[2M_A + P(2a \cdot L - 2a \cdot l_1 + L^2)\right] \tag{15.5.1.2-5}$$

at D,

$$(\sigma_b)_D = \pm \frac{e}{4I_1}\left[2M_A + P\left(2a \cdot L - 2a \cdot l_1 + L^2 - l_1^2\right)\right] \tag{15.5.1.2-6}$$

at A,

$$(\sigma_b)_A = \pm \frac{M_A e}{2I_1} \tag{15.5.1.2-7}$$

at B,

$$(\sigma_b)_B = \pm \frac{e}{4I_1}\left[2M_A + PL^2\right] \tag{15.5.1.2-8}$$

at the corner,

$$(\sigma_b)_{B-C} = \pm \frac{e}{4I_1}\left[2M_A + P\{2a(L\cos\theta - l_1(1-\sin\theta)) + L^2\}\right] \tag{15.5.1.2-9}$$

For these equations the following shall apply:

a) the maximum value of $(\sigma_b)_{B-C}$ is given where $\theta = \arctan\,(l_1 / L)$ (15.5.1.2-10)

and

b) the bending moment M_A per unit length, is given by:

$$M_A = P \cdot (-K_3) \tag{15.5.1.2-11}$$

where

$$K_3 = \frac{l_1^2\left(6\varphi^2 \cdot \alpha_3 - 3\pi\varphi^2 + 6\varphi^2 + \alpha_3^3 + 3\alpha_3^2 - 6\varphi - 2 + 1.5\pi\alpha_3^2 \cdot \varphi + 6\varphi \cdot \alpha_3\right)}{3\left(2\alpha_3 + \pi\varphi + 2\right)} \tag{15.5.1.2-12}$$

$$\alpha_1 = H_1 / h_1 \tag{15.5.1.2-13}$$

$$\alpha_3 = L / l_1 \tag{15.5.1.2-14}$$

$$\phi = a / l_1 \tag{15.5.1.2-15}$$

At a location, the maximum stress shall be obtained as stated in 15.4 by summing the membrane and bending stresses.

4 Requirements for materials to be used for pressure-bearing parts

4.1 General

4.1.1 Materials to be used for pressure-bearing parts shall meet the general requirements of 4.1 and the special provisions of 4.2, if applicable. Materials for pressure bearing parts shall be ordered complying with the technical delivery conditions in 4.3.

Marking of materials for pressure-bearing parts shall be performed in accordance with 4.4.

Materials shall be selected to be compatible with anticipated fabrication steps and to be suitable for the internal fluid and external environment. Both normal operating conditions and transient conditions occurring during fabrication transport, testing and operation shall be taken into account when specifying the materials.

NOTE 1 The requirements of 4.1 and 4.2 should also be fulfilled when technical delivery conditions are developed for European material standards, European approval of materials or particular material appraisals.

NOTE 2 When technical delivery conditions for pressure-bearing parts are developed, the structure and requirements of EN 764-4:2002 should be met. Exceptions should be technically justified.

The materials shall be grouped in accordance with CR ISO 15608:2000 to relate manufacturing and inspection requirements to generic material types.

NOTE 3 Materials have been allocated into these groups in accordance with their chemical composition and properties in view of manufacture and heat treatment after welding.

4.1.2 Materials for pressure-bearing parts compliant with the requirements of this European Standard shall be accompanied by inspection documents in accordance with EN 10204:2004. Certificate of specific control (3.1 or 3.2 certificate) shall be required for all steels if Design by Analysis – Direct Route according to Annex B of EN 13445-3:2009 is used.

NOTE The type of inspection document should be in accordance with EN 764-5:2002 and include a declaration of compliance to the material specification.

4.1.3 The materials shall be free from surface and internal defects which can impair their intended usability.

4.1.4 Steels shall have a specified minimum elongation after fracture measured on a gauge length

$$L_o = 5{,}65\sqrt{S_o} \qquad (4.1\text{-}1)$$

where

S_o is the original cross sectional area within the gauge length.

The minimum elongation after fracture in any direction shall be ≥ 14 %;

However, lower elongation values may also be applied (e.g. for fasteners or castings), provided that appropriate measures are taken to compensate for these lower values and the specific requirements are verifiable.

NOTE Examples for compensation:

— application of higher safety factors in design;

— performance of burst tests to demonstrate ductile material behaviour.

4.1.5 When measured on a gauge length other than that stated in 4.1.4, the minimum elongation after fracture shall be determined by converting the elongation given in 4.1.4 in accordance with

— EN ISO 2566-1:1999 for carbon and low alloy steels;

— EN ISO 2566-2:1999 for austenitic steels.

4.1.6 Steels shall have a specified minimum impact energy measured on a Charpy-V-notch impact test specimen (EN 10045-1:1990) as follows:

— ≥ 27 J for ferritic and 1,5 % to 5 % Ni alloy steels;

— ≥ 40 J for steels of material group 8, 9.3 and 10

at a test temperature in accordance with Annex B, but not higher than 20 °C. The other requirements of Annex B shall also apply.

4.1.7 The chemical composition of steels intended for welding or forming shall not exceed the values in Table 4.1-1. Line 2 of the table refers to vessels or parts designed using Design by Analysis – Direct Route according to Annex B of EN 13445-3:2009. Exceptions shall be technically justified.

Table 4.1-1 — Maximum carbon-, phosphorus- and sulphur contents for steels intended for welding or forming

Steel group (according to Table A-1)	Maximum content of cast analysis		
	% C	% P	% S
Steels (1 to 6 and 9)	0,23[a]	0,035	0,025
Steels (1 to 6 and 9) when DBA – Direct Route is used [c]	0,20	0,025	0,015
Ferritic stainless steels (7.1)	0,08	0,040	0,015
Martensitic stainless steels (7.2)	0,06	0,040	0,015
Austenitic stainless steels (8.1)	0,08	0,045	0,015[b]
Austenitic stainless steels (8.2)	0,10	0,035	0,015
Austenitic-ferritic stainless steels (10)	0,030	0,035	0,015

[a] Maximum content of product analysis 0,25 %.

[b] For products to be machined a controlled sulphur content of 0,015 % to 0,030 % is permitted by agreement provided the resistance to corrosion is satisfied for the intended purpose.

[c] In addition the ratio on thickness reduction (ratio of initial thickness of slab/ingot to the thickness of the final plate) shall be equal or greater than:

— 4 for NL2 steels and steels of material group 9;

— 3 for other materials.

Annex A
(normative)

Design requirements for pressure bearing welds

This annex specifies design requirements of welds for permanent use to be applied in the construction of pressure vessels.

NOTE See also EN 13445-4:2009 and EN 13445-5:2009 for possible additional requirements on welds.

The following data are included:

— a figure of the joint in finished condition;

— design requirements mainly on geometry;

— a list of applicable testing groups as referred to in EN 13445-5:2009;

— the applicable fatigue class as referred to in this Part, clauses 17 and 18 (This does not apply to testing group 4 vessels);

— recommendations for prevention of lamellar tearing;

— recommendations for prevention of corrosion;

— reference to the recommended weld details given in EN 1708-1:1998;

The following groups of welded joints are included:

— group M: longitudinal welds in cylinders and cones, welds in spheres and dished ends (Table A-1);

— group C: circumferential welds in cylinders and cones, connecting weld between dished end and shell (Table A-2);

— group E: welds for flat end to shell (Table A-3);

— group TS: welded joints for tubesheet to shell (Table A-4);

— group T: welded joints for tube to tubesheet (Table A-5);

— group S: welded joints for socket connections (Table A-6);

— group F: welded joints for flanges and collars (Table A-7);

— group N: welded joints for nozzles (Table A-8);

— group B: circumferential welds in bellows (Table A-9).

In each group the preferred joints are given first.

Table A-1 — Pressure bearing welds - Longitudinal welds in cylinders and cones, welds in spheres and dished ends

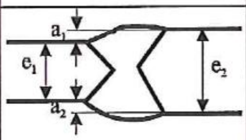
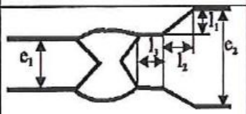
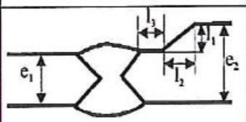
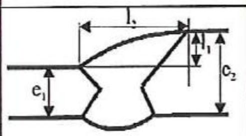
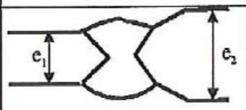
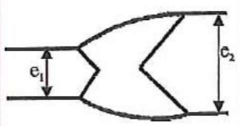
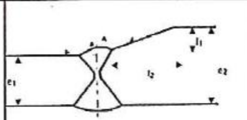
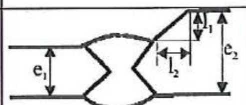
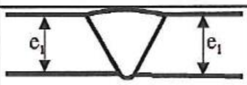
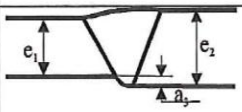
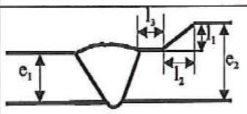
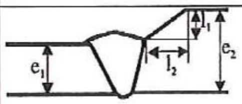
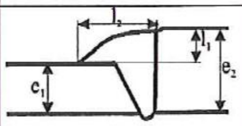
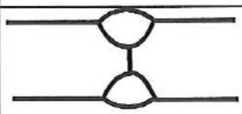
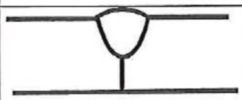
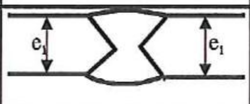
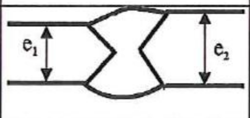
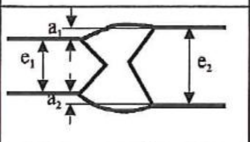
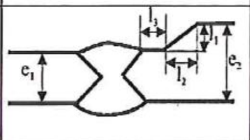
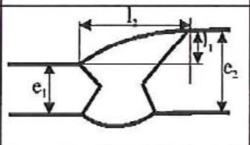
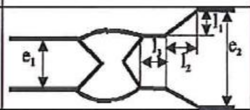
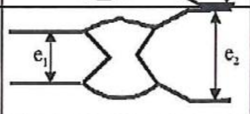
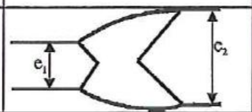
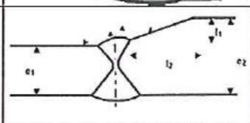
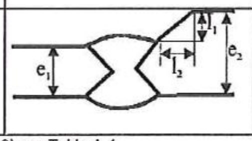
Ref.	Type of joints	Design requirements	Applicable weld testing group	Fatigue class [1)]	Lamellar tearing susceptibility [2)]	Corrosion [3)]	EN 1708 -1:1998
M 1			1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.2	A	N	1.1.4
M 2		$e_2 - e_1 \leq \text{Min}\left[0{,}3e_1 ; 6\right]$ $a_2 \leq 3\,\text{mm}$	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.2	A	N	1.1.4
M 3		$l_3 \geq 2e_1$ $l_1 / l_2 \leq 1/4$	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.2	A	N	1.1.6
M 4		$l_3 \geq 2e_1$ $l_1 / l_2 \leq 1/4$	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.2	A	N	1.1.6
M 5		$e_2 - e_1 \leq \text{Min}\left[0{,}15e_1 ; 3\right]$ $l_1 / l_2 \leq 1/4$	1, 2, 3, 4	see Table 18.4 details n° 1.3	A	N	1.1.4
M 6		slope : see M3 with smooth transition	1, 2, 3, 4	see Table 18.4 details n° 1.3	A	N	1.1.5
M 7		slope : see M3 with smooth transition	1, 2, 3, 4	see Table 18.4 details n° 1.3	A	N	1.1.4
M 8		$l_1 / l_2 \leq 1/4$ with smooth transition and angles > 150 °	1, 2, 3, 4	see Table 18.4 details n° 1.3	A	N	1.1.5
M 9		$l_1 / l_2 \leq 1/4$ with smooth transition NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4		A	N	1.1.5

Table A-1 — Pressure bearing welds - Longitudinal welds in cylinders and cones, welds in spheres and dished ends *(continued)*

Ref.	Type of joints	Design requirements	Applicable weld testing group	Fatigue class [1)]	Lamellar tearing susceptibility [2)]	Corrosion [3)]	EN 1708 -1:1998
M 10		allowed for fatigue only if full penetration can be verified at least by visual inspection	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.5	A	N	1.1.1
M 11		$e_2 - e_1 \le \text{Min}[0{,}3e_1 ; 6]$ $a_3 \le \text{Min}[0{,}1e_1 ; 2]$ see M 10 for fatigue	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.5	A	N	1.1.1
M 12		see M 4 see M 11	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.5	A	N	1.1.3
M 13		**NOT ALLOWED**					
M 14		**NOT ALLOWED**					
M 15		**NOT ALLOWED**					
M 16		**NOT ALLOWED**					

1) Fatigue class: see clauses 17 and 18.
2) Lamellar tearing susceptibility: A = no risk B = possible risk.
3) Corrosion N = normal conditions S = not permitted.

Table A-2 — Pressure bearing welds - Circumferential welds in cylinders and cones, connecting weld between dished end and shell

Ref.	Type of joint	Design requirements	Applicable weld testing group	Fatigue class 1)	Lamellar tearing susceptibility 2)	Corrosion 3)	EN 1708-1:1998
C 1			1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.2	A	N	1.1.4
C 2		$e_2 - e_1 \leq \mathrm{Min}\left[0{,}15e_1\,;3\right]$	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.2	A	N	1.1.4
C 3		$e_2 - e_1 \leq \mathrm{Min}\left[0{,}3e_1\,;6\right]$ $a_2 \leq 3\,\mathrm{mm}$	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.2	A	N	1.1.4
C 4		$l_3 \geq 2e_1$ $l_1/l_2 \leq 1/3$	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.2	A	N	1.1.6
C 5		$l_1/l_2 \leq 1/3$	1, 2, 3, 4	see Table 18.4 detail n° 1.3	A	N	1.1.4
C 6		see C 4	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.2	A	N	1.1.6
C 7		$l_1/l_2 \leq 1/3$ with smooth transition	1, 2, 3, 4	see Table 18.4 details n° 1.3	A	N	1.1.5
C 8		See C 5	1, 2, 3, 4	see Table 18.4 details n° 1.3	A	N	1.1.4
C 9		$l_1/l_2 \leq 1/3$ with smooth transition and angles > 150 °	1, 2, 3, 4	see Table 18.4 details n° 1.3	A	N	1.1.5
C 10		$l_1/l_2 \leq 1/3$ with smooth transition NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	3, 4	see Table 18.4 details n° 1.3 for testing group 3	A	N	1.1.5

1), 2), 3) see Table A-1

Table A-2 — Pressure bearing welds - Circumferential welds in cylinders and cones, connecting weld between dished end and shell *(continued)*

