DEVELOPMENT OF FUNCTION MODEL FOR HYDRAULIC STONE RELEASE SYSTEM ON PLOUGH

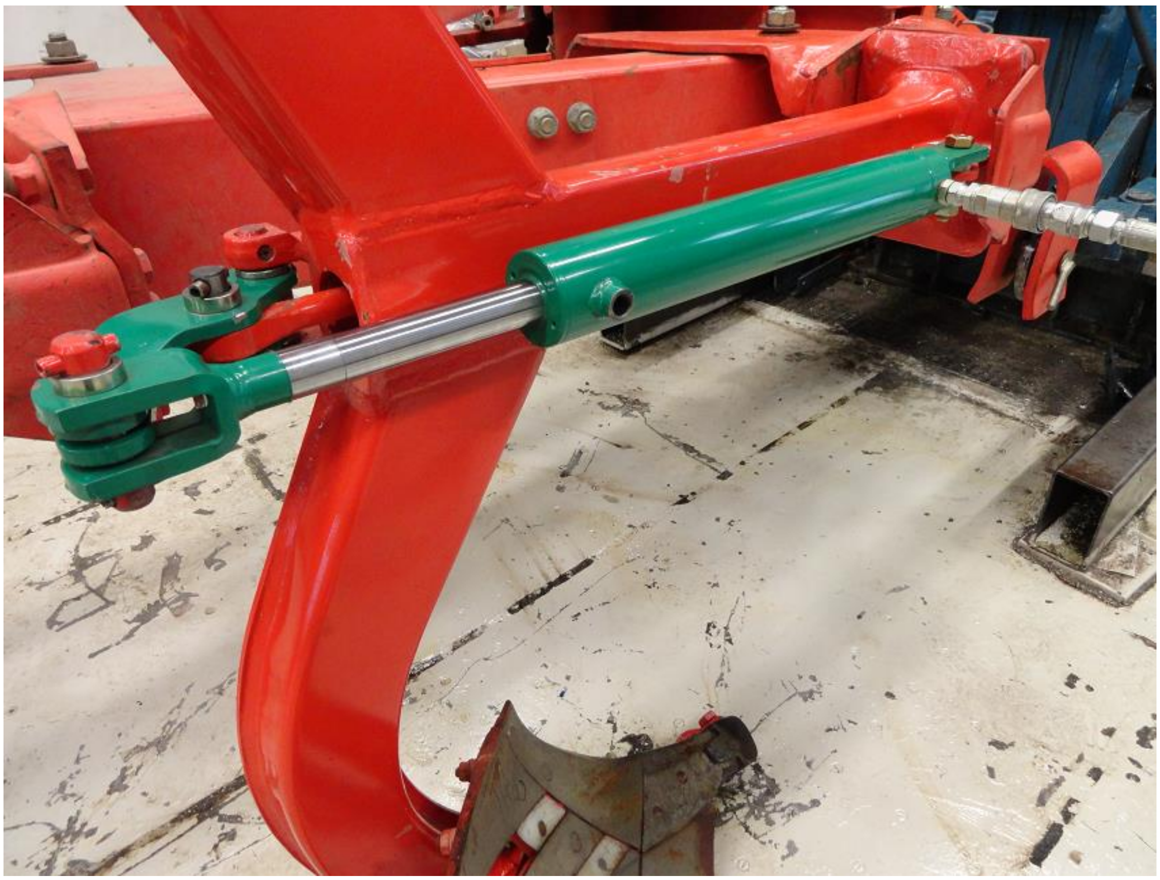
Utvikling av funksjonsmodell for hydraulisk steinutløsersystem på plog

EMIL BJØRÅS

NORWEGIAN UNIVERSITY OF LIFE SCIENCES
Department of mathematical sciences and technology
Master Thesis 30 credits 2013

DEVELOPMENT OF FUNCTION MODEL FOR HYDRAULIC STONE RELEASE SYSTEM ON PLOUGH

Master Thesis by Emil Bjørås



Norwegian University of Life Sciences

Department of mathematical sciences and technology

Spring 2013

PREFACE

This project has its foundation at the Research and Development department at Kverneland Group Klepp. From my own employment there I got the motivation to start off with this project, which has now become my Master Thesis within the Master Degree in Mechanical Engineering at Norwegian University of Life Sciences ("Universitetet for miljø- og biovitenskap"). It is a self-initiated project with the support from Kverneland Group Klepp R&D, which will also have the ownership of the product.

The project started off with a pre-study named "Kverneland Hydro Reset System", in autumn 2012. The outcome from the pre-study was a temporary product concept proposal, in which this project builds upon. In the introductory part of this report, the main findings from the pre-project is outlined, including a further evaluation of the concept that was chosen. As revealed by the pre-project, the proposed name for the prospective product is "Kverneland Hydro Reset System", but is only a temporary proposal at this point.

This project has comprised the development of a function model, where the project has been cumulating into a function test. The test has been used for evaluation of the product functionality, where the findings from the test and this project should form the basis for the prospective development of a final product.

Carrying out this study has been a very instructive process. It would never have been doable without the necessary resources in relation to testing, and guidance from other people either related to technical issues or formalities in order to carry out the study. There are several persons and institutions that I want to give gratitude:

Kverneland Group Klepp R&D with staff – technical guidance and background information, in addition to economic resources and assistance in relation with test

Per Gunnar Kraggerud – technical adviser, Kverneland Group Klepp R&D

Kverneland Group Mechatronics with staff – technical assistance on electronic control system including programming and delivery of software

Petter Forbord, Sauer-Danfoss – technical assistance on hydraulic valves
Per Nyborg, Hydac Norge – technical assistance on pressure sensors

Staff at Norwegian University of Life Sciences:
Nils Bjugstad – guide/supervisor
Jan Kåre Bøe – guide in pre-study
Carlos Salas Bringas – help related to computer flow simulation

Ås, 15.05.2013

Emil Bjørås

ABSTRACT

This report comprises the development of function model for hydraulic stone release system on plough. A function test on the product has been carried out, followed by a design revision as a final part of the project. This development project builds upon pre-project "Kverneland Hydro Reset system" conducted as a part of the unit "Concept and product realization" at Norwegian University of Life Sciences.

The mission statement of this project is the following: "Develop a function model for hydraulic stone release system on plough. Carry out function test followed by design revision proposal".

The project has consisted of an introductory literature search and –study, based on preliminary project and gathered technical information. The product development process itself has been consisting of a conceptualization phase including the selection of a final product concept. The chosen product concept has been developed further towards a function model, where the physical model has been subject to a function test. In terms of measurements performed in the function test, a design revision has been accomplished with regards to the findings from the test. Throughout the development process contact has been established towards hydraulics suppliers, for advice in order to choose the most suitable components relative to the system specification. Software development for the electronic control system has been outsourced to Kverneland Mechatronics.

After carrying out the function test, the final conclusion from this project is that a re-test is required. The re-test should be implemented with proposed re-design stated in the design revision, in order to cover up for lack in hydraulic system capacity revealed during test. This lack in capacity regards the system ability to drain oil and keep constant system pressure during plough body release. The basic problem is that the pressure relief valve has a delay by 0,1 seconds until fully open, which induces a major pressure increase in the hydraulic system within the time it takes for it to fully open.

By design from function model, the system now withholds mechanical release characteristics with a calculated decrease by 25%, whereas the measured decrease from test was 21% at a lifting height by 350 mm. Maximum plough body lifting height is measured to 396 mm with the existing mechanical design. With the revised cylinder design, we now have a 0.36 l displacement of oil during a release sequence at maximum lifting height. In order to absorb the required excess energy for the first 0.1 seconds of the plough body lift, a gas accumulator has been implemented to cope with the opening time and delay of the pressure relief valve. The calculated minimum requirement of the accumulator imply an initial accumulator gas chamber volume by 1.9 liter in order to keep system pressure within 10% of pre-set pressure during those first 0.1 seconds of the sequence.

Through this project it is developed a function model for the product, whereas a function test has been carried out, accompanied by design revision proposal. Mission statement, including problems stated in introduction is hereby regarded as answered.

SAMMENDRAG

Denne rapporten innbefatter utvikling av funksjonsmodell for hydraulisk steinutløsersystem på plog. Prosjektet har videre bestått i funksjonstest av foreløpig utviklet produkt, etterfulgt av en design revisjon som siste del av prosjektet. Dette produktutviklingsprosjektet bygger på forprosjekt «Kverneland Hydro Reset System» gjennomført som del av emnet «Konsept og produktrealisering» ved Universitetet for miljø- og biovitenskap.

Prosjektets hovedmål er som følger: «Utvikle en funksjonsmodell for hydraulisk steinutløsersystem på plog. Utføre funksjonstest på prototyp, etterfulgt av forslag til design revisjon».