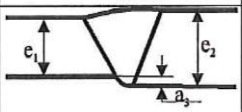
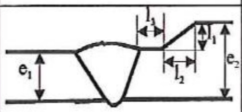
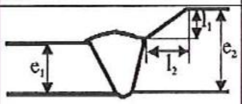
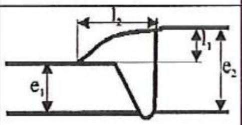
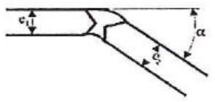
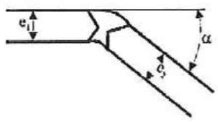
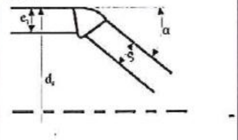
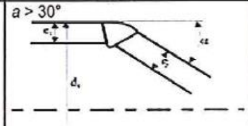
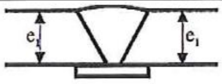
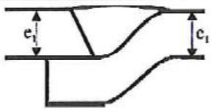
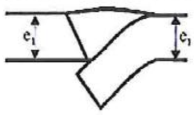
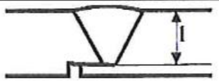
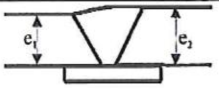
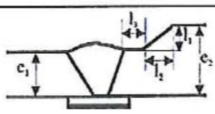
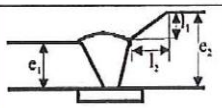
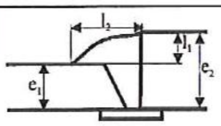
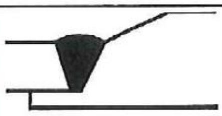
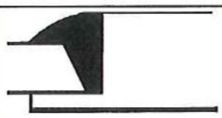
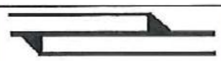
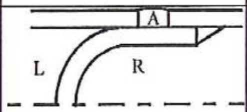
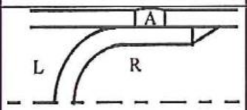
Ref	Type of joint	Design requirements	Applicable weld testing group	Fatigue class [1]	Lamellar tearing susceptibility [2]	Corrosion [3]	EN 1708-1:1998
C 11		allowed for fatigue only if full penetration can be verified	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.5	A	N	1.1.1
C 12		see C 3	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.5	A	N	1.1.1
C 13		see C 4	1, 2, 3, 4	see Table 18.4 details n° 1.1 and 1.5	A	N	1.1.3
C 14		see C 10 with smooth transition	1, 2, 3, 4	see Table 18.4 details n° 1.3 and 1.5	A	N	1.1.2
C 15		NOT ALLOWED					
C 16	$\alpha \leq 30°$ 	in case of unequal thicknesses, limited to: $e_2 - e_1 \leq \mathrm{Min}[0{,}3e_1\,;4]$	1, 2, 3, 4	see Table 18.4 detail n° 1.4	A	N	-
C 17	$\alpha > 30°$ 	in case of unequal thicknesses, limited to: $e_2 - e_1 \leq \mathrm{Min}[0{,}3e_1\,;4]$ — calculation of stresses — round the weld inside by grinding	1, 2, 3, 4	see Table 18.4 detail n° 1.4	A	N	-
1), 2), 3) see Table A-1							

Table A-2 — Pressure bearing welds - Circumferential welds in cylinders, cones and dished ends *(continued)*

Ref.	Type of joint	Design requirements	Applicable weld testing group	Fatigue class [1]	Lamellar tearing susceptibility [2]	Corrosion [3]	EN 1708-1:1998
C 18	$a \le 30°$ 	in case of unequal thicknesses, limited to: $e_2 - e_1 \le \mathrm{Min}[0{,}3e_1 ; 4]$	1, 2, 3, 4	63 with 100 % surface NDT 80 if root flush grounded	A	N	-
C 19	$a > 30°$ 	in case of unequal thicknesses, limited to: $e_2 - e_1 \le \mathrm{Min}[0{,}3e_1 ; 4]$ $d_o \le 600\,\mathrm{mm}$	1, 2, 3, 4	50 with 100 % surface NDT 71 if root flush grounded	A	N	-
C 20		NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	see § 5.7.4.2	see Table 18.4 detail n° 1.6	A	S	-
C 21		see § 5.7.4.1 NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	see § 5.7.4.1	see Table 18.4 detail n° 1.7	A	S	-
C 22		see § 5.7.4.1 NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	see § 5.7.4.1	see Table 18.4 detail n° 1.7	A	S	-
C 23		*l* is the minimum required thickness NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	see § 5.7.4.2	see Table 18.4 detail n° 1.6	A	S	-
C 24		see C 2 NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	see § 5.7.4.2	see Table 18.4 detail n° 1.6	A	S	-
C 25		see C 4 NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	see § 5.7.4.2	see Table 18.4 detail n° 1.6	A	S	-

1), 2), 3) see Table A-1

Table A-2 — Pressure bearing welds - Circumferential welds in cylinders, cones and dished ends *(continued)*

Ref	Type of joint	Design requirements	Applicable weld testing group	Fatigue class [1]	Lamellar tearing susceptibility [2]	Corrosion [3]	EN 1708-1:1998
C 26		see C 10 NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	see § 5.7.4.2 testing group 4	-	A	S	-
C 27		NOT ALLOWED					
C 28		see C 4 NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	see § 5.7.4.2	see Table 18.4 detail n° 1.6	A	S	-
C 29		see C 4 NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	see § 5.7.4.2 testing group 4	not allowed	A	S	-
C 30		NOT ALLOWED					
C 31		NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	N	-
C 32		A = circumferential weld $l > 2\min(e_1, e_2)$ see C 35 L left side R right side Pressure applied on either side NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	S on L side N on R side	9.1.2
C 33		A = plug weld $l > 2\min(e_1, e_2)$ see C 35 L left side R right side Pressure applied on either side NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	S on L side N on R side	9.1.2

1), 2), 3) see Table A-1

Table A-2 — Pressure bearing welds - Circumferential welds in cylinders, cones and dished ends *(concluded)*

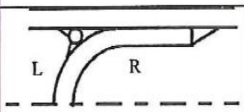
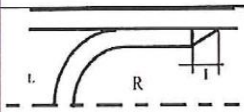
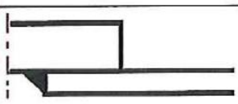
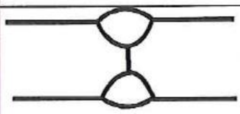
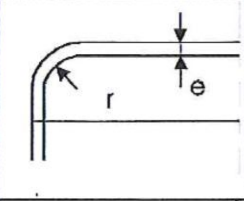
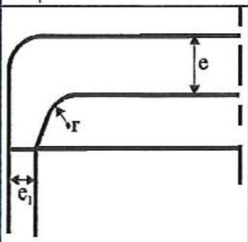
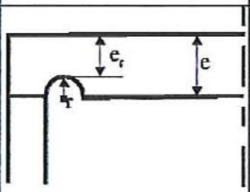
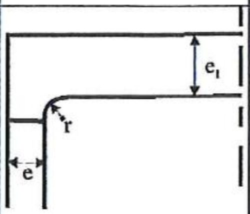
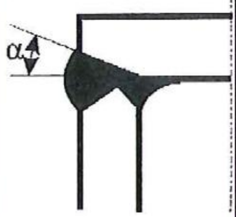
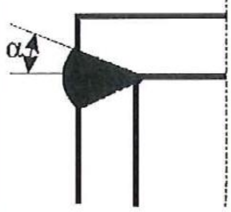
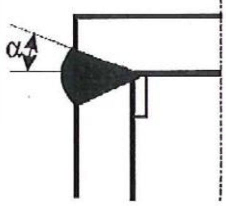
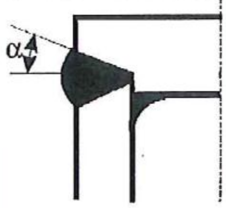
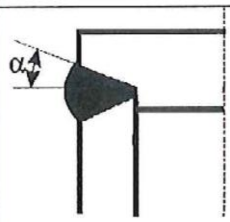
Ref	Type of joint	Design requirements	Applicable weld testing group	Fatigue class 1)	Lamellar tearing susceptibility 2)	Corrosion 3)	EN 1708-1:1998
C 34		$l > 2\min(e_1, e_2)$ see C 35 L left side R right side Pressure applied on either side NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	N	-
C 35		$l > 2\min(e_1, e_2)$ if the weld is at the end of a shell, minimum distance between the weld and the end shall be 5 mm. L left side R right side Pressure applied on either side NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	S on L side N on R side	9.1.1
C 36		**NOT ALLOWED**					
C 37		**NOT ALLOWED**					
C 38		**NOT ALLOWED**					
1), 2), 3) see Table A-1							

Table A-3 — Pressure bearing welds - Flats ends

Ref.	Type of joint	Design requirements	Applicable weld testing group	Fatigue class [1]	Lamellar tearing susceptibility [2]	Corrosion [3]	EN 1708-1:1998
E 1		all allowed circumferential joints can be used $r \geq 1{,}3e$	1, 2, 3, 4	adopt class of relevant reference C	A	N	see for relevant reference C
E 2		all allowed circumferential joints can be used $r \geq 1{,}3e$ and $r \geq 8\,\text{mm}$	1, 2, 3, 4	adopt class of relevant reference C	A	N	see for relevant reference C
E 3		all allowed circumferential joints can be used $r \geq 0{,}2e_r$	1, 2, 3, 4	see Table 18.4 detail n° 2.2	B	N	8.1.9
E 4		all allowed circumferential joints can be used $r \geq e/3$	1, 2, 3, 4	see Table 18.4 detail n° 2.2	A if forged B if machined from plate	N	-

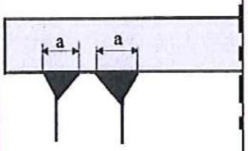
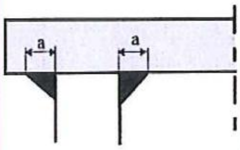
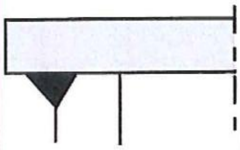
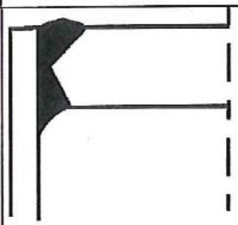
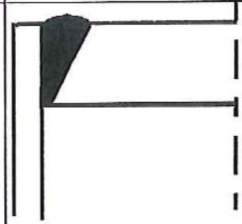
1), 2), 3) see Table A-1

Table A-3 — Pressure bearing welds - Flats ends *(continued)*

Ref.	Type of joint	Design requirements	Applicable weld testing group	Fatigue class [1]	Lamellar tearing susceptibility [2]	Corrosion [3]	EN 1708-1:1998
E 5		NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	3, 4	see Table 18.4 detail n° 2.1 a for testing group 3	A if $\alpha \geq 15°$ B if $\alpha < 15°$	N	8.1.2
E 6		NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	3, 4	see Table 18.4 detail n° 2.1 c for testing group 3	A if $\alpha \geq 15°$ B if $\alpha < 15°$	N	8.1.3
E 7		NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	A if $\alpha \geq 15°$ B if $\alpha < 15°$	S	-
E 8		NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	3, 4 1, 2 if ground and back welded	see Table 18.4 detail n° 2.1 a or b for testing groups 1, 2, and 3	A if $\alpha \geq 15°$ B if $\alpha < 15°$	N	8.1.8
E 9		NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	A if $\alpha \geq 15°$ B if $\alpha < 15°$	S	8.1.7

1), 2), 3) see Table A-1

Table A-3 — Pressure bearing welds - Flats ends *(continued)*

Ref.	Type of joint	Design requirements	Applicable weld testing group	Fatigue class 1)	Lamellar tearing susceptibility 2)	Corrosion 3)	EN 1708-1:1998
E 10		$a \geq e_s$ NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	3, 4 if $a \geq 16\,\mathrm{mm}$ 4 if $a < 16\,\mathrm{mm}$	see Table 18.4 detail n° 2.1 b for testing group 3	B	N	-
E 11		$a \geq e_s$ NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	3, 4 if $a \geq 16\,\mathrm{mm}$ 4 if $a < 16\,\mathrm{mm}$	see Table 18.4 detail n° 2.1 b for testing group 3	B	N	8.1.1
E 12		NOT ALLOWED					
E 13		NOT ALLOWED					
E 14			1, 2, 3, 4	see Table 18.4 detail n° 2.3 a	B	N	8.1.5
E 15			1, 2, 3, 4	see Table 18.4 detail n° 2.3 c	B	N	8.1.5

1), 2), 3) see Table A-1

Table A-3 — Pressure bearing welds - Flats ends *(continued)*

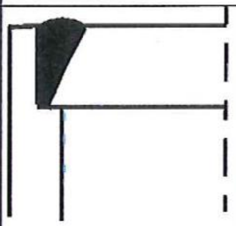
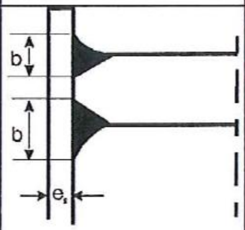
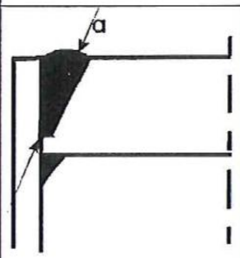
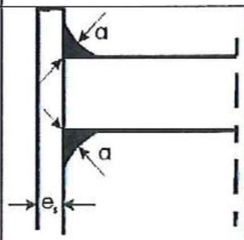
Ref.	Type of joint	Design requirements	Applicable weld testing group	Fatigue class [1]	Lamellar tearing susceptibility [2]	Corrosion [3]	EN 1708-1:1998
E 16		NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	S	-
E 17		$b \geq e_s$ NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	3, 4 if $b < 16\,\text{mm}$ 1, 2, 3, 4 if $b \geq 16\,\text{mm}$	see Table 18.4 detail n° 2.3 b	B	N	8.1.5
E 18		$a \geq 1{,}4\,e_s$ NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	N	8.1.6
E 19		$a \geq 0{,}7\,e_s$ NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	3, 4 if $a \geq 16\,\text{mm}$ 4 if $a < 16\,\text{mm}$	see Table 18.4 detail n° 2.3 b for testing group 3	B	N	8.1.5
1), 2), 3) see Table A-1							

Table A-3 — Pressure bearing welds - Flats ends *(continued)*

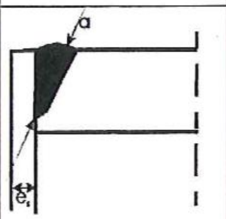
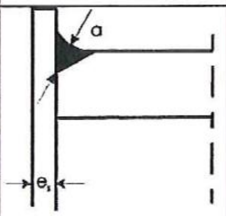
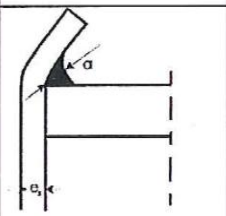
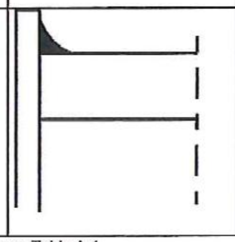
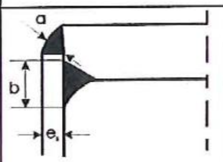
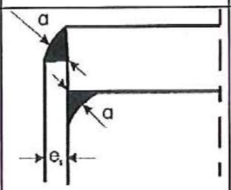
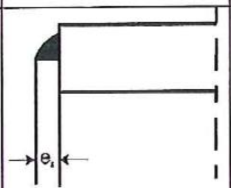
Ref.	Type of joint	Design requirements	Applicable weld testing group	Fatigue class [1]	Lamellar tearing susceptibility [2]	Corrosion [3]	EN 1708-1:1998
E 20		$a \geq 1{,}4\, e_s$ NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	S	-
E 21		$a \geq 1{,}4\, e_s$ NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	S	-
E 22		$a \geq 0{,}7\, e_s$ NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	S	-
E 23		NOT ALLOWED					
1), 2), 3) see Table A-1							

Table A-3 — Pressure bearing welds - Flats ends *(concluded)*

Ref.	Type of joint	Design requirements	Applicable weld testing group	Fatigue class [1]	Lamellar tearing susceptibility [2]	Corrosion [3]	EN 1708-1:1998
E 24		$a \geq 0{,}7\,e_s$ $b \geq e_s$ NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	N	-
E 25		$a \geq e_s$ NOT ALLOWED FOR DBA-DR AND CREEP DESIGN	4	-	B	N	-
E 26		NOT ALLOWED					
1), 2), 3) see Table A-1							

Appendix D: Letter from CEN regarding NS-EN 13445-3:2009

European Committee for Standardization
Comité Européen de Normalisation
Europaïsches Komitee für Normung

EN 13445 MHD

form (2009) 3-15-04-2 _ACP

Registration number	*Date of submission*	*Target date for answer*	*Date of acceptance*
(2009) 3-15-04-2	2012/03/16	2012/03/30	2012/04/17

Part number	*Page number*	*Subclause number*	*Reference of the national standard used*
3-15	320	15.5.1.2 , eq 15.5.1.2-12, 15.5.1.2-11	

Question

Question 1: Is the method for calculating stresses in rectangular pressure vessels described in 15.5.1.2 applicable to pressure vessels of same type with no or very small radius (a)?

Question 2: Is there something wrong with the equation 15.5.1.2-12 or 15.5.1.2-11 ? As I don`t manage to get the correct unit.

Answer(s)

EN 13445/MHD answer

1) The method in 15.5.1.2 makes use of design formulas based on membrane+bending stresses derived from standard beam theory (straight beams and curved beams of large curvature). These formulas are relevant for design against static loading.

They may also be used for fatigue assessment provided the true stresses do not seriously deviate from those given by the standard beam theory. This condition is fulfilled as far as the corner radius a is not smaller than 3e. When a is smaller than 3e , the corner region behaves as a beam of small curvature, giving rise to inside stresses higher than those given by the formulas. This is the reason why a is limited to 3e in 15.4 for the case of "vessels operating with extensive fatigue loads".

So, when a < 3e , the method in 15.5.1.2 may be used only for static design and should be completed with a fatigue assessment based on more accurately calculated stresses at corners, if fatigue has to be considered.

2) K3 has the dimension 'length * length' and P is 'force / (length * length)', giving MA the required dimension 'force'

No further action needed

Question from:

Name Jakobsen Sven Tore

Company University of Stavanger *Country* Norway

Date 2012/03/05

Appendix E: S690QL (Masteel 2012)

S690QL – High Yield Steel

GET A QUOTE - CLICK HERE

Section Links :
Home Page : High Yield Steel Home Page
Specification : EN 10025 PT6:2004
Grades : S690QL | S890QL | S960QL | S1100QL

S690 QL is a high yield structural steel grade produced in compliance with EN 10025:6:2004. The material is heat treated using the quench and temper process and has good bending and welding properties.

Due to the materials high strength nature, using S690QL will promote leaner designed structures with increased payload capacity and energy efficiency.

Please refer to the technical details below referring to S690QL.

Grade Designation

- **S** = Structural Steel
- **690** = minimum yield strength (MPa)
- **Q** = Quenching & Tempering
- **L** = Low notch toughness testing temperature

Chemical Composition

Grade	C	Si	Mn	P	S	N	B	Cr	Cu	Mo	Nb*	Ni	Ti*	V*	Zr*
S690QL	0.20	0.80	1.70	0.025	0.015	0.015	0.0050	1.50	0.50	0.70	0.06	2.0	0.05	0.12	0.15

* At least 0.015% of grain-refining element present. Aluminium is also one of these elements.

Mechanical Properties

Designation		Mechanical Properties (ambient temperature)						
Steel Name	Steel Number	Min. Yield Strength Reh MPa			Tensile Strength Rm MPa			Min. % elongation after fracture
		Nominal thickness (mm)			Nominal thickness (mm)			
		≥3 ≤50	≥50 ≤100	≥100 ≤150	≥3 ≤50	≥50 ≤100	≥100 ≤150	
S690QL	1.8931	690	650	630	770/940	760/930	710/900	14

V Notch Impact Testing

Grade	Sample Orientation	@ 0°C	@-20°C	@-40°C	@-60°C
S690QL	Longitudinal	50 J	40 J	30 J	--
	Traverse	35 J	30 J	27 J	--

(Please note: the technical information above is for guidance only – for exact specifications please check with our Sales Team)

The steel is used in a variety of sectors including...

- Heavy transportation
- Machine building
- Steel constructions
- Lifting equipment

For more information about S690QL high yield steel or to receive a competitive quotation, please CLICK HERE.

Appendix F: S355 EN 10025:2004 (Leeco steel 2012)

S355 European Standard Steel

S355 EN 10025 : 2004 Standard Structural Steel Plate

S355 structural steel plate is a high-strength low-alloy European standard structural steel covering four of the six "Parts" within the **EN 10025 – 2004** standard. With minimum yield of 50,000 KSI, it meets requirements in chemistry and physical properties similar to ASTM A572 / 709. Careful attention should always be placed on the specific variation of S355 required if considering substitute material.

S355 is used in almost every facet of structural fabrication. Typical applications include:

- Structural steelworks: bridge components, components for offshore structures
- Power plants
- Mining and earth-moving equipment
- Load-handling equipment
- Wind tower components

For more information, please contact a Leeco Steel representative.

Part 2 - Non-Alloy Structural Steels	
S...	Structural Steel
E...	Engineering Steel
355	Minimum yield strength (Reh) in Mpa up to 16mm
...JR..	Charpy V-notch (Longitudinal) 27 J @ +20 Celsius
...JO..	Charpy V-notch (Longitudinal) 27 J @ 0 Celsius
...J2..	Charpy V-notch (Longitudinal) 27 J @ -20 Celsius
...K2..	Charpy V-notch (Longitudinal) 40 J @ +20 Celsius
...+AR	Supply condition "As Rolled"
...+N	Supply condition "Normalized or Normalized Rolled"
Additional Options	
...Z..	Grade with improved properties perpendicular to the surface
Part 3 - Normalised / Normalised Rolled Weldable Fine Grain Structural Steel	
S...	Structural Steel
355	Minimum yield strength (Reh) in Mpa up to 16mm
...N..	Longitudinal Charpy V-notch impacts at a temperature not lower than minus 20 Celsius
...NL..	Longitudinal Charpy V-notch impacts at a temperature not lower than minus 50 Celsius
Additional Options	
...Z..	Grade with improved properties perpendicular to the surface

Part 4 - Thermomechanically Rolled Weldable Fine Grain Structural Steels	
S...	Structural Steel
355	Minimum yield strength (Reh) in Mpa up to 16mm
...M..	Longitudinal Charpy V-notch impacts at a temperature not lower than minus 20 Celsius
...ML..	Longitudinal Charpy V-notch impacts at a temperature not lower than minus 50 Celsius
Additional Options	
...Z..	Grade with improved properties perpendicular to the surface
Part 4 - Thermomechanically Rolled Weldable Fine Grain Structural Steels	
S...	Structural Steel
355	Minimum yield strength (Reh) in Mpa up to 16mm
...JO..	Charpy V-notch (Longitudinal) 27 J @ 0 Celsius
...J2..	Charpy V-notch (Longitudinal) 27 J @ -20 Celsius
...K2..	Charpy V-notch (Longitudinal) 40 J @ +20 Celsius
...W..	Improved Atmospheric Corrosion Resistance
...P..	Greater phosphorus content (grade S355 only)
...+AR	Supply condition "As Rolled"
...+N	Supply condition "Normalized or Normalized Rolled"
Additional Options	
...Z..	Grade with improved properties perpendicular to the surface

Mechanical Properties									(1MPa = 145.03 PSI)
Grade	Yield Strength (Mpa) in Nom. Thicknesses (mm)								
	>3 - ≤16	>16 - ≤40	>40 - ≤63	>63- ≤80	>80 - ≤100	>100 - ≤150	>150 - ≤200	>200 - ≤250	>250 - ≤400
S355JR	355 min.	345 min.	335 min.	325 min.	315 min.	295 min.	285 min.	275 min.	-
S355JO	355 min.	345 min.	335 min.	325 min.	315 min.	295 min.	285 min.	275 min.	-
S355J2	355 min.	345 min.	335 min.	325 min.	315 min.	295 min.	285 min.	275 min.	265 min.
S355K2	355 min.	345 min.	335 min.	325 min.	315 min.	295 min.	285 min.	275 min.	265 min.
S355N	355 min.	345 min.	335 min.	325 min.	315 min.	295 min.	285 min.	275 min.	
S355NL	355 min.	345 min.	335 min.	325 min.	315 min.	295 min.	285 min.	275 min.	
S355M	355 min.	345 min.	335 min.	325 min.	325 min.	295 min.			
S355ML	355 min.	345 min.	335 min.	325 min.	325 min.	295 min.			
S355J0WP	355 min.	345 min.	-	-	-	-			
S355J2WP	355 min.	345 min.	-	-	-	-			
S355J0W	355 min.	345 min.	335 min.	325 min.	315 min.	295 min.			

Grade	Tensile Strength (Mpa) in Nom. Thicknesses (mm)								
	>3 - ≤16	>16 - ≤40	>40 - ≤63	>63- ≤80	>80 - ≤100	>100 - ≤150	>150 - ≤200	>200 - ≤250	>250 - ≤400
S355JR	470 - 630	470 - 630	470 - 630	470 - 630	470 - 630	450 - 600	450 - 600	450 - 600	-
S355JO	470 - 630	470 - 630	470 - 630	470 - 630	470 - 630	450 - 600	450 - 600	450 - 600	-
S355J2	470 - 630	470 - 630	470 - 630	470 - 630	470 - 630	450 - 600	450 - 600	450 - 600	450 - 600
S355K2	470 - 630	470 - 630	470 - 630	470 - 630	470 - 630	450 - 600	450 - 600	450 - 600	450 - 600
S355N	470 - 630	470 - 630	470 - 630	470 - 630	470 - 630	450 - 600	450 - 600	450 - 600	-
S355NL	470 - 630	470 - 630	470 - 630	470 - 630	470 - 630	450 - 600	450 - 600	450 - 600	-
S355M	470 - 630	470 - 630	450 - 610	440 - 600	440 - 600	430 - 590	-	-	-
S355ML	470 - 630	470 - 630	450 - 610	440 - 600	325 440 - 600.	430 - 590	-	-	-
S355J0WP	470 - 630	470 - 630	-	-	-	-	-	-	-
S355J2WP	470 - 630	470 - 630	-	-	-	-	-	-	-
S355J0W	470 - 630	470 - 630	470 - 630	470 - 630	470 - 630	450 - 600	-	-	-

Chemical Composition															
Grade	Thickness (mm)	C	Mn	P	S	Si	Cr	Ni	Mo	V	Al	Cu	Ti	Nb	N
S355JR	<40	.24 max	1.60 max	.035 max	.035 max	.55 max	-	-	-	-	-	.55 max	-	-	.012 max
S355JR	41 - ≤150	.24 max	1.60 max	.035 max	.035 max	.55 max	-	-	-	-	-	.55 max	-	-	.012 max
S355JR	151 - ≤250	.24 max	1.60 max	.035 max	.035 max	.55 max	-	-	-	-	-	.55 max	-	-	.012 max
S355JO	<40	.20 max	1.60 max	.030 max	.030 max	.55 max	-	-	-	-	-	.55 max	-	-	.012 max
S355JO	41 - ≤150	.22 max	1.60 max	.030 max	.030 max	.55 max	-	-	-	-	-	.55 max	-	-	.012 max
S355JO	151 - ≤250	.22 max	1.60 max	.030 max	.030 max	.55 max	-	-	-	-	-	.55 max	-	-	.012 max
S355J2	<40	.20 max	1.60 max	.025 max	.025 max	.55 max	-	-	-	-	-	.55 max	-	-	-
S355J2	41 - ≤150	.22 max	1.60 max	.025 max	.025 max	.55 max	-	-	-	-	-	.55 max	-	-	-
S355J2	151 - ≤250	.22 max	1.60 max	.025 max	.025 max	.55 max	-	-	-	-	-	.55 max	-	-	-
S355J2	>250 - ≤400	.22 max	1.60 max	.025 max	.025 max	.55 max	-	-	-	-	-	.55 max	-	-	-
S355K2	<40	.20 max	1.60 max	.025 max	.025 max	.55 max	-	-	-	-	-	.55 max	-	-	-
S355K2	41 - ≤150	.22 max	1.60 max	.025 max	.025 max	.55 max	-	-	-	-	-	.55 max	-	-	-
S355K2	151 - ≤250	.22 max	1.60 max	.025 max	.025 max	.55 max	-	-	-	-	-	.55 max	-	-	-
S355K2	>250 - ≤400	.22 max	1.60 max	.025 max	.025 max	.55 max	-	-	-	-	-	.55 max	-	-	-
S355N	<250	.20 max	.90 - 1.65	.030 max	.025 max	.50 max	.030 max	.50 max	.10 max	.12 max	.02 min	.55 max	.05 max	.05 max	.015 max
S355NL	<250	.18 max	.90 - 1.65	.025 max	.020 max	.50 max	.030 max	.50 max	.10 max	.12 max	.02 min	.55 max	.05 max	.05 max	.015 max
S355M	<150	.14 max	1.60 max	.030 max	.025 max	.50 max	.030 max	.50 max	.10 max	.10 max	.02 min	.55 max	.05 max	.05 max	.015 max
S355ML	<150	.14 max	1.60 max	.025 max	.020 max	.50 max	.030 max	.50 max	.10 max	.10 max	.02 min	.55 max	.05 max	.05 max	.015 max
S355J0WP	<40	.12 max	1.0 max	.06 - .15	.035 max	.75 max		-	-	-	-	.25 - .55	-	-	.009 max
S355J2WP	<40	.12 max	1.0 max	.06 - .15	.030 max	.75 max		-	-	-	-	.25 - .55	-	-	-
S355J0W	<150	.16 max	.50 - 1.50	.035 max	.035 max	.50 max		-	-	-	-	.25 - .55	-	-	.009 max

Charpy V-Notch Testing												
	Elongation in Nom. Thicknesses (mm)											
Grade	>3 - ≤16	>16 - ≤40	>40 - ≤63	>63 - ≤80	>80 - ≤100	>100 - ≤150	>150 - ≤200	>200 - ≤250	>250 - ≤400	Grain	@ Degrees	Min. Absorbed Energy
S355JR	22 min (L)		21 min (L)	20 min (L)		18 min (L)	17 min (L)		-	-	+20 Celsius	>27 J
S355JO	22 min (L)		21 min (L)	20 min (L)		18 min (L)	17 min (L)		-	-	0 Celsius	27 J
S355J2	22 min (L)		21 min (L)	20 min (L)		18 min (L)	17 min (L)			-	-20 Celsius	27 J
S355K2	22 min (L)		21 min (L)	20 min (L)		18 min (L)	17 min (L)			-	-20 Celsius	40 J
S355N	22 min (Longitudinal)		21 min (Longitudinal)						-	6 min	-20 Celsius	40 J
S355NL	22 min (Longitudinal)		21 min (Longitudinal)						-	6 min	-50 Celsius	27 J
S355M	22 min (Longitudinal)						-	-	-	6 min	-20 Celsius	40 J
S355ML	22 min (Longitudinal)						-	-	-	6 min	-50 Celsius	27 J
S355J0WP	22 min (L)		-	-	-	-	-	-	-	-	0 Celsius	27 J
S355J2WP	22 min (L)		-	-	-	-	-	-	-	-	-20 Celsius	27 J
S355J0W	20 min (Longitudinal)					18 min (L)	-	-	-	-	0 Celsius	27 J

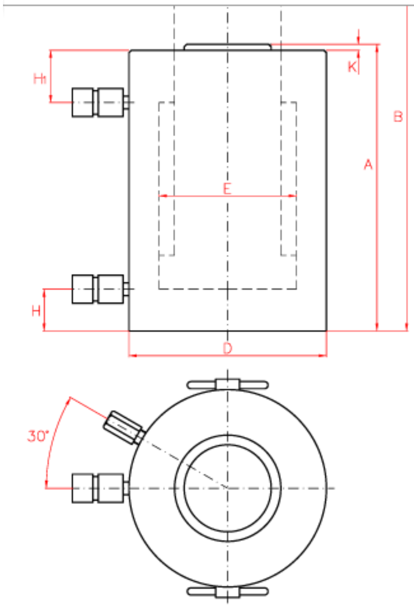
Appendix G: Larzep cylinder (K. Lund Offshore 2012)