Prosjektet har bestått av en innledende litteraturstudie, basert på forprosjekt og annen innhentet teknisk informasjon. Selve produktutviklingsprosessen har bestått av en konseptualiseringsfase, inkludert seleksjon av endelig valgt konsept. Valgt konsept har blitt videreutviklet fram til en funksjonsmodell for videre funksjonstest av denne. Gjennom målinger foretatt ved funksjonstesten, har det blitt satt opp forslag til revidert design av prototypen. Gjennom dette prosjektet har det vært etablert kontakt mot hydraulikkleverandører i forhold til valg og bestilling av hydraulikkomponenter for det utarbeidede hydrauliske systemet. Utvikling av software til bruk for å styre systemet ble satt bort til Kverneland Group Mechatronics.

Etter å ha gjennomført funksjonstesten av systemet, er den endelige konklusjonen for dette prosjektet at en re-test av systemet er nødvendig. Dette skal gjøres i henhold til den foreslåtte design revisjonen, i forhold til å dekke opp for den manglende kapasiteten i systemet som ble avdekket under testingen. Denne manglende kapasitet relateres til systemets evne til å overholde kravet om tilnærmet konstant systemtrykk. Kilden til problemet er trykkbegrensningsventilen i det hydrauliske systemet og dens reaksjons-/åpningstid på 0,1 sekunder, som medfører en betydelig trykkøkning i systemet inntil ventilen får åpnet under første del av utløsersekvensen.

Med design fra funksjonsmodell innehar systemet en mekanisk utløserkarakteristikk teoretisk beregnet til et fall i karakteristikk på 25 %, hvorpå dette ble målt til 21 % under funksjonstest. Begge disse verdier er ut i fra en utløserhøyde på 350 mm. Maksimal utløserhøyde er målt til 396 mm med det mekaniske systemet utarbeidet. Med det reviderte sylinder designet impliserer dette et oljevolum på 0,36 l som må transporteres ut av og tilbake til sylinderen i løpet av en full utløsersekvens med maksimal løftehøyde. I forhold til å absorbere overskytende energi i systemet for de nevnte første 0,1 sekundene av utløsersekvensen, så har en 1,9 liter akkumulator blitt foreslått integrert i systemet. Denne er da beregnet til å kunne holde systemtrykket innenfor et avvik på 10 % .

Gjennom dette prosjektet har det blitt utviklet en funksjonsmodell for produktet med påfølgende test av denne og foreslått revidert design som følge av funn fra testingen. Med dette anses prosjektets mål, i tillegg til problemstilling fra introduksjonskapittel, å være besvart.

TABLE OF CONTENTS

1. INTRODUCTION

1.1 Background

1.1.1 The history of the agricultural plough

The history of agriculture is as ancient as the modern man. Hunting and day-to-day survival was the usual way of living before cultivation of land and livestock husbandry became an additional custom. By the introduction of what we know as agriculture, techniques of land cultivation emerged. One of the very main technical principles of preparing agricultural land is by the use of ploughs. Since the early Egyptians over 4000 years ago, the agricultural plough has been present, developing from simple hand-held hoes. Ploughing is often thought of as the ideal way of cultivating agricultural land. Compared to other types of soil cultivation, proper work by a plough makes a sealed seed bed, where all organic material is hidden beneath the soil, making ploughing an effective weed-killing mechanism, generating a beneficial growth environment for the intended grass stand on the field.

Since the industrial revolution, the plough has been developing in a more rapid scale, mainly due to the implementation of steam engines and finally the combustion engine on tractors for agricultural use. Following the extensive growth in engine power and complexity of tractors over the past decades, the size and efficiency of ploughs has increased at the same rate. This has raised the design requirement of ploughs when it comes to material strength, functional design and complexity. The integration of more automated systems for adjustable plough settings and power transmission systems on the plough by using hydraulics has come along, even in combination with electronics. The modern plough shall serve the purpose of creating an agronomic beneficial seed bed, while satisfying the demand for effectiveness that characterizes the modern industrialized type of agriculture.

Information in section 1.1.1 is obtained from (Jones, A.)

1.1.2 Kverneland Group

Kverneland Group is originated from the forging business of Ole Gabriel Kverneland, established in 1879 at Klepp in the region of Jæren in south west of Norway. He started off by producing scythes, until agricultural ploughs finally turned out to be the core-product of the business at Klepp. Situated in stony conditions in the south west of Norway, the local demand for robust equipment have enforced the development of ploughs with heat treated steel and robust design distinctive of the Kverneland brand and especially the Kverneland ploughs. Additionally these stony conditions have been part of increasing the need of a proper stone protection system, where Kverneland have been acknowledged for their mechanical stone release system – the Kverneland Auto Reset System.

Starting off from that smaller forging business, Kverneland has today grown into a bigger corporation, with widespread production plants in several European countries. Sales businesses world-wide has come along, but with Europe as the definite main market. What is today called Kverneland Group consists of around 2500 employees spread around several factories in Europe, but with the headquarters at the factory in Klepp, Norway. Kverneland Group was recently acquired by the Kubota Corporation, and is now part of the Japanese corporation consisting of 29 000 employees.

Information in section 1.1.2 is obtained from (Kubota Corporation 2013), (Kverneland Group 2013) and (Bjørås 2012).

1.1.3 Why a stone release system?

The mission of the stone release system is to protect both the plough and the tractor against the forces that occur when hitting stones and other obstacles while ploughing the soil. Later in this section, the most essential parts on the plough beam that needs this protection will be introduced. The stone release system is a shock absorber system, where the energy absorption can be achieved either mechanically, using mechanical springs or by hydro-pneumatics where the basic idea is to facilitate the use gas compression to absorb energy. A simpler type of stone protection system used by manufacturers is the shear bolt protection. In the shear bolt system, energy is not absorbed, but the connection between the plough body ploughing the soil, and the rest of the plough is broken using a shear-bolt. The shear bolt comes with a material weakness in form of a slot screw, making it break at a certain amount of shear stress. In that way the rest of the plough is protected against unwanted material stress.

In the following section possible breakages to the plough beam due to soil forces are illustrated. The plough beam is chosen since it is the part on the plough along with the plough body that is in direct interaction with the soil forces. Therefore it is a critical part with regards to breakages. Note: The following illustrations are not from actual breakages during use, but ones that are initiated by fatigue tests in test lab. However, they illustrate possible types of breakages on the plough beam due to material stress during ploughing. The stone release system will be able to protect the plough from the highest and most critical stresses, by absorbing some of the energy from forces added to the plough, consequently increasing the life time of the plough.

Common to all the illustrated breakages, is that it is caused by fatigue and initiated in micro cracks in relation to a weld.

Figure 1.1: *By applying a combination of vertical and horizontal forces onto the plough beam, a crack in the middle of the plough leg has been initiated. Bending stress is applied onto the beam, and as illustrated, the crack is on the front side of the leg. This is where the tensional stress is working. Repeatedly loaded and unloaded tensional stresses initiate crack growth at the front of the leg. Contrary, there is only compressional stress the along the backside of the leg - on the other side of the stress neutral axis. Obviously, compressional stress doesn't interfere with fatigue.*

Illustration reworked from (Kverneland Group 2005).

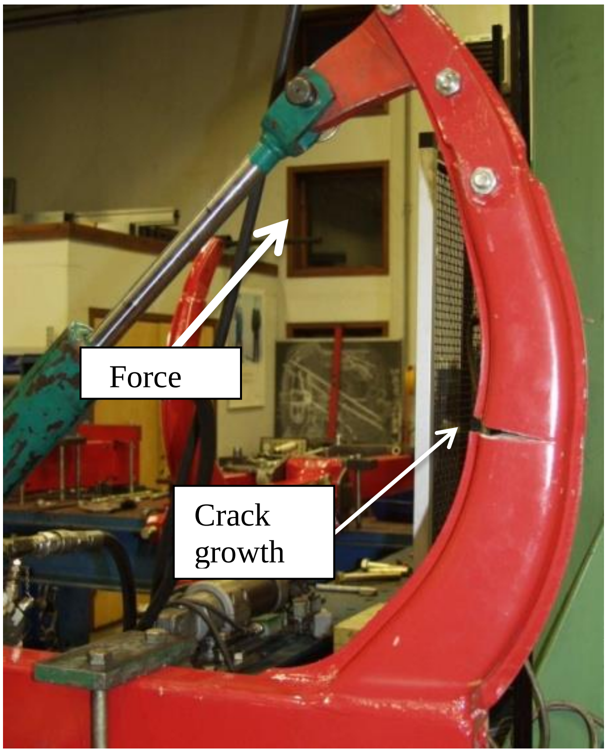


Figure 1.2: *In this illustrated case, the crack is caused by horizontal- and part sideways forces on the plough beam. This creates tensional stress on both the front and the side of the beam. The crack growth is appearing at the welded intersection between the leg and the main tube of the beam. This is not very surprising, since the area is subject to the highest bending moment, and crack growth is then made from the micro cracks around the weld line. Illustration obtained from (Kverneland Group 2002).*

1.2 Kverneland stone release systems in market today

Kverneland Auto Reset System

This is the most common stone release system that is found within the Kverneland range today. It is a mechanical system, comprising different types of leaf spring combinations due to what system resistance that is desired. It is a very reliable system and is practically maintenance free. The downside to the system compared to hydraulic systems is the lack of adjustability of resistance against soil forces. To adjust this mechanical system the customer needs to replace each leaf spring pack on the plough with a new one, if an adjustment of resistance is desired.