LARZEP HYDRAULIC

Performance under Pressure

[Español] CONTACT

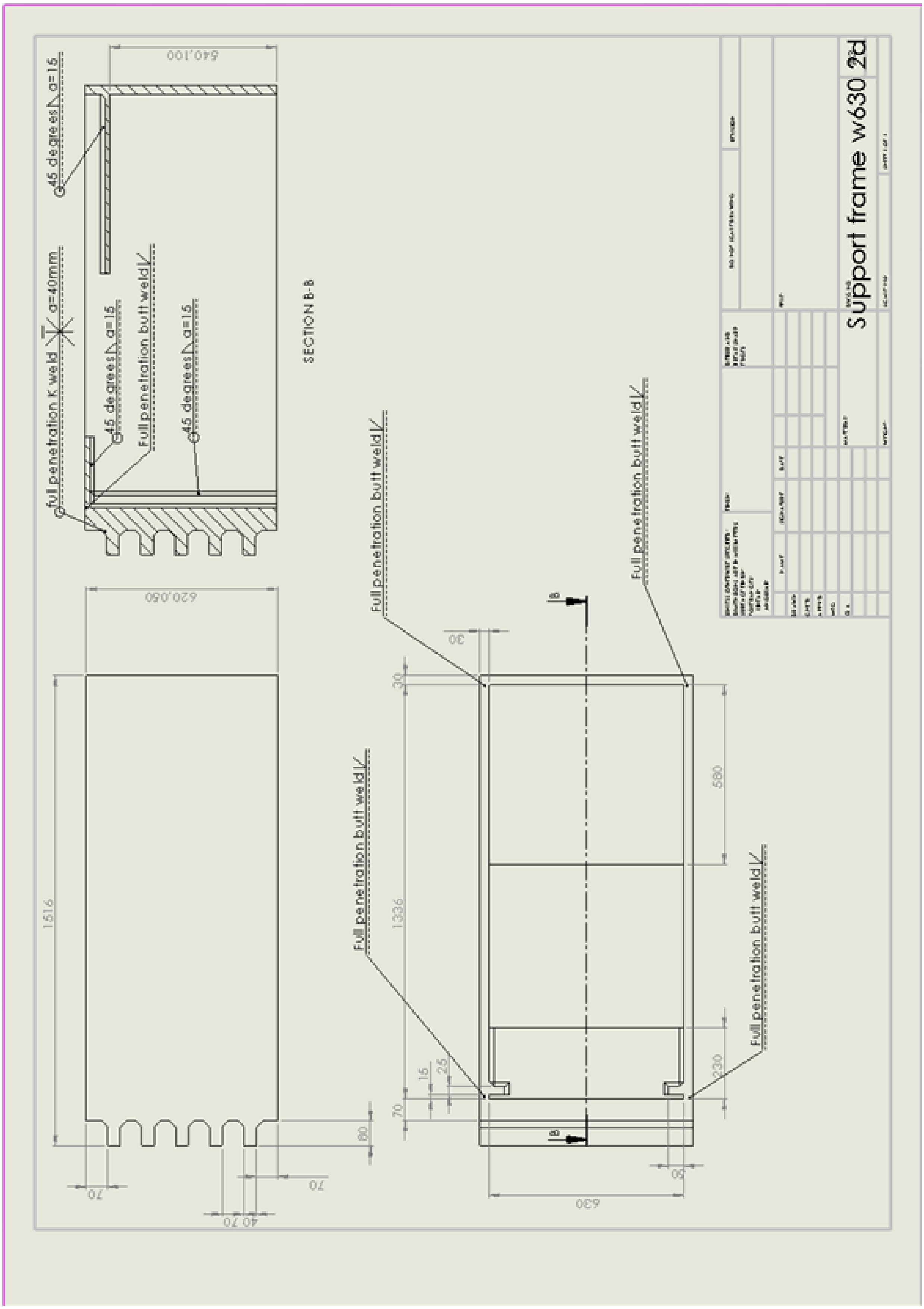
Search Site



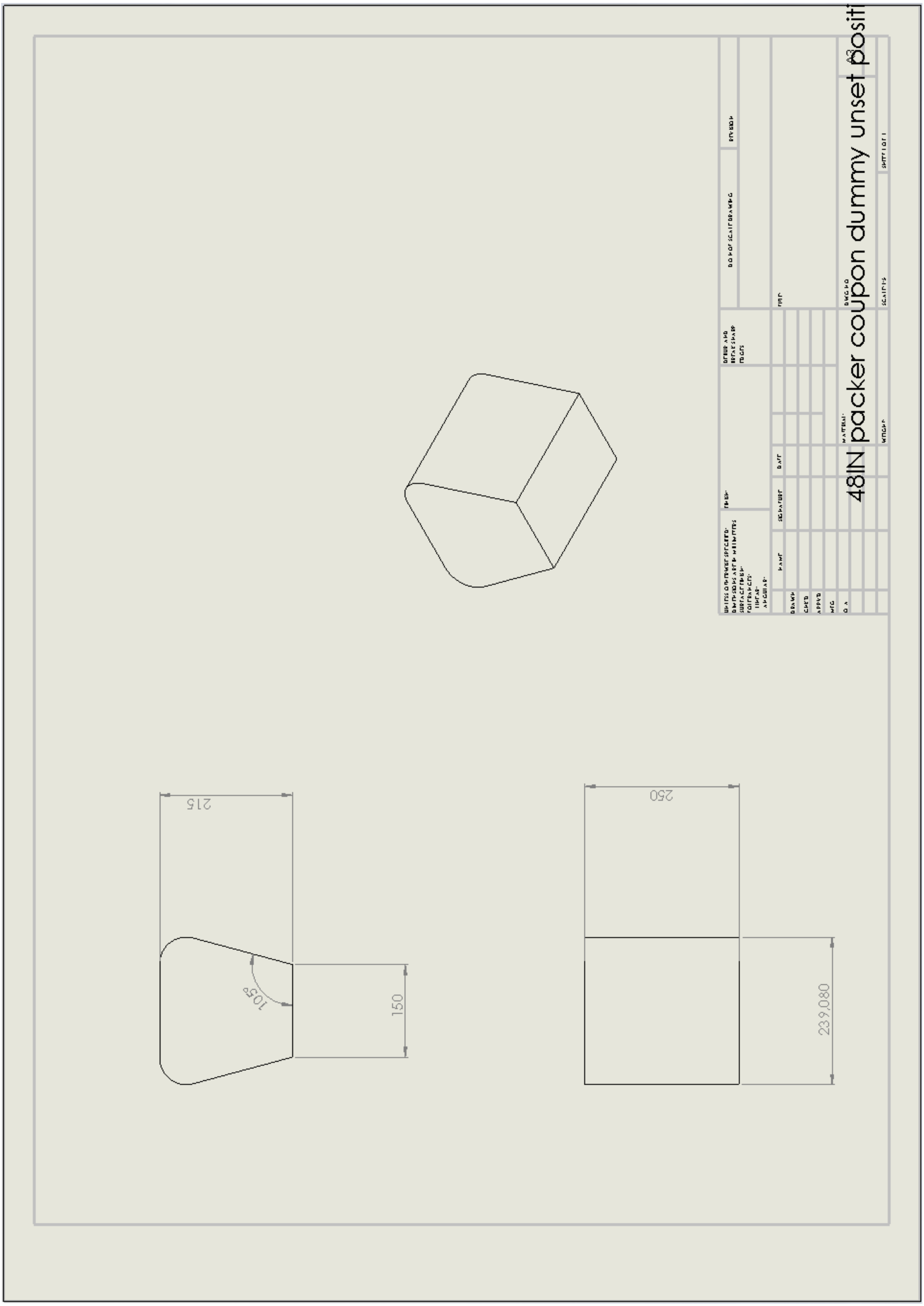
Capacity	Stroke	Model	Capacity		Oil Cap.	A	B	D	E	F	H	H1	K	L	N	Area	Weight	Model	With Tilting Saddle	
			Push	Pull																
Tn	mm	LARZEP	kN	kN	cm³	mm	mm	mm	mm	mm	mm	mm	mm	mm	mm	cm²	kg	LARZEP	A	Model
350	50	DDR35005	3.370	1.213	2.454	236	286	310	250	200	50	80	6	178	23	491	127	DDR35005	280	AZ0408
	100	DDR35010	3.370	1.213	4.909	286	386	310	250	200	50	80	6	178	23	491	148	DDR35010	330	AZ0408
	150	DDR35015	3.370	1.213	7.363	341	491	310	250	200	50	80	6	178	23	491	175	DDR35015	385	AZ0408
	200	DDR35020	3.370	1.213	9.817	391	591	310	250	200	50	80	6	178	23	491	195	DDR35020	435	AZ0408
	250	DDR35025	3.370	1.213	12.272	451	701	310	250	200	50	85	6	178	23	491	222	DDR35025	495	AZ0408
	300	DDR35030	3.370	1.213	14.726	501	801	310	250	200	50	85	6	178	23	491	243	DDR35030	545	AZ0408

Appendix H: SolidWorks drawings

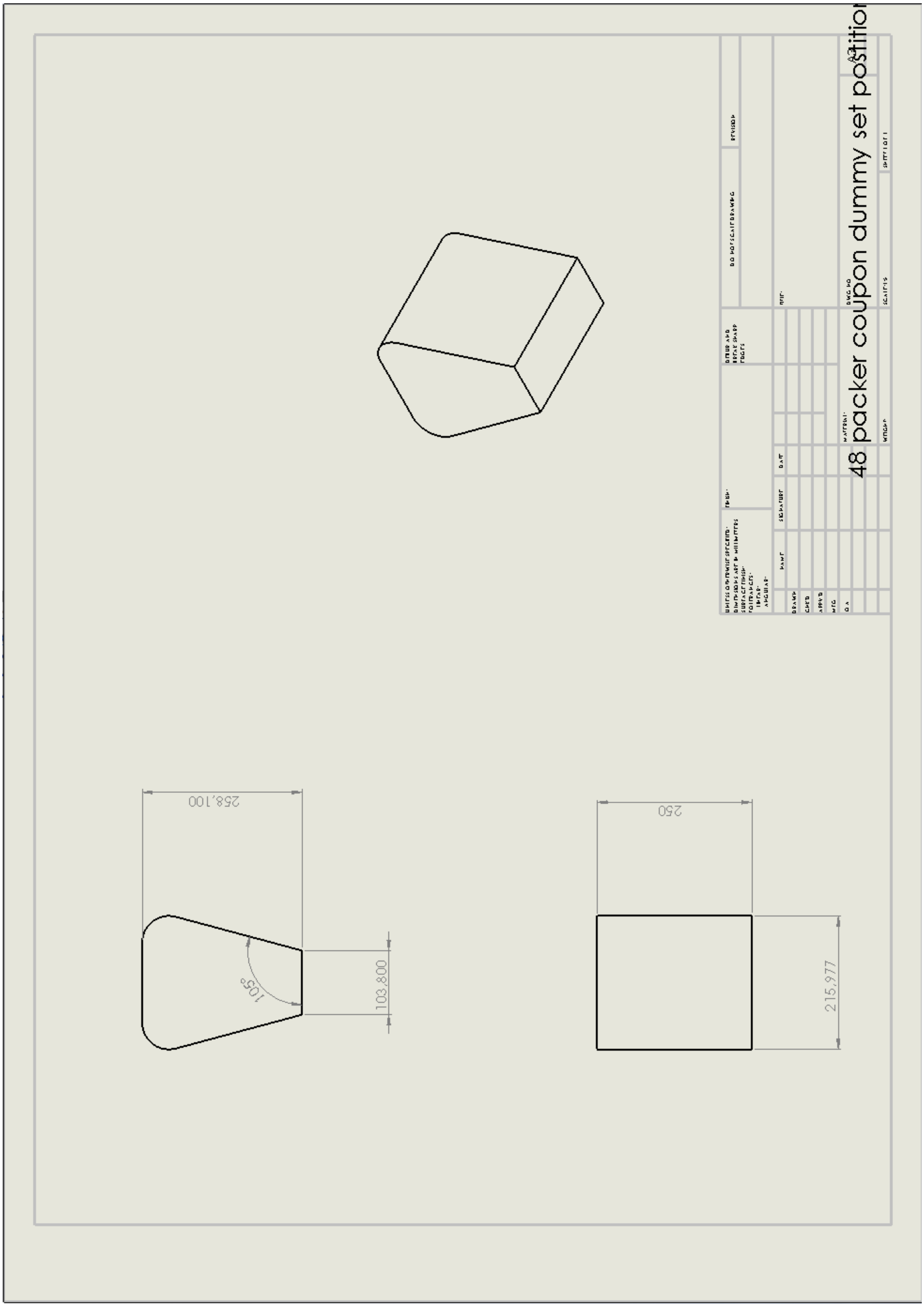
H1: Support frame



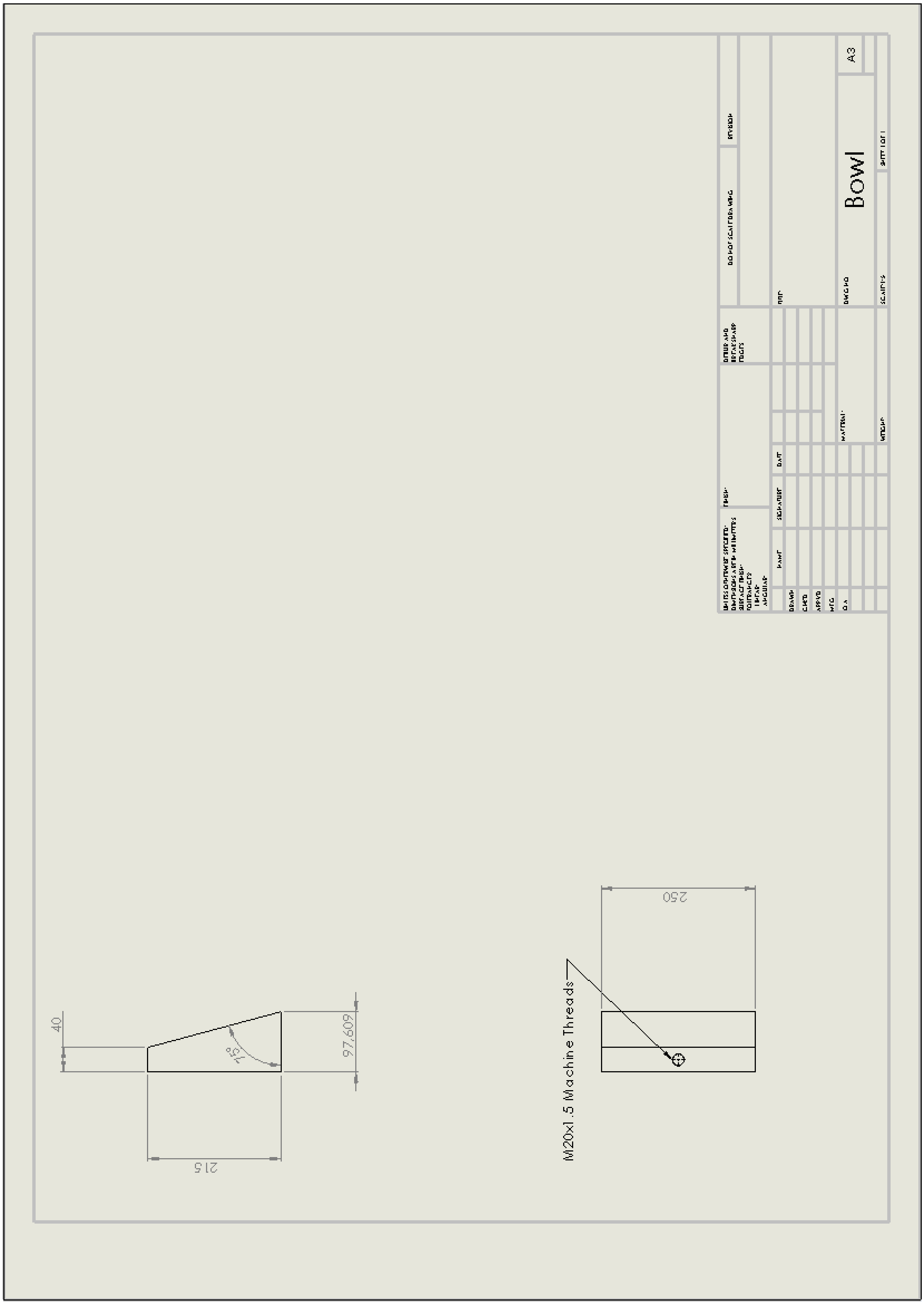
H2: Pakcer coupon dummy unset position



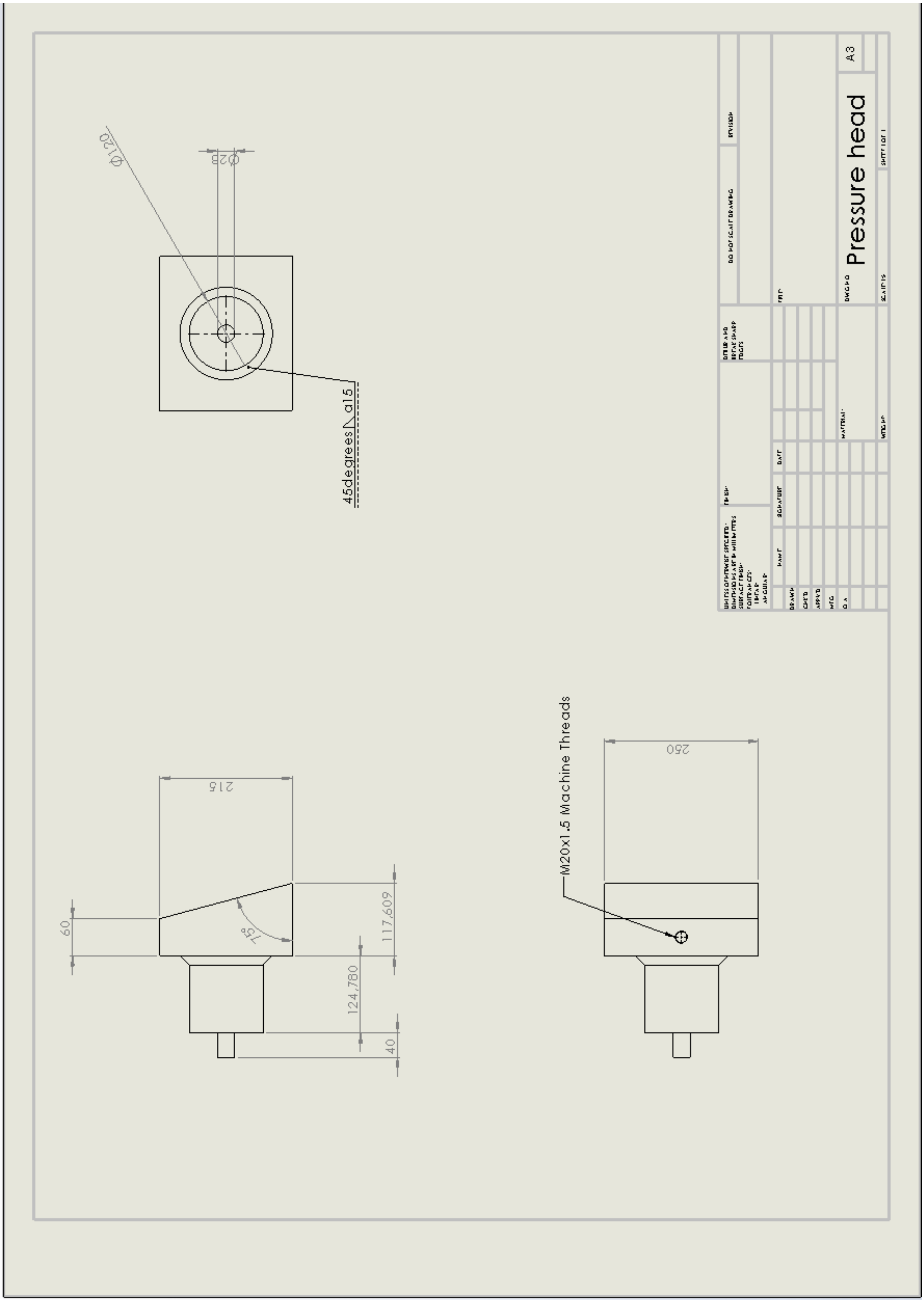
H3: Pakcer coupon dummy set position



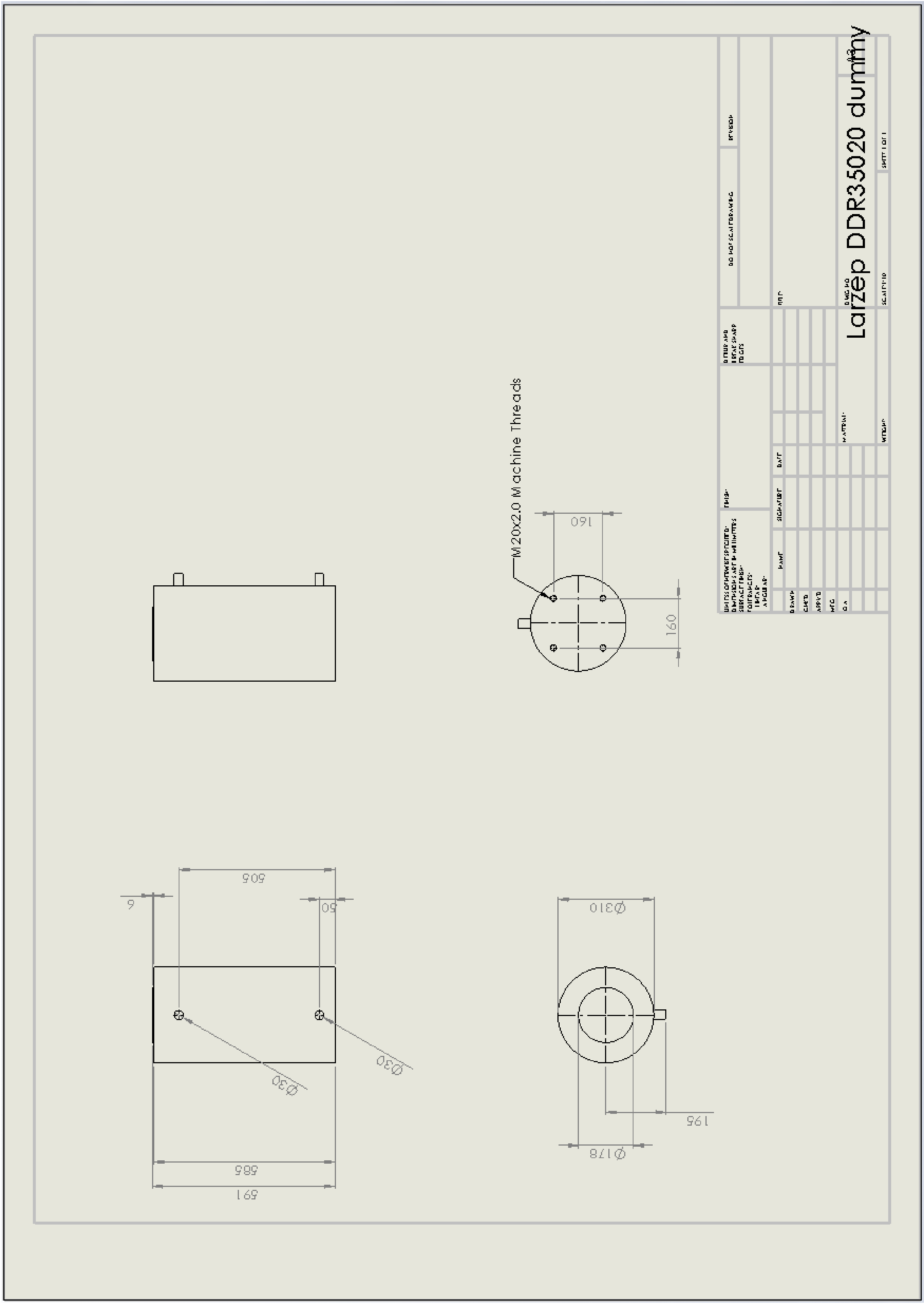
H4: Bowl



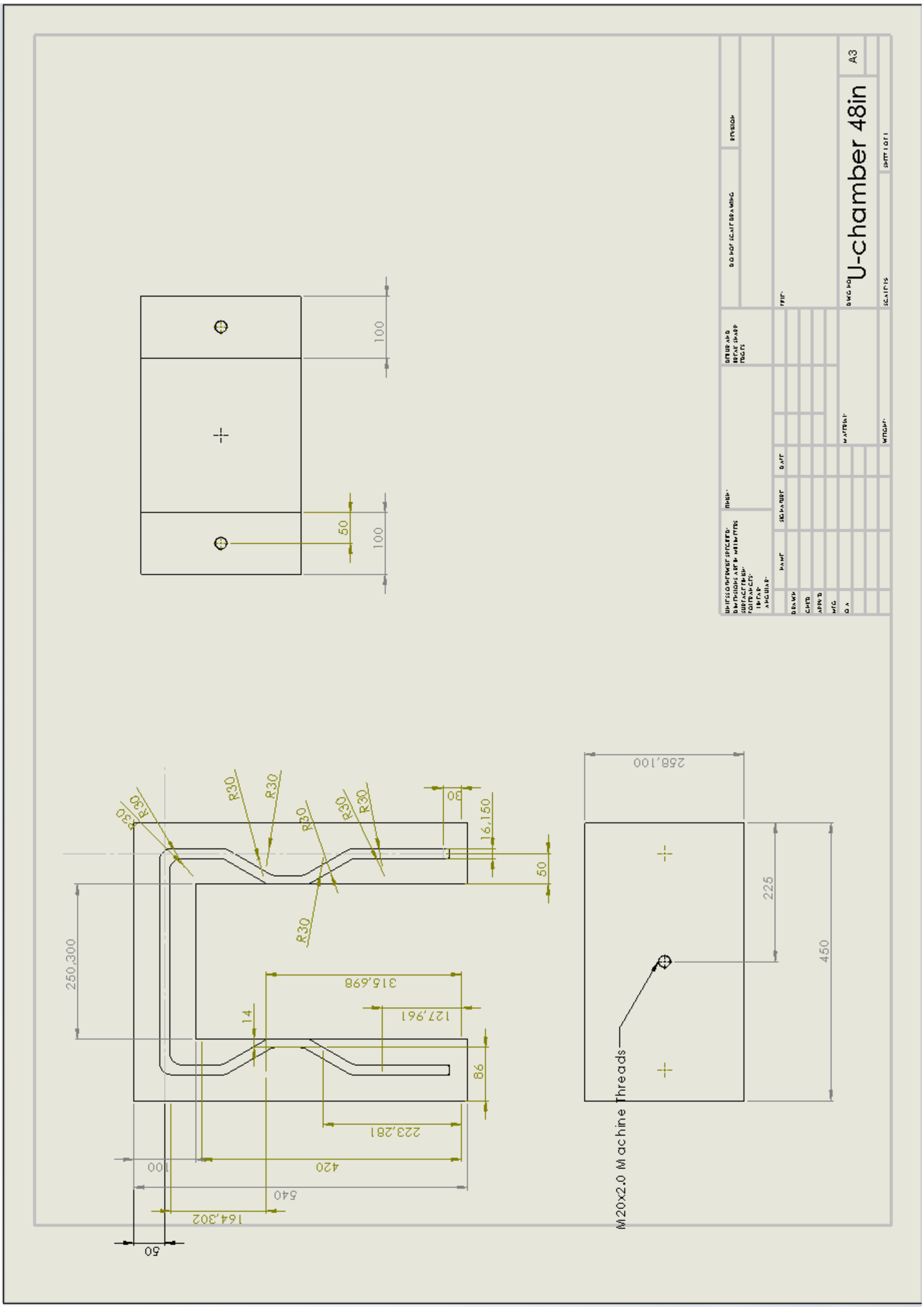
H5: Pressure head



H6: Larzep DDR35020 dummy

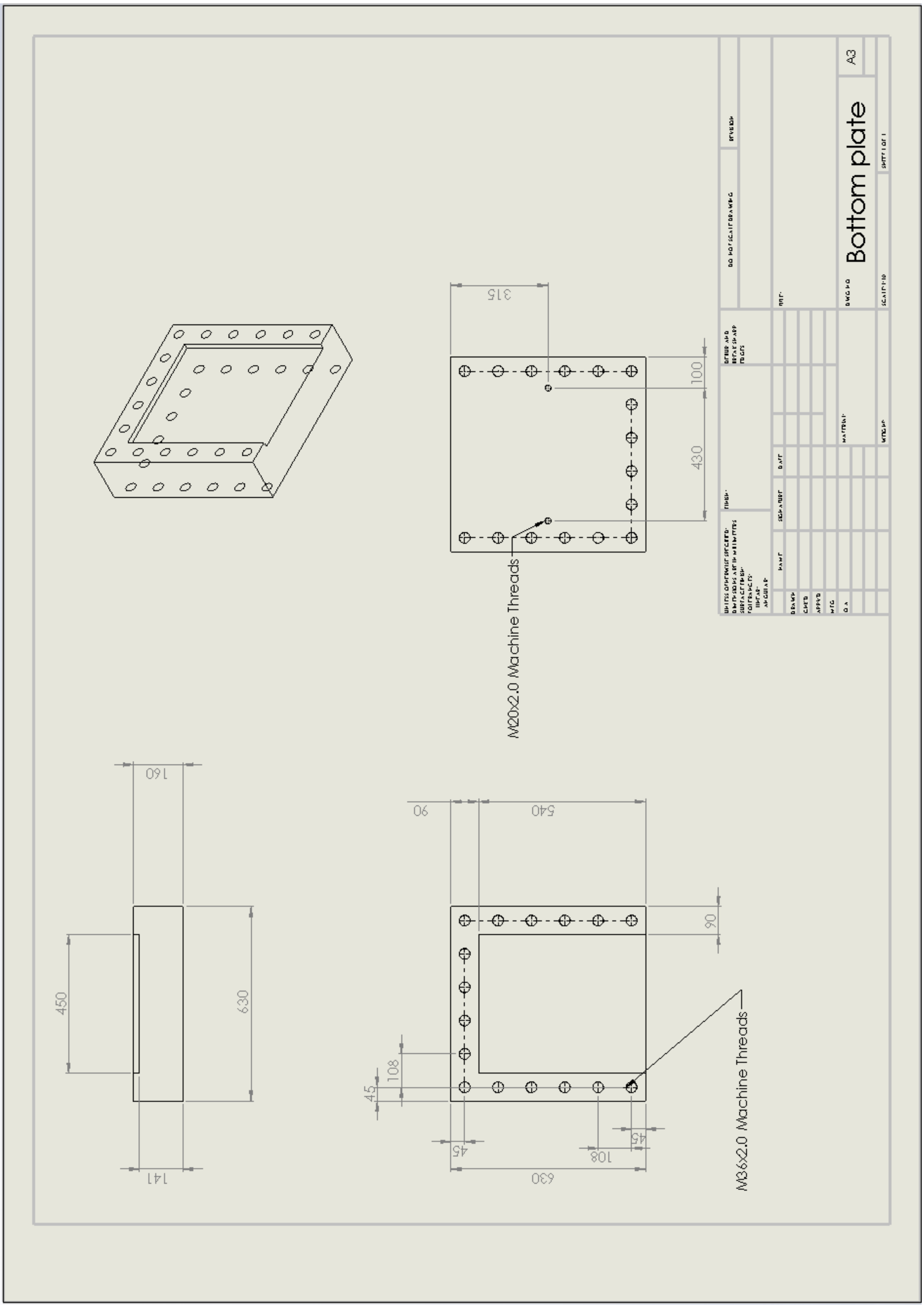


H7: U-chamber