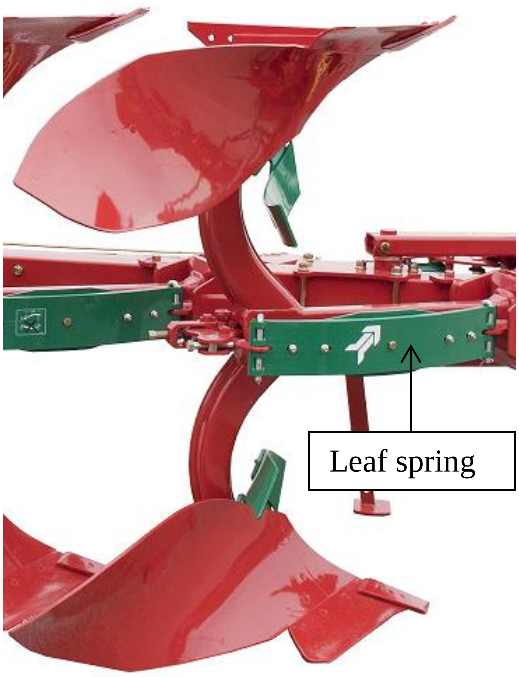


Figure 1.3 (left): *Kverneland Auto Reset System with the leaf spring coloured green.*
Figure 1.4 (right): *Kverneland shear bolt system.*

Illustrations obtained from (Kverneland Group).

Kverneland bolt protection system

The bolt protection system is simply a stone release system that uses a shear-bolt for each plough body. The shear-bolt is constructed with a notch which makes the bolt break at a certain shear force as described earlier. This system is very cost effective, but given the fact that shear bolts most likely need to be replaced for every bigger stone or obstacle, this system is only recommended in more or less stone-free land.

Information in section 1.2 obtained from (Kverneland Group) and (Bjørås 2012).

1.3 Basic working principle of the stone release system

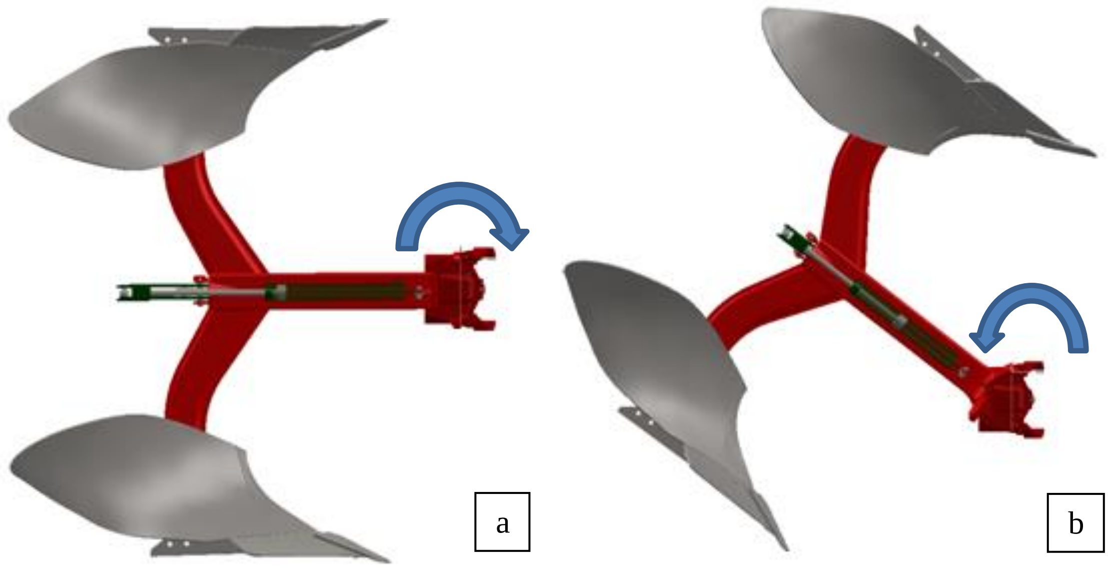


Figure 1.5 a: *In figure a, the plough body is in nominal ploughing position. When reaching a certain force on the point, e.g. when hitting an obstacle, the plough body starts releasing around its pivot point in the direction of the curved arrow. The part of the release sequence where the plough body is forced to lift by the forces added to the plough point is termed "phase 1" of the release sequence in this report. The "release sequence" is the term for plough point travel out of- and back into soil upon hitting obstacle.*

Figure 1.5 b: *When the obstacle is passed, the forces from the hydraulic system will once more outdo the forces from the soil and the plough body is ready to retract back into the soil in the direction of the second curved arrow.*

Illustrations reworked from (Kverneland Group 2012)

1.4 Review of concept proposal from pre-project

As already mentioned, this project will be based on the preliminary project and the concept proposal carried out in that project beforehand of this main project. In the following section the main findings and the product concept from the pre-project is presented.

1.4.1 Mechanical design

In the mechanical design from concept proposal in pre-project, the intention has been to exploit the design used on today's Kverneland Auto Reset beam. The Auto Reset spring on the outside of the beam has been replaced by a hydraulic cylinder, and the spring regulator is replaced by two tie plates bolted together with the tie rod and the cylinder.

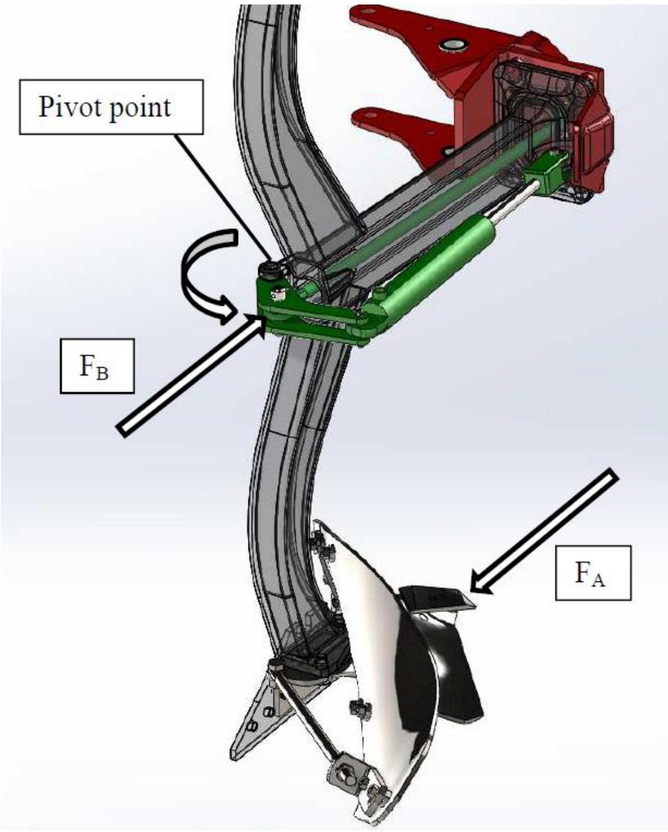


Figure 1.6 (left): *The system as it looks on the plough beam for the mechanical design from the concept proposal. Obtained from (Bjørås 2012).*

Figure 1.7 (right): *An overview of the mechanical set up of the system. Force F_A is acting on the plough point, with an opposing force F_B along the tie rod center axis. Obtained from (Bjørås 2012).*

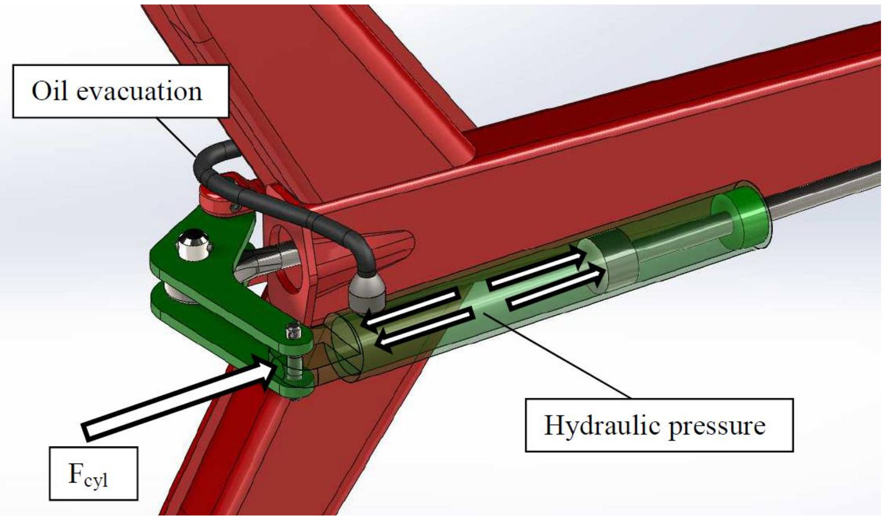


Figure 1.8: *This illustration shows how the hydraulic pressure in the cylinder is acting against the external forces distributed from the plough point and further onto the cylinder. F_{cyl} is representing that external force which the hydraulic pressure has to withstand. When hitting an obstacle, greater external forces are created, forcing the plough body to release, with an oil evacuation from the cylinder as the primary result. Obtained from (Bjørås 2012).*

1.4.2 Hydraulic system design

The hydraulic system is implemented with an adjustable pre-set pressure . It consists of a directional valve that adds and evacuates oil on demand to maintain constant system pressure. The pressure is measured by a pressure sensor, which in the next instance sends input to the system electronics, which process the signal. An output signal is then sent to the directional valve in order to either add or evacuate oil to or from the system. During a release sequence oil is being evacuated due to the pressure rise in the system. When the plough body is fully released, the force on the point will eventually drop, and the pressure drops. When the system pressure goes below the pre-set value oil will be added back to the system, once again enforcing a stable system pressure, and which forces the plough body back into ploughing position, after passing the obstacle.

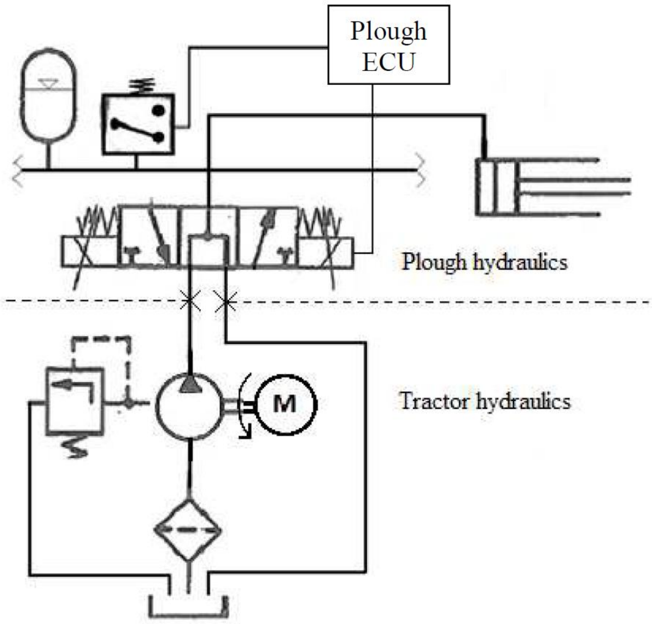


Figure 1.9: *This is the hydraulic schematics of the hydraulic system retained from the concept proposal in preliminary project. Obtained from (Bjørås 2012).*

In the pre-project there was however obtained some possible problems to the system functionality. This was mainly regarded to the directional valve and whether it is able to actually maintain system pressure. The fact that it is electronic actuated by solenoid creates a certain delay in terms of responding to system pressure increase in the system. Spool valves like the directional valve chosen from pre-project does also have some limitations with regards to oil flow capacity. This has to be looked further into in this study, possibly looking into alternative solutions when it comes to choice of valve system.

1.5 Hydraulic components

As a prelude to the later concept phase, some research on hydraulic directional valves and optionally pressure relief valves is done, in order to overcome the problems related to oil flow capacity and reaction time of the solenoid-operated solenoid valve.

Hydraulic directional valves

The hydraulic directional valve shall be 3-postion/3-way valve, optionally a 4-position/3-way with plumbed B-port. There are vast amounts of suppliers. Research will be done on standard valve solutions carrying out a search on what the most regarded suppliers have to offer, in order to find a valve that satisfies the system requirements. The valve or valve combination must be connected towards an electronic control of the system, which implies using a solenoid-operated valve with electronic actuation in some kind of way. The problem with a solenoid-operated valve with electromagnets in this case is that the flow capacity of such a valve rarely exceeds oil flow capacities of 130-150 l/min, which is the approximate flow ratio calculated for system in pre-project concept proposal, where problems related to flow capacity was issued.

Proportional valve in combination with servo mechanism

One alterative might be to use a proportional directional valve as a pilot valve together with a main directional valve. The pilot valve might be a smaller electronic actuated solenoid-operated directional valve, which in the next instance, when actuated, directs oil to one of the sides of the spool of the main valve, meaning that this valve is now hydraulic operated, where the valve system can be regarded as a servo mechanism.

Pressure relief valves

Due to the fact that directional valves have certain limitations with regards to oil flow capacity and reaction time, they might not suit the system in an acceptable manner. The pressure relief valves however are built to evacuate greater flows of oil in order to regulate system pressures effectively, which is just what the system should do. The downside to the use of pressure relief valves in combination with this system, is the that they only works one way, so the integration of a directional valve is inevitable in order to direct oil from tractor pump back into the system, unless implementing two pressure relief valves working each way.

Direct-operated/pilot-operated

Pressure relief valves can roughly be separated into direct-operated and pilot-operated. The direct-operated valve has the advantage of short reaction time, but has a bit more unstable pressure characteristics at high oil flows. The pilot operated valve takes just a bit longer to activate, but is more stable and delivers a practically constant pressure regardless of the oil flow, as long as it is operated within the valve setting.

Shock valves

These are special types of pressure relief valves which are designed with the mission to even out bigger system pressure shocks for a limited time interval. In other words it might not function as a relief valve in the way that it is able to drain oil through a complete release sequence. It will rather function as a system assist in order to maintain system pressure in the moment just after release until a main valve starts draining oil.

Cartridge valves

Cartridge valves are valves that can be fitted into valve housings in almost any wished arrangement and in combination with other circuit elements in the same valve block. It serves the purpose of compact design for hydraulic systems. The cartridge valves are one-directional, and normally have the function of a pressure valve. The biggest advantage by using a cartridge valve for the hydraulic stone release system in this project is that the oil flow capacity of the valve compared to a directional valve is normally bigger. Cartridge valves can be controlled by a pilot pressure towards its pilot pressure port, often referred to as “port x”, whereas flow goes from port A to B, or vice-versa.

Pressure sensor/-switch

Pressure switches are used to measure and maintain system pressure, by means of sending an output signal to the electronic system controlling the hydraulics. That output signal is used as an input signal by the electronics to control a hydraulic valve, in order to regulate the system pressure according to the system pressure setting.

Technical information on valves in section 1.5 is obtained from (Brautaset 1983), (Kjølle 1995) and (Forbord 2013).

1.6 Fluid dynamics

In the design analysis chapter, oil flow characteristics and the basic fluid dynamics for the pipeline system between cylinder and valve will be determined. Further down this section an introduction within the subject of fluid dynamics is carried out.

Oil flow and the velocity field

Within the field of hydraulics in motion, flow can be described by a velocity field. This velocity field describes the velocity of a given particle as a function of x, y, z coordinates and time – v (x,y,z,t). With regards to space coordinates this function is describing a three dimensional flow. In reality we are dealing with three-dimensional flows in most cases. But for straight pipelines, for instance, the flow can be regarded as either one- or two-dimensional throughout the pipe. If the velocity is constant throughout the whole pipe, it can even be regarded as one-dimensional, by using cylindrical coordinates (r,θ,x), where the particle velocity varies only for the radius, hence velocity as a function of radial distance from centerline of tube - v(r). The flow will normally be slower towards the surface wall of the pipeline, due to frictional forces between solid and liquid. If the velocity also varies along the pipeline (coordinate x), the flow is two-dimensional, hence velocity as a function of radial distance from pipeline centerline and distance x along same centerline – v(r,x).

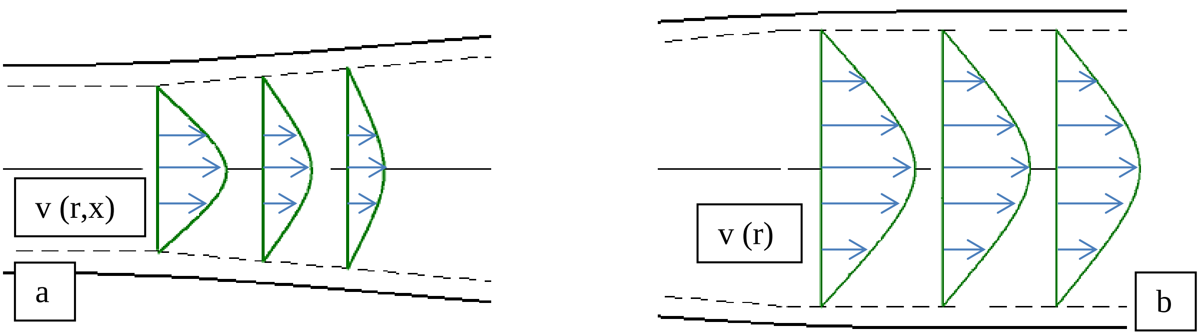


Figure 1.10 a, b: *As the illustrations show, at the junction in Figure a, the flow velocity decreases as the tube gets wider. This makes the flow dependent of the displacement along pipeline (x-direction). Additionally; the velocity in the outer layer towards pipeline wall surface is lower due to friction between solid and liquid. As mentioned, this makes the velocity dependent of the r-variable as well as the x-variable. Well passed the junction, in Figure b, the velocity is once again constant along x, and is only dependent on the radial distance from pipeline centerline. Reworked from (Fox et al. 2010).*