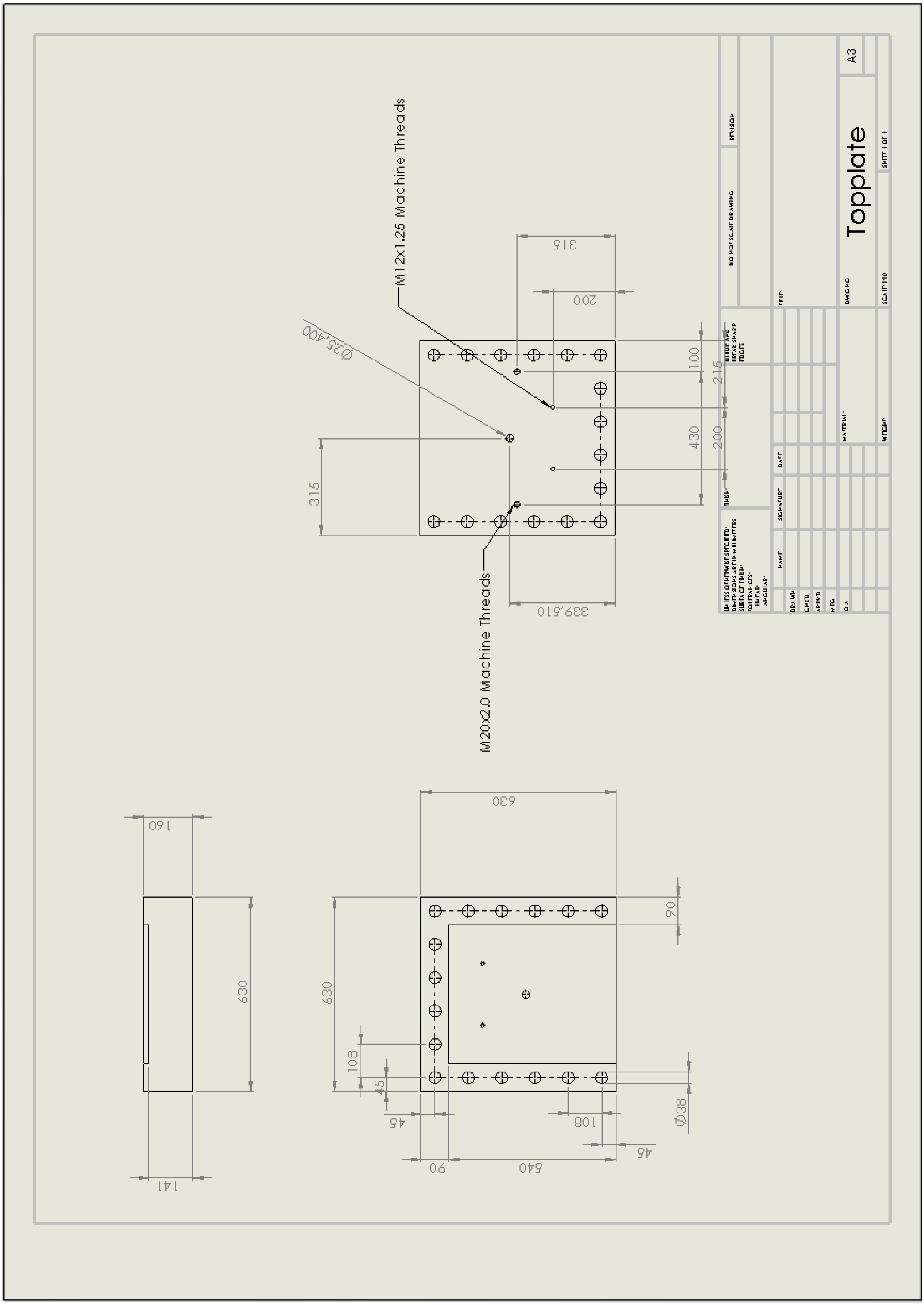


U-chamber 48in
A3
M20x2.0 Machine Threads
R30
100
50
258,100
225
450
250,300
315,698
127,961
14
86
223,281
420
540
164,302
16,150
30

H8: Bottom plate

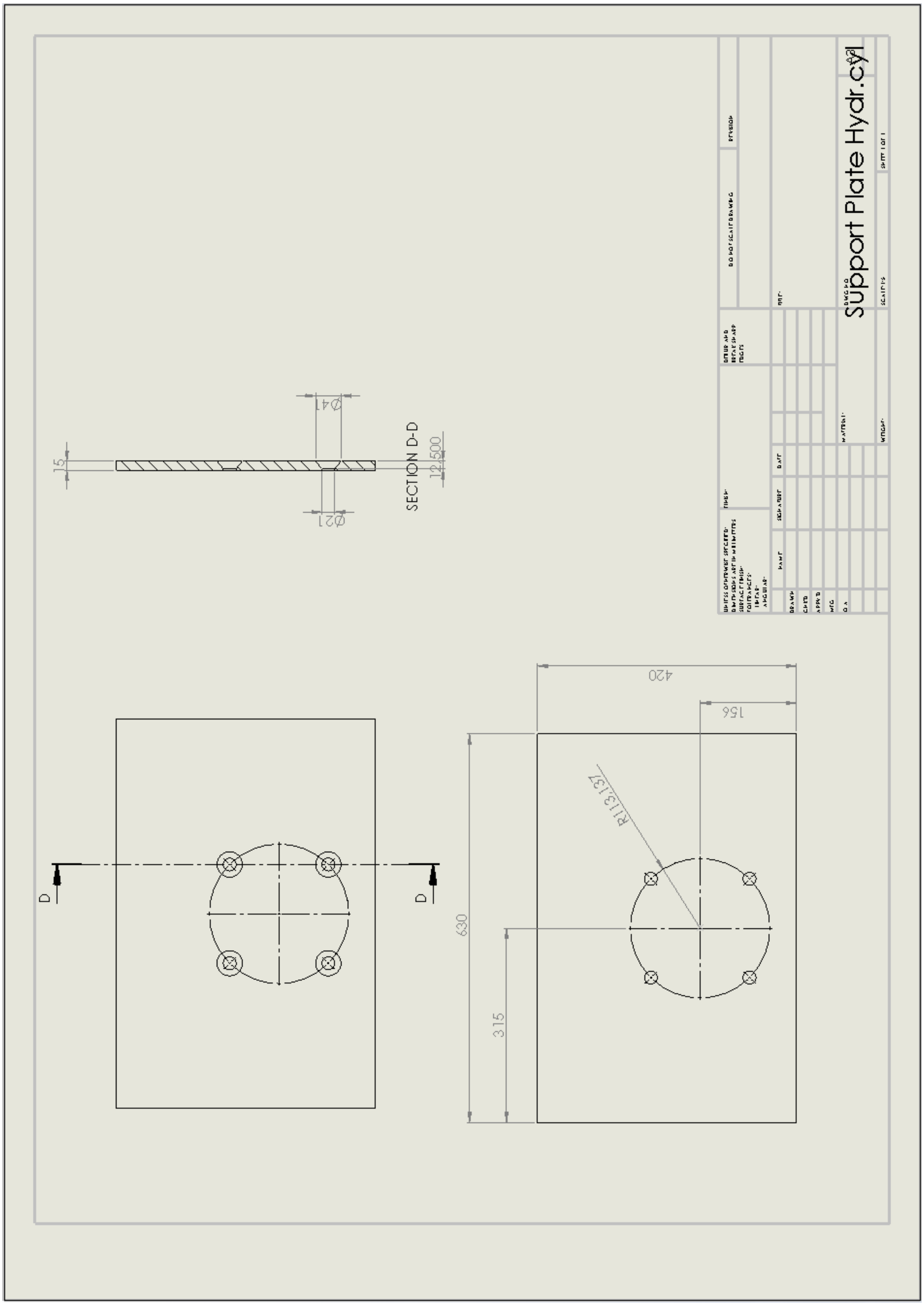


H9: Top plate

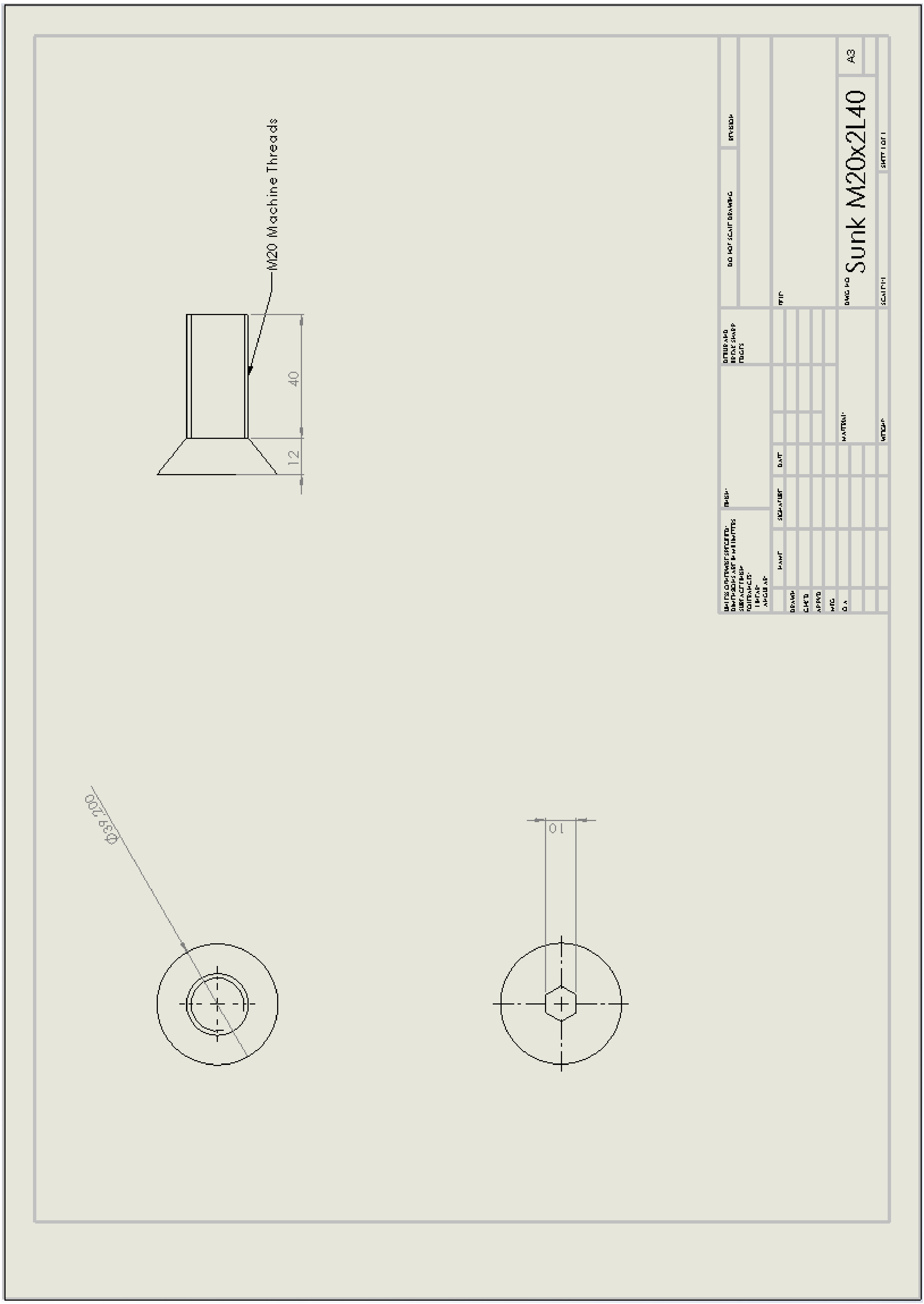


Topplate
A3
M12x1.25 Machine Threads
M20x2.0 Machine Threads
Ø25,400
315
200
100
215
430
200
315
339,510
630
630
630
160
141
90
108
45
45
108
Ø38
45
90
540