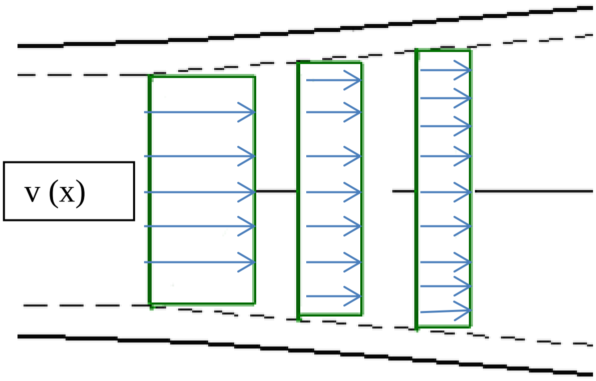


Figure 1.11: *As a further simplification of flow passing junction, it can be regarded as one-dimensional by looking at it as a uniform flow at the respective cross section along pipeline. In this case the flow is only dependent on the x-variable when passing the illustrated junction. Reworked from (Fox et al. 2010).*

Viscosity and shear stress

The viscosity of a fluid describes its ability to resist motion. For Newtonian fluids, the ratio between fluid shear stress and rate of deformation (shear rate) are proportional to each other. When taking the viscosity into account, we get the following equation describing this relationship for the shear stress:

$$\tau_{xy} = \mu * \frac{du}{dy}$$

; where μ is the viscosity, and du/dy is the shear rate.

Classification of fluid motion

To classify fluid motion, we can use what is termed continuum fluid mechanics, where we assume that we can look at a fluid as a continuous medium. Within continuum fluid mechanics flow can be classified as either viscous or inviscid. In cases where we deal with relative high viscosity, like oil, flows will be classified as viscous. Further down the classification ladder, flows can be separated into either laminar or turbulent flows where the calculation of the Reynolds number is essential to determine whether a flow is laminar or turbulent.

Basic hydrodynamic equations

The continuity equation

The continuity equation is based on continuum fluid mechanics, describing the transport of the fluid matter, as in this case hydraulic oil. The law for conservation of mass is used for the continuity equation. Within this equation several variables like pressure and oil flow and its velocity can be determined as a function of time.

The continuity equation which is derived from the law of conservation of mass yield:

$$\Delta m = \Delta t * \frac{\partial}{\partial t}\int_{\Delta V} \rho * dV$$

,where:

$$m = mass$$
$$t = time$$
$$\rho = density$$
$$V = volume$$

It describes the mass flow in the hydraulic system and its respective volume displacement, assuming incompressible flow, hence constant fluid density.

The Bernoulli equation

The Bernoulli equation describes the relationship between the kinetic- and potential energy of a fluid system. It based on the law of conservation of energy. This implies that an increase in oil flow, which in the next instance leads to an increase in kinetic energy, is followed by a decrease in static pressure and consequently potential energy. However; the Bernoulli equation can also be used to determine pressure loss along a pipeline due to frictional forces, where there is an actual energy loss or more precisely a transition from mechanical energy towards heat energy in the system.

This energy transition with regards to friction can be described by the following equation, for an incompressible fluid with constant flow:

$$p_1 + \rho g z_1 + \rho \frac{c_1^2}{2} = p_2 + \rho g z_2 + \rho \frac{c_2^2}{2} + \Delta p_{loss}$$

; where the pressure loss, Δp_{loss}, represents the loss in mechanical energy transistion towards heat energy in the system.

Information in section 1.6 is obtained from (Fox et al. 2010), (Brautaset 1983) and (Kjølle 1995).

1.7 Expected life time and applied forces onto plough beam

With regards to soil forces and the protection of the plough, some components are more exposed than others. For instance, the soil forces have more or less direct impact on the plough beam. In the following section the plough beam is reviewed with regards to fatigue. Some specific data on fatigue has been obtained for some critical parts on the plough beam. Having a clear picture of the acceptable forces and stresses applied onto the plough beam is important in order to design a suitable stone release system.

There are several parts on the beam that is critical with regards to crack initiation and final fractures. For that reason, three main test methods are used at the test lab of Kverneland Group to test the three main critical parts of the beam with regards to fatigue. The following section is moreover related to the information from previous section 1.1.3, which emphasizes the need of a stone release system on the plough. Furthermore, when referring to the number of load cycles as part of the fatigue requirement, that is understood as the number of load cycles until the component is broken in terms of a final fracture.

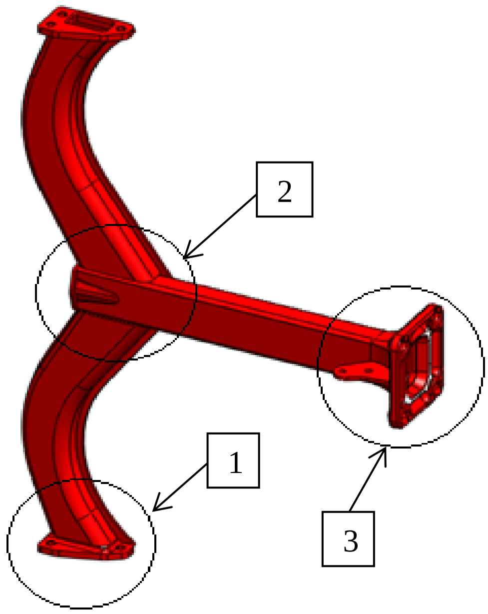


Figure 1.12: *The plough beam illustrated with its critical parts 1, 2 and 3 outlined. Reworked from (Kverneland Group 2012).*

Part 1: Connection between footplate and leg

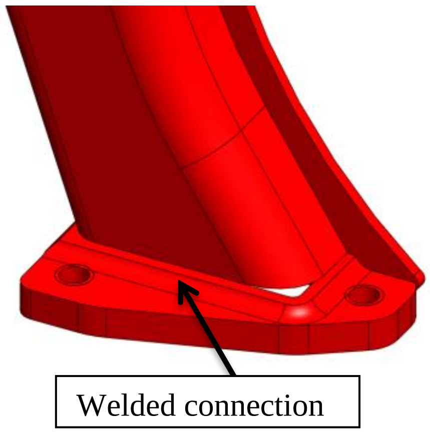


Figure 1.13: *The weld between the footplate and the leg is critical, and is tested against fatigue. Reworked from (Kverneland Group 2012).*

<u>Test</u>

The footplate – leg connection is tested as outlined below. The welded connection should withstand about 330 000 load cycles with regards to the minimum requirement in terms of fatigue for the applied force equivalent to 20 kN.

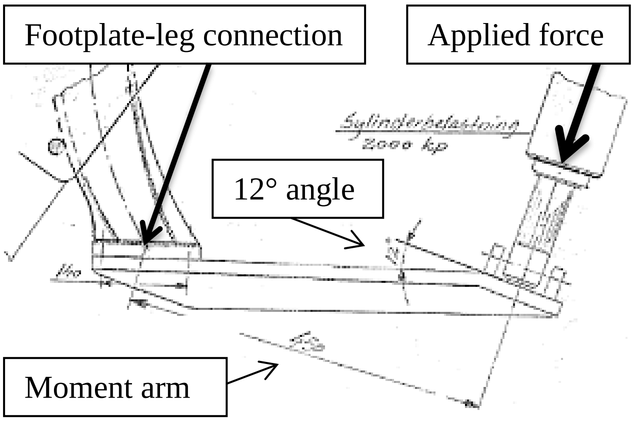


Figure 1.14: *This is the test setup for the test of the footplate – leg, welded connection. With an angle of 12° and a moment arm of 450 mm from the neutral axis of the weld, a force of 20 kN is applied. Reworked from (Kverneland Group 2006).*

Part 2: Connection between leg and horizontal tube

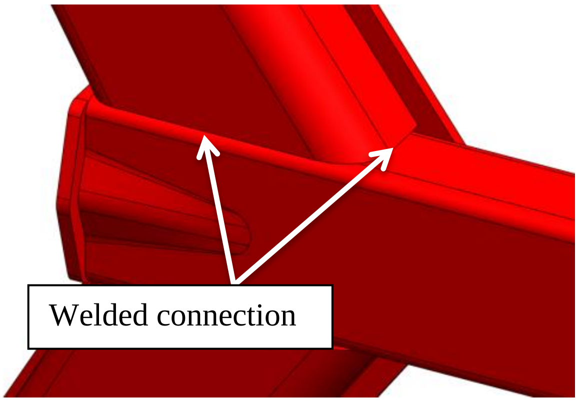


Figure 1.15: *The weld between the horizontal tube and the legs on each side are as well critical points with regards to fatigue. Reworked from (Kverneland Group 2012).*

<u>Test</u>

The test is performed by applying a force of 30 kN to the tip of the leg, with a 45° angle relative to the horizontal tube, as illustrated below. According to the test requirement it should withstand 600 000 load cycles.

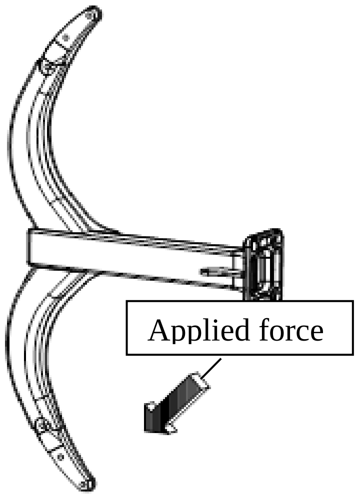


Figure 1.16: *The test setup with an applied force with 45° angle. Reworked from (Kverneland Group 2006).*

Part 3: Connection between horizontal tube and front plate

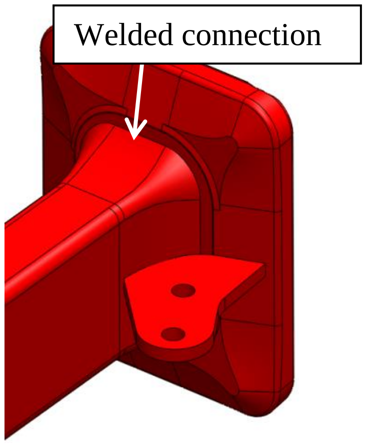


Figure 1.17: *Between the horizontal tube and the front plate of the beam there is a peripheral weld which also suffers from the applied stress distributed through the beam during plouging. Reworked from (Kverneland Group 2012).*

Test

The test against fatigue for this connection is carried out by applying a force of 15 kN in a 30° angle relative to the vertical plane, parallel to the side of the horizontal tube. The fatigue requirement is 250 000 load cycles.

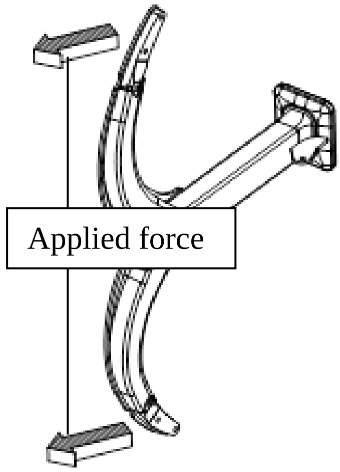


Figure 1.18: *The test setup. As seen from the illustration, the beam is applied with a 15 kN force at the end of each of the two legs, by using hydraulic cylinders. The load cycles are alternating between those two. While leg 1 is loaded, leg two is unloaded, and vice-versa. Reworked from (Kverneland Group 2006).*

Information in section 1.7 regarding the fatigue tests on the plough beam is obtained from (Kverneland Group 2006).

1.8 Main competitors

During the preliminary project, a minor research was carried out on the main competitors and the respective stone release systems they offer. Information on the pricing on a comparable hydraulic stone release system from one main competitor of Kverneland was also obtained. The main findings from the research were that all the main competitors offer a hydraulic stone release system. Many of the systems from competitors did also offer an adjustable hydraulic pressure setting. However, none of them came with any kind of pressure control in order to maintain stable hydraulic pressure in the system during the release sequence. Below is a review of comparable systems from the very main competitors: Lemken and Kuhn. However, there are other important competitors as well, like the Pöttinger Servo Nova system, but the market study will not be taken any further in this report other than some of the information already obtained from the pre-study.

Kuhn Varibar/Maxibar systems

The Kuhn Varibar system is hydraulic adjustable, with a pressure range from 90 to 150 bar, equivalent to a force on point ranging from about 6 to 13 kN. Kuhn also offers what they have called the Maxibar system, where a hydraulic pressure amplifier is integrated into the system. It is able to drive the force on point up to about 26 kN at the start of the release sequence, whereas the pressure drops down to the regular preset pressure just after release. This gives an advantage when driving in particularly heavy soils. Unlike most other systems, the release characteristics of the Kuhn stone release systems have an increasing resistance throughout the release sequence. This makes it a bit harder for the plough body to reach a full release and increases the stress on the plough beam to a certain extent. The biggest advantage with this type of release characteristics, however, is that the plough body retracts very effectively back into the soil while ploughing.

Lemken Hydromatic Auto-Reset Overload Device

The Lemken system features an adjustable pressure setting, ranging from 50-140 bar. The system has also a 3-dimensional suspension. This means that the body is able to release sideways in perpendicular direction of driving direction, in addition to releasing in upwards, vertical direction. For the Lemken system there has been obtained an approximate price level, where the price per plough body has been calculated to € 1130,-.

Information on competitors obtained from (Bjørås 2012).

1.9 Problem

From the preliminary project, some problems regarding system capacity were revealed. The main issues and bottlenecks of the present concept design from preliminary project can be comprised in the following:

- How to drain such high oil flows as present in this system at plough body lift in 1st phase of release sequence, in order to maintain constant system pressure?
- How to bypass the challenges regarding reaction time of the valve due to spool travel time and delay in electronic signal of solenoid-operated directional valve?

Bearing these problems in mind, outlining alternative solutions to improve the performance of the system will be a part of the concept generation phase of this main project. However, the concept proposal from pre-project still forms the basis for the further design.

2. PROJECT PLAN

2.1 Mission statement

Develop a function model for hydraulic stone release system on plough. Carry out function test followed by design revision proposal.

2.2 Project steps

- Introduction to the project
- Project planning
- Product Specification
- Concept generation, screening and –selection
- Design analysis and product architecture
- Function test and interpretation of results
- Design revision
- Product- and process evaluation
- Conclusion

2.3 Working schedule with milestones

Table 2.1: *Project working schedule with pinpointed deadlines.*

Deadline	Project phase
20.01	Project specification and planning
03.02	Basic research and project introduction
03.02	Product specification
01.03	Concept proposal
31.03	Product design proposal
05.04	Function test
28.04	Results interpretation, design revision
10.05	Process-, product evaluation and conclusion
15.05	Report deadline

The project plan itself can be found in appendix A4.

2.4 Project limitations

- Function testing will mainly comprise phase 1 of release sequence, with regards to system capacities in terms of hydraulic pressure and oil flow characteristics.
- No re-test will be performed as part of this project
- Multiple body tests or full-scale test out in field will not be performed as part of this project
- Implemented system pressure adjustment only by manual activation of directional valve through user interface.
- Mechanical design on new parts is limited to tie plate, whereupon only static structural analysis has been performed.
- System adaptions towards different types of tractor hydraulic systems is part of further investigation, and is not dealt with in this project
- This is a product development project, leaving further market analysis out of this specific project.

- Product cost is not taken into consideration at this point
- No research on Intellectual Property Rights (IPR) has been performed

3. METHODS

3.1 Project phases and solution tools

Research

The research will be carried out searching library databases as: BIBSYS, Science Direct and ISI Web of science. Additionally internet resources and internal documents at Kverneland Group will be used as references.

Project planning

The project plan will consist of a milestone plan in addition to a more detailed plan attached in appendix.

Product specification

The product specification shall come with an introduction to the different product properties and how they interfere with the system functionality. The product specification includes metric limit specifications for the product.

Conceptualization

The conceptualization will be separated into two parts:

- Detailed concept generation based on concept from preliminary project
- Concept screening and selection of concept comprising both hydraulic and mechanical part of system.

When putting up the hydraulic system schematics, the Microsoft Visio software will be used.

Product design analysis

Hydraulic system and components

The hydraulic system and schematics from the conceptualization will be presented and further discussed with a hydraulic supplier for implementation of specific valves to the system.

In relation with hydraulic system electronic control will be implemented. All development on electronics and software will outsourced to Kverneland Mechatronics.

Mechanical system

This part comprises static structural stress analysis on tie plate from chosen mechanical design.

- Manual hand calculations
- Finite Element Analysis using Ansys Workbench software

System characteristics

Calculation of system performance with regards to both mechanical release characteristics and calculation on hydraulic system characteristics will be performed. The calculation of mechanical release characteristics will be handled in Microsoft Excel, while hydraulic system characteristics will be handled by both hand calculations and further flow analysis using the

Solid Works Flow Simulation software, using calculation performed in Excel as the basis of these calculations.

Function testing

- Preparation of functional model and test
- Functional test in test lab
- Interpretation of test results
- Comparison between calculations and test results
- Product evaluation and design revision

During the testing the Senso Control will be used as the test measurements managing software. Components will have to be bought from hydraulics suppliers. The test results shall mainly be used to reveal possible insufficient capacity in hydraulic system. As far as practically feasible, the test results shall be compared to flow analysis and manual hand calculations performed. If the system fails to meet requirements stated in product specification, a product design revision will be performed.

The final part of the project comprises the product- and process discussion followed by the final conclusion.

3.2 Quality assurance

Product

In order to quality assure the product in terms of the functional model, the product specification and the metric limit specifications have been used as the governing guideline for the product functionality, during conceptualization, product design, testing and the design revision. By functional test and design revision, issues to the product functionality have been obtained and measures to improve product functionality have been made.