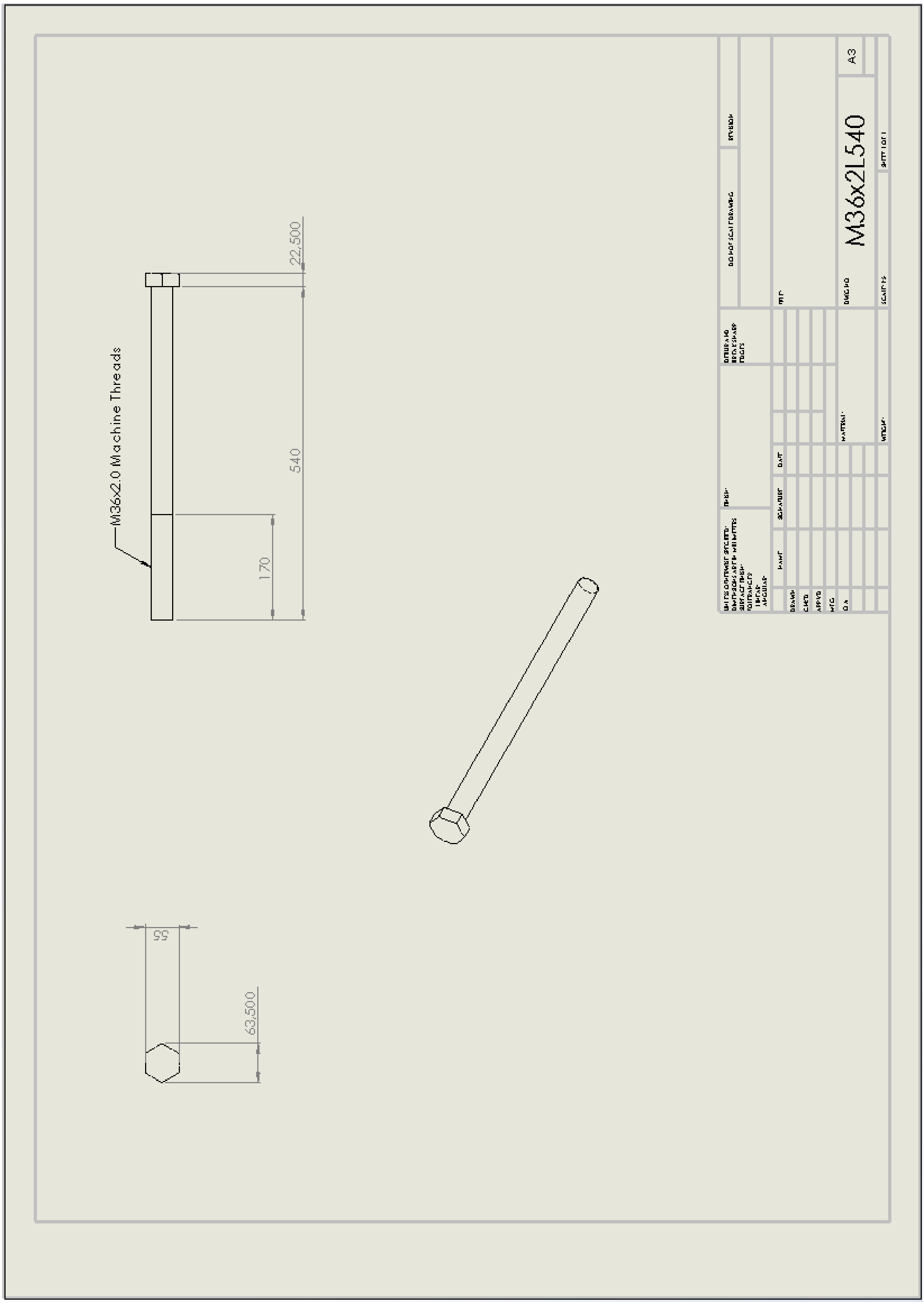
H10: Support plate hydraulic cylinder



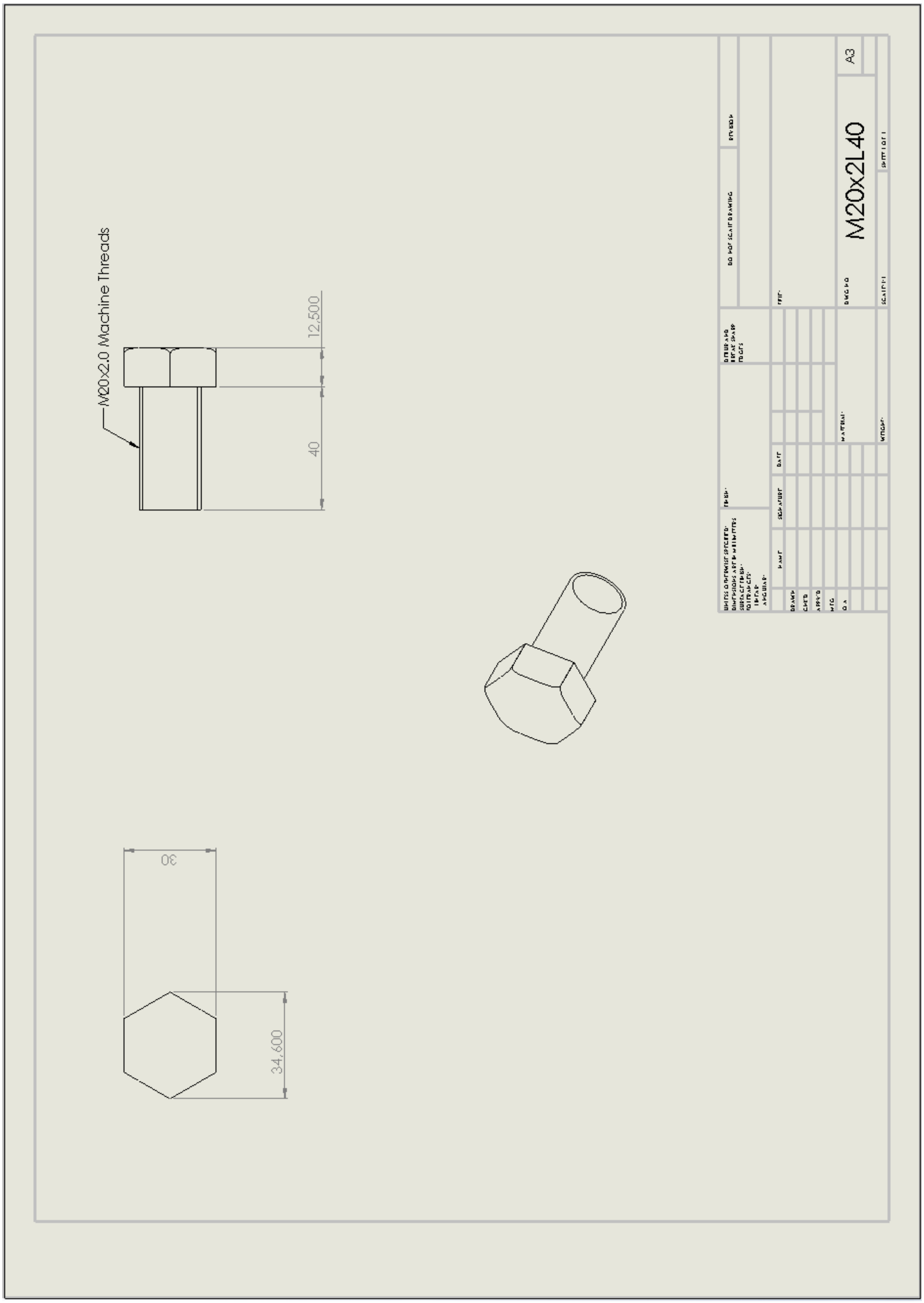
H11: Countersunk M20x2 L40 bolt



H12: M36x2 L540 hexagonal bolt



H13: M20x2 L40 hexagonal bolt



Appendix I: Cost estimate on test rig parts from Maskinering & Sveiseservice AS

Maskinering & Sveiseservice AS

Telefon: 51 77 03 00 postmaster@msas.no
Telefax: 51 77 03 01 www.msas.no
Org.nr. 977 066 719 MVA IBAN.
Konto nr. 3201.39.04463 Swift. Sprono22

MASKINERING & SVEISESERVICE A/S

Tilbud 9086

Postadresse	Leveringsadresse	Fakturadresse
TDW Offshore Services AS	TDW Offshore Services AS	TDW Offshore Services AS
Pb. 8011	Fabrikkveien 15	Pb. 8011
4068 STAVANGER	4068 STAVANGER	4068 STAVANGER

Vår ref: Gabriel Pollestad
Deres ref: Tore Gundersen
Referanse: Deres forespørsel fra Svein Tore Jakobsen

Bet.bet. netto pr. 30 dg
Lev.bet. FCA
Endrefrist:
Kundenr: 10186

Tilbuds Dato: 23.03.2012
Fakturadato:
Forfallsdato:
Levering:

Kundeinfo:
Tlf: 51 44 32 40
Fax:
Mobil
tore.gundersen@tdwilliamson.com

Linje	Produkt beskrivelse	Antall	Pris	Rabatt	Sum
1	167734 Taper angle	1	4 900,00		4 900,00 kr
2	Topplate m/pakningspor	1	15 500,00		15 500,00 kr
3	Bunnplate m/gjenger og pakningspor	1	16 900,00		16 900,00 kr
4	166322 Cylinder 8IN	1	4 500,00		4 500,00 kr
5	110072 Cylinder 24IN	1	6 200,00		6 200,00 kr
6	167734 Cylinder 48IN	1	22 200,00		22 200,00 kr
7	Interface Cylinder	1	3 500,00		3 500,00 kr
8	166322 Piston 8IN	1	4 500,00		4 500,00 kr
9	110072 Piston 24IN	1	4 800,00		4 800,00 kr
10	167734 Piston 48IN	1	5 900,00		5 900,00 kr
11	Support frame	1	10 500,00		10 500,00 kr
12	166322 Taper angle 8IN	1	5 500,00		5 500,00 kr
13	110072 Taper angle 24IN	1	6 500,00		6 500,00 kr
14	Tilbudsbetingelser	1	0,00		0,00 kr

Leveringstid: 3-4 uker fra ordre
Overflatebehandling: Ingen
Pris forstås: Alle materialer er basert på Karbonstål S355J2+N

Med forbehold om mellomsalg. Generelle leveringsbetingelser iht. NL01.
Tilbudet er gyldig i 30 dager.

Netto totalt ekskl. mva	111 400,00 kr
Mva	27 850,00 kr
Total inkl. mva.:	139 250,00 kr

Appendix J: Cylinder price offers

J1: Hytorc as

Powerteam cylinders, 35% company discount:

Duble acting cylinders (Length in inch):

- RD2006 NOK: 18756,00kr
- RD2806 NOK: 35970,00kr
- RD28010 NOK: 34505,49kr
- RD3556 NOK: 42283,74kr
- RD3006 NOK: 45269,92kr
- RD30013 NOK: 55821,50kr

Single acting cylinders (Length in inch):

- R2006C NOK: 17427,00kr
- R2006L NOK: 17569,00kr
- R2806L NOK: 27914,00kr

J2: K. Lund Offshore as

Larzep cylinders, 40% company discount:

DDR Double acting cylinders, high tonnage (Length in cm):

- DDR35020 NOK: 56228kr
- DDR35025 NOK: 62171kr

D Double acting cylinders (Length in cm):

- D35015 NOK: 75658kr
- D35030 NOK: 93829kr

SM Single acting spring return cylinders (Length in cm):

- SM22015 NOK: 22700kr

Appendix K: Packer coupon price offer from Rubberstyle AS

Hi,

I've looked at your drawings and give you a price estimate on mold cost and production cost as follows:
Molds: 30,000 NOK per mold x 3 = 90,000 NOK for all 3.
Production: I want to cast this on an hourly basis. 800 NOK per hour for the press and 600 NOK per man-hour. You can expect two press hours per man-hour. The rubber we use, is estimated to be 100 NOK per kg. You must keep the springs.

Call me or visit me if something is unclear.

Regards

Med vennlig hilsen

RUBBERSTYLE AS

Kjell Ivar Ueland

Key Account Manager

Direktetelefon: 911 41 347

Sentralbord: 51 54 28 00

Telefaks: 51 54 25 00

E-post: kjell.ivar@rubberstyle.com

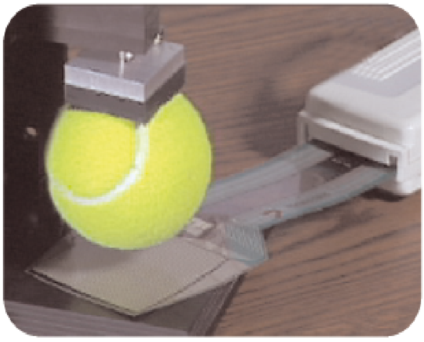
www.rubberstyle.com

Appendix L: I-Scan piezoelectric pressure recording

Tekscan

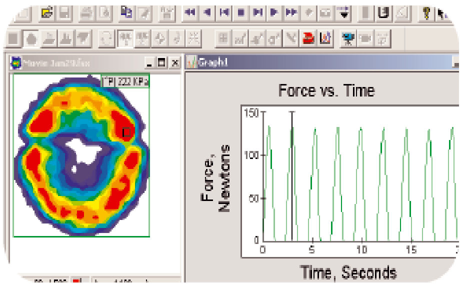
The I-Scan® System

Tactile Force and Pressure Measurement System



I-Scan, the user-friendly force and pressure measurement system, displays, records, and saves static and dynamic pressure data directly to your PC. The system includes software, interface electronics, and sensors. With a choice of over 200 available sensors in a variety of shapes and sizes, this versatile system is tailored to meet your application needs.

I-Scan's patented technology has played a key role in research and development, test and measurement, and quality control applications worldwide. Our thin, flexible sensors are minimally disruptive to the true pressure pattern and fit almost any application. These characteristics, coupled with our sensors' high spatial resolution (~1,600 sensels/in² or 248 sensels/cm²) have made us the industry leader in solving difficult pressure measurement problems.



Force vs. Time Graph

One frame of an I-Scan pressure "movie"

The *I-Scan* system provides advanced, yet simple to use, software that displays contact pressure data in real-time. Data can be captured, saved, easily analyzed using a variety of graphs, or exported as an ASCII file to be used with other programs. Tekscan's systems have saved companies millions of dollars in design, design verification, and reengineering costs.

Applications:

- Test and measurement
- Research and development
- Machine set-up
- Quality control
- Automotive
- Brake pad and friction plates
- Catalytic converter
- Hard gaskets and bolted joints
- Soft seals
- Hose clamps and crimps
- Grip and ergonomic
- Fuel cell stack assembly
- Fastener
- Nip and pinch rollers
- Wafer and glass polishing
- Lamination
- Liquid crystal display processing
- Mold filling
- Pressure garments
- Robotics
- Nozzle spray patterns
- Packaging and sealing
- Squeegee balancing
- Railroad

Key Features:

- Pressure mapping
- Dynamic recording and playback
- Graphing and analysis capabilities
- Real-time viewing
- Large variety of ready-to-use sensors
- High spatial resolution
- Flexible, thin-film sensors
- Sensors are durable and reusable
- Sampling rates: 0-127 Hz (over 830,000 sensels/sec)
- Pressure ranges: 0-25,000 PSI (0-175 MPa)
- 8-bit pressure resolution
- Quality engineering support

Found in:

Research & Development Labs, Manufacturing, Test Facilities

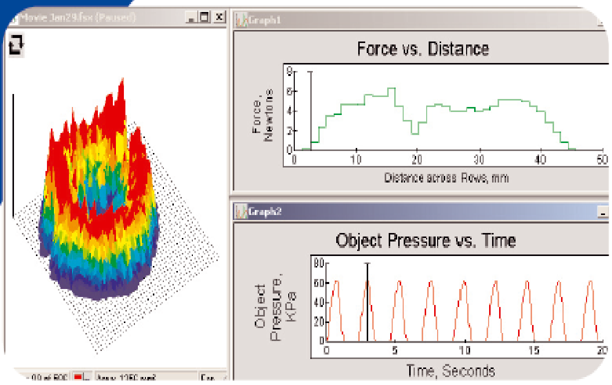
Industries:

Automotive, Semiconductor, Pharmaceutical, Ergonomics, Packaging, Paper, Printing, Government Agencies, Universities, and many more...

Specifications and Features

Software Features:

- Display real-time and recorded data as 2-D and 3-D images
- Capture dynamic pressure data
- Play-back pressure "movies"
- Display data frame-by-frame or as a multi-frame movie
- Graph and analyze real-time or stored data (Pressure, Force, Area)
- Export ASCII file capability
- Isolate and analyze specific areas
- Display Center of Force and its trajectory
- View and compare multiple tests simultaneously
- And much, much more!



Software Display

3-D contour display of a tennis ball pushing on a sensor
Also shown graphically: Pressure vs. Time and
Force vs. Distance across the sensor rows

Sensor Description:

Below is a summary of our sensors' characteristics. Sensors can be selected based on application requirements.

No. of Sensing Elements
Typically 2,288 (to over 146,000)

Spatial Density
Up to 1,600 sensels/in²
(248 sensels/cm²)

Spatial Resolution
0.025 x 0.025 in - 0.7 x 0.7 in
(0.64 x 0.64 mm - 17 x 17 mm)

Operating Temperature
15°F - 140°F (-9°C - 60°C)

Size of Sensing Area
0.12 x 0.12 in - 22.7 x 34.8 in
(3.0 x 3.0 mm - 578 x 884 mm)

Technology
Resistive

Calibration
With application of a controlled force by user

Pressure Range
0 - 25,000 PSI (0-175 MPa)

Sensor Thinness
Typically 0.004 in (0.1 mm)

Add-On Options

Virtual System Architecture™ (*VSA*) - Larger areas are easily accommodated with our *VSA* software solution. *VSA* allows you to view multiple sensors, positioned adjacent to one another, creating a continuous measurement region.

Video Synch™ - Video sequences can be recorded and synchronized with your pressure data and visualized in Tekscan software, enhancing the utility of collected data.

API (Application Program Interface) - API software enables a user, with programming knowledge, to write programs that directly access Tekscan sensors and electronics or our sensor data buffers.

Matlab® Interface - Allows you to export our standard file format into the *Matlab* environment for greater flexibility in analysis.

Equilibration/Calibration Devices- Pneumatic devices apply a uniform pressure to the active area of a sensor to normalize output of each sensing element. The system electronically compensates for variation in individual sensing elements.

Call Today for a Demonstration!