Furthermore the product quality assurance has been achieved by:

- Double checking calculations made in addition to comparing hand calculations with computer analysis (Finite Element Method and Computational Fluid Dynamics) along with comparisons to test results where applicable
- Consulting and exchanging ideas with the Kverneland Group Klepp R&D department

Project report

The quality assurance of the report has been achieved by:

- Register references consecutive while writing the report, using the EndNote citation software, in addition to double check on references up to delivery of report
- Once more double checking hand-calculations and assumptions made
- Checking language with regards to orthography and wording
- Checking report layout

3.3 Terminology

3.3.1 Technical terms and units

Table 3.1 a: *Basic physics terms and units of measure.*

Term	Symbol	SI-unit measurement	SI-unit measurement - abbreviated
Force	F	Newton (kilo-)	$N\ (kN)$
Length	l	Meter (milli-)	$m\ (mm)$
Area	A	Square-meter (milli-)	$m^2\ (mm^2)$
Volume	V	Cubic-meter (milli-)	$m^3\ (mm^3, L - non\ SI)$
Time	t	Seconds (milli-)	$s\ (ms)$
Temperature	T	Kelvin	$K(°\mathrm{C} - \text{non SI})$

Table 3.1 b: *Combined physics terms and units of measure.*

Term	SI-unit measurement	SI-unit measurement - abbreviated
Hydraulic pressure	Pascal	$Pa\ (MPa, bar\ (= 10^5\ Pa))$
Hydraulic flow	See table 3.1 a	$\frac{m^3}{s}\left(\frac{L}{min}\ (non\ SI)\right)$
Energy	Joule	$J\ (kJ)(= Nm)$

Table 3.2: *Technical terms.*

Term	Description
Release characteristics	(see chapter 4)
CAD	Computer Aided Design
FEM (-analysis)	Finite Element Method (-analysis)
CFD (-analysis)	Computational Fluid Dynamics (-analysis)

3.3.2 Hydraulic symbols

In the outlining of hydraulic system alternatives, there have been used hydraulic schematics to specify the systems.

Table 3.3: *Hydraulic symbols with descriptions. Hydraulic symbols are obtained from (Microsoft Corporation 2010).*

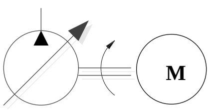
Symbol	Description
	Variable displacement pump, mechanically driven

Table 3.3 continued.

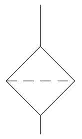
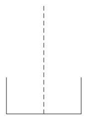
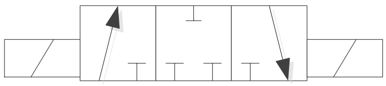
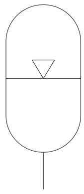
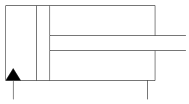
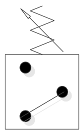
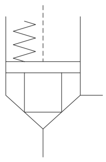
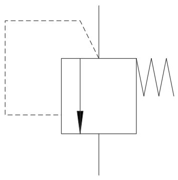
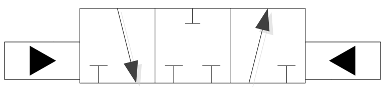
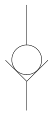
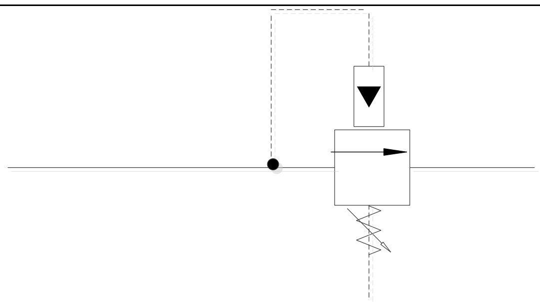
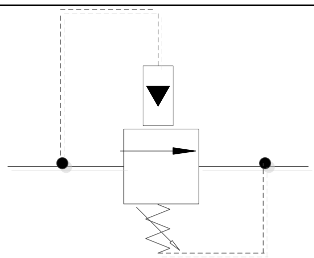
	Oil filter
	Oil reservoir/tank
	3/3 directional valve, solenoid-operated
	Hydro-pneumatic accumulator
	Hydraulic cylinder
	Pressure switch
	Cartridge valve with pilot pressure at spring chamber
	Pressure relief valve, pilot-operated, constant spring mechanical spring resistance
	3/3 directional valve, hydraulic-operated

Table 3.3 continued.

	Check valve, mechanically unloaded
	Hydraulic sequence valve, pilot-operated, hydraulic adjustable
	Hydraulic pressure relief valve, pilot-operated, mechanical adjustable

3.3.3 Equations

Table: 3.4: *List of equations used in the later calculations. Equations are obtained from (Brautaset 1983), (Fox et al. 2010), (Kjølle 1995), (Terjesen 2012), (Tipler et al 2008), (CFD-Online) and (cfdesign.com).*

Equation number	Equation	Description	SI-unit -abbreviated
1	$F_{cyl} = \frac{F_A}{\left(\frac{y4}{y1}\right) * \left(\frac{z6_{corr}}{z5_{corr}}\right)}$	Axial force on cylinder with regards to force-on-point and moment arm relations	N (kN)
2	$A_{cyl} = \frac{F_{cyl}}{p}$	The required cylinder piston cross-sectional area with regards to axial force on cylinder and the preset hydraulic pressure	$m^2\ (mm^2)$
3	$d_{cyl} = \sqrt{\frac{4 * A_{cyl}}{\pi}}$	Required cylinder piston cross-sectional diameter	$m\ (mm)$
4	$m = \rho\ (rho) * V$	Mass of the displaced hydraulic oil during a release sequence	kg

Table: 3.4 continued.

5a	$V = \frac{1}{2}\int_{t_0}^{t_n} Q(t)\,dt$	Displaced volume during release sequence, by integration	m^3 $(dm^3 = L, mm^3)$
5b	$V = Q_{avg} * t$	Displaced volume during release sequence, by average flow	m^3 $(dm^3 = L, mm^3)$
6	$E_p = p * V$	Total energy in system during a whole release sequence in terms of system pressure and volume displacement	$J\ (kJ)$
7a	$s_{cyl} = \frac{1}{2}\int_{t_0}^{t_n} v(t)\,dt$	Cylinder displacement by integration	$m\ (mm)$
7b	$s_{cyl} = v_{avg} * t$	Cylinder displacement by average velocity	$m\ (mm)$
8	$Re = \frac{Q * d}{\vartheta * A}$	Reynolds number	-
9	$v = \frac{Q}{A}$	Oil flow velocity through system	m/s
10	$v = v_{max}\left(\frac{y}{r}\right)^{\frac{1}{7}}$	Velocity profile for turbulent flow across cross-sectional area of pipeline	m/s
11	$P_{tot} = p + \lambda * \frac{l}{d} * \rho * \frac{v_m^2}{2}$	Bernoulli equation with regards to pressure	$Pa\ (MPa, bar\ (= 10^5\ Pa))$
12	$p_{tot} = p + \rho(rho) * \frac{v^2}{2}$	Bernoulli equation with regards to mechanical energy – no friction effects	$Pa\ (MPa, bar\ (= 10^5\ Pa))$
13	$\lambda = \frac{0{,}316}{R_e^{0{,}25}}$	Fluid internal friction coefficient	-
14	$p_{loss} = \delta\rho\frac{v_m^2}{2}$	Pressure loss in pipeline bends	$Pa\ (MPa, bar\ (= 10^5\ Pa))$
15	$h_l = \frac{\Delta p}{\rho}$	Loss in fluid momentum with regards to pressure drop and fluid density (head loss)	$\frac{m^2}{s^2}$

Table 3.4 continued.

16	$E_{loss} = h_l * m$	Energy loss/transition	J
17	$C_{oil} = \rho * c_{oil}$	Heat capacity of oil	$\frac{J}{K}$
18	$\Delta T = \frac{E_{loss}}{C_{oil}}$	Change in temperature	K
19	$I_t = 0{,}16 * Re^{-(1/8)}$	Turbulence intensity	% (non SI)
20	$k = \frac{3}{2} * (v_{oil} * I_t)^2$	Turbulence "energy"	$\frac{J}{kg}$
21	$\epsilon = \frac{c_\mu^{(3/4)} * k^{(3/2)}}{0{,}07 * L}$	Turbulence dissipation	$\frac{W}{kg}$
22	$tk = 0{,}7 \sqrt{\frac{F_{Ed} * \gamma_{M0}}{f_y}}$	Required material thickness with regards to external forces and material quality	$m\ (mm)$
23	$N_{t,Rd} = \frac{0{,}9 * A_{net} * f_u}{\gamma_{M2}}$	Material resistive force	N
24	$M_{el,Rd} = \frac{W_{el} * f_y}{\gamma_{M0}}$	Material moment resistance	$Nm\ (Nmm)$
25	$\sigma_{b,A-A} = \frac{M_{b,A-A}}{W_{b,A-A}}$	Bending stress	MPa
26	$M_{b,A-A} = F * l$	Bending moment	$Nm\ (Nmm)$

Table 3.4 continued.