Tekscan, Inc.
307 West First Street
South Boston, MA 02127-1309 USA
tel: 617.464.4500/800.248.3669
fax: 617.464.4266
e-mail: marketing@tekscan.com
website: www.tekscan.com

RevC_080604

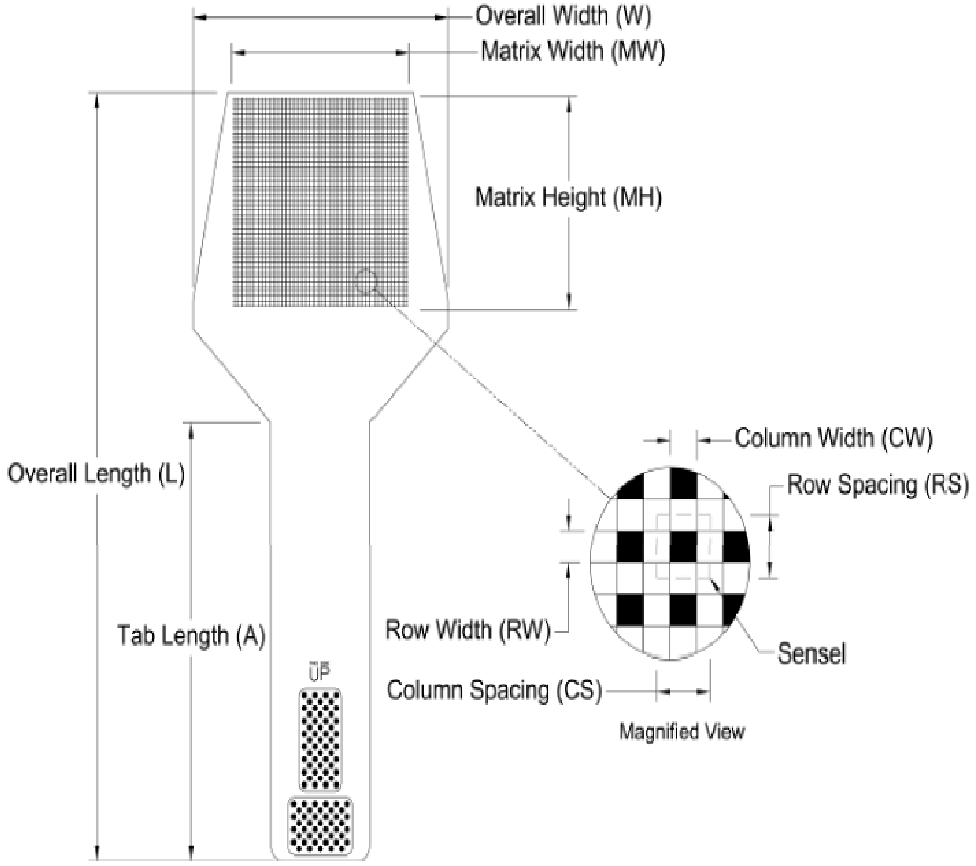
Tekscan

Sensor Model: 5051/5076/5101

Application Example: Excellent for general purpose use

Features:

- Wide range of available pressures
- Internal vents



Model	Overall Length L	Overall Width W	Tab Length A	Matrix Width MW	Matrix Height MH	Columns CW	Columns Pitch CS	Columns Qty.	Rows RW	Rows Pitch RS	Rows Qty.	Total No.of Sensels	Resolution Sensel Density
	General Dimensions			Sensing Area Dimensions									
US	(in.)	(in.)	(in.)	(in.)	(in.)	(in.)	(in.)		(in.)	(in.)			(sensel per sq. in.)
5051	9.94	3.20	6.54	2.20	2.20	0.030	0.050	44	0.030	0.050	44	1936	400.0
5076	12.03	4.78	6.93	3.30	3.30	0.040	0.075	44	0.040	0.075	44	1936	177.8
5101	13.39	5.86	6.59	4.40	4.40	0.050	0.100	44	0.050	0.100	44	1936	100.0
Metric	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)		(mm)	(mm)			(sensel per sq. cm)
5051	252.5	81.3	166.2	55.9	55.9	0.8	1.3	44	0.8	1.3	44	1936	62.0
5076	305.5	121.3	175.9	83.8	83.8	1.0	1.9	44	1.0	1.9	44	1936	27.6
5101	340.0	149.0	167.3	111.8	111.8	1.3	2.5	44	1.3	2.5	44	1936	15.5

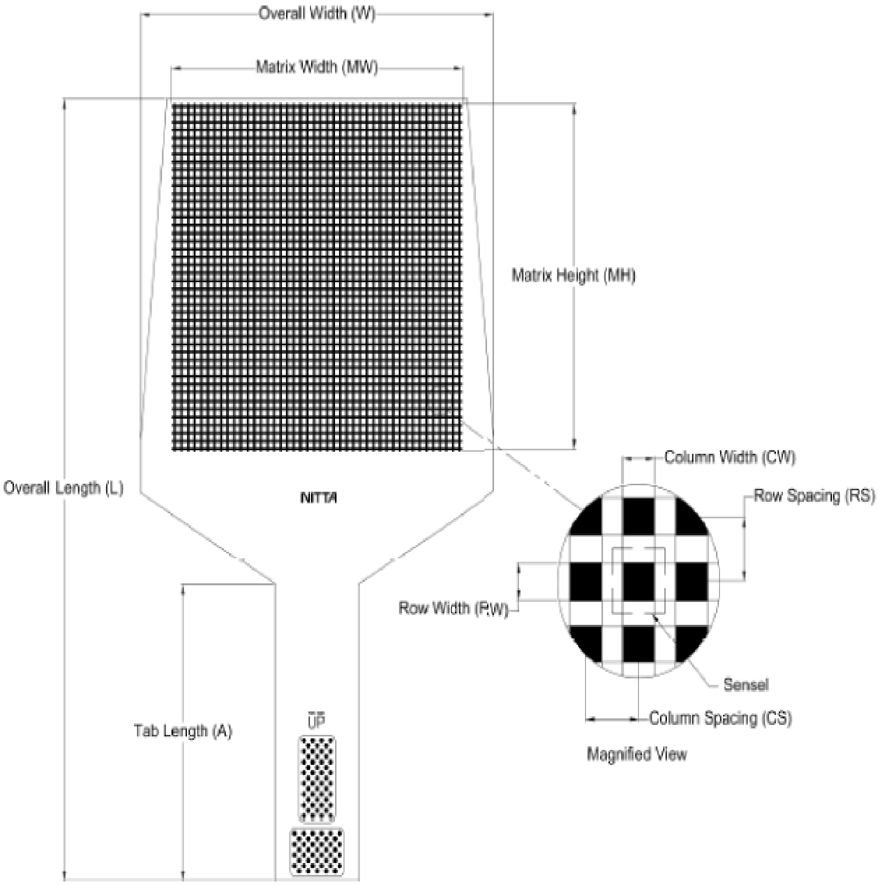
307 West First St., South Boston, MA 02127
Tel: 617.464.4500/800.248.3669 fax: 617.464.4266 web: www.tekscan.com

SENSOR MODEL: 5040N/5150N/5210N

Application Example: Excellent for general purpose use

Features:

- Wide range of available pressures
- Internal vents



	General Dimensions			Sensing Area Dimensions		Columns			Rows				Resolution
Model	Overall Length L	Overall Width W	Tab Length A	Matrix Width MW	Matrix Height MH	CW	Pitch CS	Qty.	RW	Pitch RS	Qty.	Total No.of Sensels	Sensel Density
US	(in.)	(in.)	(in.)	(in.)	(in.)	(in.)	(in.)		(in.)	(in.)			(sensel per sq. in.)
5040N	14.57	3.05	11.57	1.73	1.73	0.020	0.039	44	0.020	0.039	44	1936	645.2
5150N	12.60	7.87	3.54	6.50	6.50	0.098	0.148	44	0.098	0.148	44	1936	45.9
5210N	21.26	11.18	7.87	9.35	9.35	0.118	0.213	44	0.118	0.213	44	1936	22.1
Metric	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)		(mm)	(mm)			(sensel per sq. cm)
5040N	370.0	77.5	294.0	44.0	44.0	0.5	1.0	44	0.5	1.0	44	1936	100.0
5150N	320.0	200.0	90.0	165.0	165.0	2.5	3.8	44	2.5	3.8	44	1936	7.1
5210N	540.0	284.0	200.0	237.6	237.6	3.0	5.4	44	3.0	5.4	44	1936	3.4

307 West First St., South Boston, MA 02127
Tel: 617.464.4500/800.248.3669 fax: 617.464.4266 web: www.tekscan.com

Appendix M: Price offer on I-Scan system from CA Mätsystem AB

CA MÄTSYSTEM
CHAUVIN ARNOUX GROUP

Täby den 24 januari 2012

University Of Stavanger

Sven Tore Jakobsen

OFFERT NR: 12702/RN **Vår ref: Roger Nilsson**

Refererande till vårt telefonsamtal idag har vi nöjet att offerera följande:

Pos	Antal	Benämning
1	1	**TEKSCAN typ-Iscan** System för yttrycksanalys bestående av: Programvara för Windows XP, Vista eller 7 Hårdvara att ansluta till USB-2-port på befintlig PC. Anslutning, handtag för givare. Drivrutiner ingår för hantering av givare 5076 och 5150N. Eller 2 från valfri familj. Dessutom ingår 5 st 5076 givare. Fri telefonsupport av CA Mätsystem AB och 1 års garanti Installation och utbildning ingår. **Pris 230 000 Sek**
1	10	Givare typ 5150N för yttrycksanalys bestående av: **Pris/st 2 200 Sek**

Betalningsvillkor: 30 dgr netto, efter sedvanlig kreditprövning
Leveranstid: 2-3 veckor
Offertens giltighetstid: 1 mån

Offererade priser gäller för Er netto, fritt vårt lager i Täby, exklusive mervärdesskatt. Vi reserverar oss för prisändringar förorsakade av förändringar i valutaläget, statliga pålagor eller dylikt. I övrigt tillämpas IMs leveransvillkor. CA Mätsystem AB är certifierat enligt ISO 9001.
Vid köp av nya produkter tar vi emot gamla i enlighet med Elektronikförordningen.

Med vänlig hälsning
CA Mätsystem AB

Telefon 08-50 52 68 00
Fax 08-50 52 68 10

Postadress CA Mätsystem AB,Box 4501,183 04,Täby
Huvudkontor Sjöflygvägen 35, 183 62,Täby
e-mail info@camatsystem.com
www.camatsystem.com

Org.nr 556273-6719
Certifiering ISO 9001

Appendix N: Standard bolt dimensions (Euler, Standard bolt dimensions 2002)

Table 1 Standard Metric and USA Bolt Shank Dimensions (All dimensions in units of mm or mm^2 unless otherwise specified.)

			Coarse Pitch Threads						Fine Pitch Threads					
Size Desig- nation	Nominal (Major) Diam- eter, D_n	Nominal Shank Area, A_n	Threads per inch, n	Pitch (mm per thread), p	Pitch Dia- meter, d_p	Minor Dia- meter Area, A_s	M, UNC, UNRC Tensile Stress Area, A_{ts}	MJ, UNJC Tensile Stress Area, A_{ts}	Threads per inch, n	Pitch (mm per thread), p	Pitch Dia- meter, d_p	Minor Dia- meter Area, A_s	M, UNF, UNRF Tensile Stress Area, A_{ts}	MJ, UNJF Tensile Stress Area, A_{ts}
0	1.5240	1.8241							80	0.31750	1.3178	0.9704	1.1588	1.3639
M1.6	1.6000	2.0106		0.35000	1.3727	1.0762	1.2700	1.4799						
1	1.8542	2.7002	64	0.39688	1.5964	1.4074	1.6915	2.0016	72	0.35278	1.6251	1.5304	1.7920	2.0741
M2	2.0000	3.1416		0.40000	1.7402	1.7890	2.0732	2.3784						
2	2.1844	3.7476	56	0.45357	1.8898	1.9986	2.3847	2.8049	64	0.39688	1.9266	2.1874	2.5383	2.9153
M2.5	2.5000	4.9087		0.45000	2.2077	2.9801	3.3908	3.8280						
3	2.5146	4.9662	48	0.52917	2.1709	2.6222	3.1386	3.7014	56	0.45357	2.2200	2.9116	3.3741	3.8707
4	2.8448	6.3561	40	0.63500	2.4324	3.2045	3.8922	4.6467	48	0.52917	2.5011	3.6555	4.2611	4.9130
M3	3.0000	7.0686		0.50000	2.6752	4.4734	5.0308	5.6210						
M3.5	3.5000	9.6211		0.60000	3.1103	5.9997	6.7752	7.5979						
6	3.5052	9.6497	32	0.79375	2.9896	4.8075	5.8615	7.0199	40	0.63500	3.0928	5.6424	6.5440	7.5124
M4	4.0000	12.566		0.70000	3.5453	7.7496	8.7787	9.8720						
8	4.1656	13.628	32	0.79375	3.6500	7.7166	9.0379	10.464	36	0.70556	3.7073	8.2910	9.5016	10.795
10	4.8260	18.292	24	1.0583	4.1386	9.3546	11.311	13.452	32	0.79375	4.3104	11.311	12.899	14.593
M5	5.0000	19.635		0.8000	4.4804	12.683	14.183	15.766						
12	5.4864	23.641	24	1.0583	4.7990	13.277	15.590	18.088	28	0.90714	4.8972	14.576	16.638	18.836
M6	6.0000	28.274		1.0000	5.3505	17.894	20.123	22.484						
0.2500	6.3500	31.669	20	1.2700	5.5251	17.351	20.530	23.976	28	0.90714	5.7608	21.006	23.467	26.065
0.3125	7.9375	49.483	18	1.4111	7.0210	29.267	33.826	38.715	24	1.0583	7.2501	33.826	37.462	41.284
M8	8.0000	50.265		1.2500	7.1881	32.841	36.609	40.581		1.0000	7.3505	36.030	39.167	42.435
0.3750	9.5250	71.256	16	1.5875	8.4939	43.741	49.994	56.663	24	1.0583	8.8376	52.170	56.663	61.342
M10	10.000	78.540		1.5000	9.0257	52.292	57.990	63.981		1.2500	9.1881	56.297	61.199	66.304
0.4375	11.112	96.987	14	1.8143	9.9341	60.210	68.586	77.508	20	1.2700	10.288	70.327	76.591	83.123
M12	12.000	113.10		1.7500	10.863	76.247	84.267	92.687		1.2500	11.188	86.037	92.072	98.311
0.5000	12.700	126.68	13	1.9538	11.431	81.103	91.548	102.63	20	1.2700	11.875	95.903	103.20	110.76
M14	14.000	153.94		2.0000	12.701	104.71	115.44	126.70		1.5000	13.026	116.13	124.55	133.26
0.5625	14.288	160.33	12	2.1167	12.913	104.55	117.38	130.96	18	1.4111	13.371	121.82	130.96	140.42
0.6250	15.875	197.93	11	2.3091	14.375	130.20	145.81	162.30	18	1.4111	14.958	154.86	165.13	175.74
M16	16.000	201.06		2.0000	14.701	144.12	156.67	169.74		1.5000	15.026	157.47	167.25	177.32
0.7500	19.050	285.02	10	2.5400	17.400	194.84	215.78	237.79	16	1.5875	18.019	226.65	240.62	255.00
M20	20.000	314.16		2.5000	18.376	225.19	244.79	265.22		1.5000	19.026	259.00	271.50	284.30
M22*	22.000	380.13		2.5000	20.376	281.53	303.40	326.09		1.5000	21.026	319.20	333.06	347.21
0.8750	22.225	387.95	9	2.8222	20.392	270.51	297.89	326.59	14	1.8143	21.047	310.03	328.69	347.90
M24	24.000	452.39		3.0000	22.051	324.27	352.50	381.91		2.0000	22.701	364.61	384.42	404.74
1.0000	25.400	506.71	8	3.1750	23.338	355.51	390.80	427.77	12	2.1167	24.025	402.94	427.77	453.34
M27*	27.000	572.56		3.0000	25.051	427.09	459.41	492.90		2.0000	25.701	473.22	495.74	518.79
1.1250	28.575	641.30	7	3.6286	26.218	447.18	492.44	539.88	12	2.1167	27.200	523.82	552.08	581.08
M30	30.000	706.86		3.5000	27.727	518.99	560.59	603.79		2.0000	28.701	595.96	621.20	646.97
1.2500	31.750	791.73	7	3.6286	29.393	574.10	625.23	678.55	12	2.1167	30.375	660.54	692.22	724.65
M33	33.000	855.30		3.5000	30.727	647.19	693.55			2.0000	31.701	732.83	760.80	789.29
1.3750	34.925	957.99	6	4.2333	32.175	680.06	745.09	813.09	12	2.1167	33.550	813.09	848.20	884.06
M36	36.000	1017.9		4.0000	33.402	759.28	816.72	876.26		2.0000	34.701	883.85	914.53	945.74
1.5000	38.100	1140.1	6	4.2333	35.350	834.73	906.62	981.47	12	2.1167	36.725	981.47	1020.0	1059.3