27	$W_{b,A-A} = W_{el,A-A} = \frac{b * h^2}{6}$	Section modulus – rectangular cross section	mm^3
28	$\sigma_{b\perp} = \tau_{b\perp} = \frac{\sigma_b}{\sqrt{2}}$	Perpendicular tensile- and shear stress	MPa
29	$\tau_{parallel} = \frac{F_{cyl}}{A_{A-A}}$	Parallel shear stress	MPa
30	$\sigma_{uniform} = \sqrt{\sigma_{b\perp}^2 + 3 * \left(\tau_{b\perp}^2 + \tau_{parallel}^2\right)}$	Uniform stress	MPa
31	$V_{cyl,displacement} = \left(\frac{\pi * {d_{cyl}}^2}{4}\right) * s_{cyl}$	Cylinder volume displacement	mm^3
32	$p_2 * V_2^\gamma = p_1 * V_1^\gamma$	Thermo-dynamic relation between initial and final system pressure and –volume (Quasi-static, adiabatic gas compression)	-
33	$\gamma = \frac{C_p}{C_v} = \frac{C_v + nR}{C_v}$	Thermo-dynamic relation between heat-capacities at constant pressure –and volume respectively	-

Table 3.5: *Description of equation terms. Terms are obtained from (Brautaset 1983), (Fox et al. 2010), (Kjølle 1995), (Terjesen 2012), (Tipler et al 2008), (CFD-Online) and (cfdesign.com).*

Term	Description	SI-unit – abbreviation
F_A	Force-on-point	$N\ (kN)$
p	Hydraulic system pressure	$Pa\ (MPa)$ $bar\ (= 10^5\ Pa - non\ SI)$
ρ (rho)	Fluid density	$\frac{kg}{m^3}$

Table 3.5 continued.

Q	Fluid flow	$\frac{m^3}{s}\left(\frac{L}{min}\ (non\ SI)\right)$
V	Hydraulic oil volume	$m^3\ (dm^3 = L,\ mm^3)$
d	Cross-sectional diameter	$m\ (mm)$
ϑ	Kinetic viscosity	$\frac{m^2}{s}\left(\frac{mm^2}{s}\right)$
A	Cross-sectional area	$m^2\ (mm^2)$
y	Radial distance from inner surface wall of pipeline towards its centerline	$m\ (mm)$
l	Length	$m\ (mm)$
δ	Friction coefficient, pipe bend	-
c_{oil}	Specific heat capacity of oil	$\frac{J}{kg * K}$
c_{μ}	Dissipation coefficient	-
F_{Ed}	External force	N
γ_{M0}	Material factor	(1.05)
f_y	Material yield stress limit	MPa
A_{net}	Net cross-sectional area	$m^2\ (mm^2)$
f_u	Material ultimate strength	MPa
γ_{M2}	Material factor	(1.25)
C_p	Heat capacity at constant pressure	$\frac{J}{kg * K}$
C_v	Heat capacity at constant volume	$\frac{J}{kg * K}$

4. PRODUCT SPECIFICATION

The product specification consists of the most essential product properties, their importance and the metric limit specification that the product shall satisfy. When putting up and evaluating the later concept proposals, they shall be evaluated towards all the listed product properties.

However, when evaluating the final product, it shall be referred to the metric limit specifications only. This is due to the fact that some of the product properties listed does not apply for the metric limit specification at this stage. Either they are not relevant to this project at this stage or they are not possible to quantify at this point. However they should be kept in mind, but will be dealt with in more detail at a later point.

4.1 Product properties

To be able to define the product and put up a product specification, a set of important product properties are listed. The different product properties are described before going through the product specification itself.

Release characteristics

One of the most important system properties are the release characteristics, as stated in Table 4.3. Since the hydraulic system should operate at constant pressure during the release sequence, the release characteristics will be more or less defined by the mechanical set up. To evaluate whether rearrangements in the mechanical design are needed, the relationship between the moment arms in the mechanical set up during a release sequence is calculated up to a lifting height of 350 mm. The complete moment arm relation defining the release characteristics is made out of two individual moment arm relations, giving the final release characteristics as described in more detail in appendix A1.

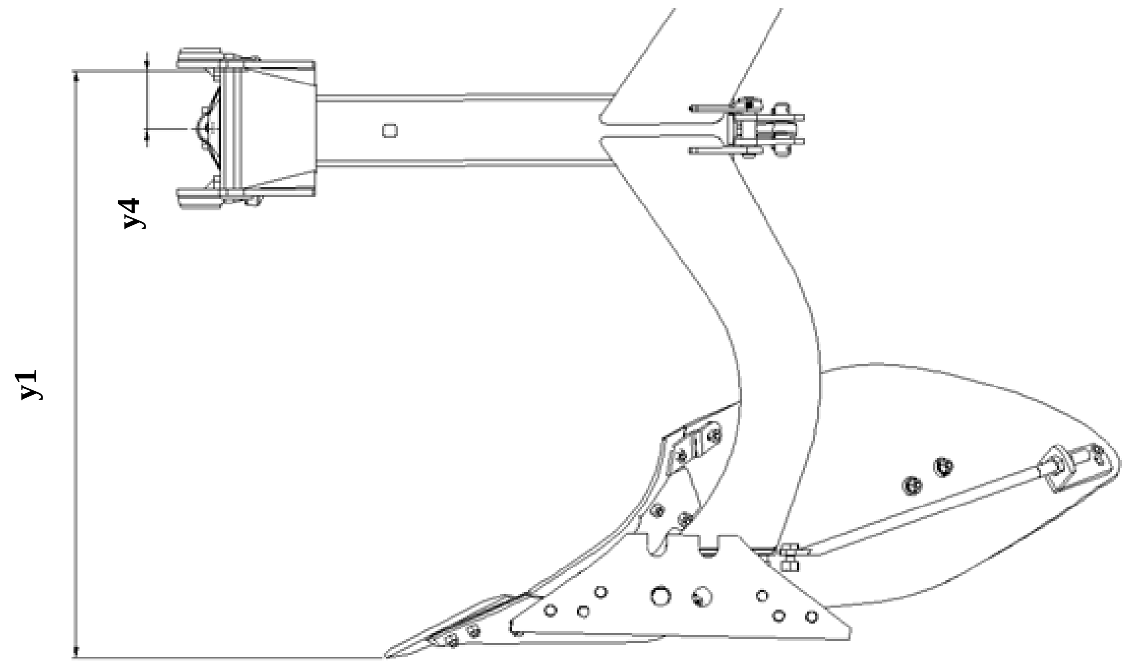


Figure 4.1: *Illustration of the "moment arm relation 1", which is the relation of y4 divided by y1. Reworked from (Kverneland Group 2012).*

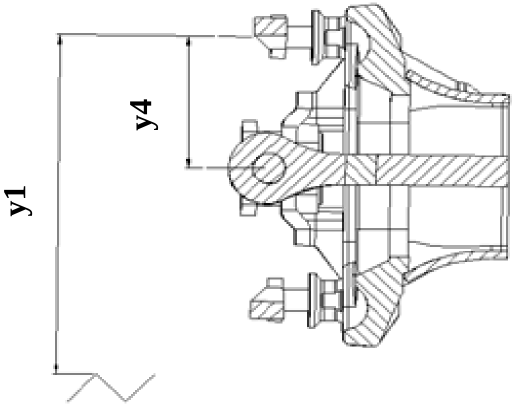


Figure 4.2: *Both moment arm y1 and y4 starts at the top of the beam holder on its double ball joint, where y1 terminates down at the plough point, while moment arm y4 terminates at the center axis of the tie rod. Reworked from (Kverneland Group 2012).*

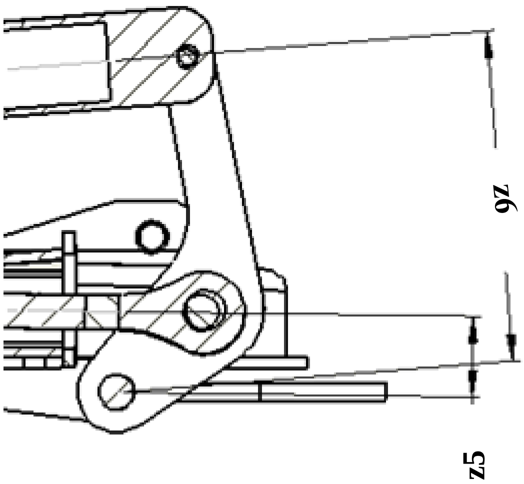


Figure 4.3: *Illustration of "moment arm relation 2" which is the relation of z6 divided by z5. Reworked from (Bjørås 2012).*

<u>Release characteristics from preliminary project concept proposal</u>

The calculation of the release characteristics as a function of time can be found in appendix A1, along with descriptions, and more detailed illustrations of the different moment arm relations. The release characteristics from the chosen design in the preliminary project have been calculated as part of this main project. The characteristics are calculated as a function of release height, while release characteristics for the further concepts in this study are calculated as a function of time. However, the calculation basis is the same.

As seen from the Figure 4.4, the release characteristics at release height of 350 mm, imposes a system resistance to the plough point that is more than half on full release compared to when body is in nominal ploughing position. When taking into consideration that the plough body should retract relatively fast back to the soil to continue the ploughing after an obstacle, the release characteristics outlined might be decreasing too much during the release sequence with the proposed design from the concept proposal. Therefore, a more thoroughly conceptualization phase on the mechanical design is required, by looking at different alternatives giving the acceptable characteristics.

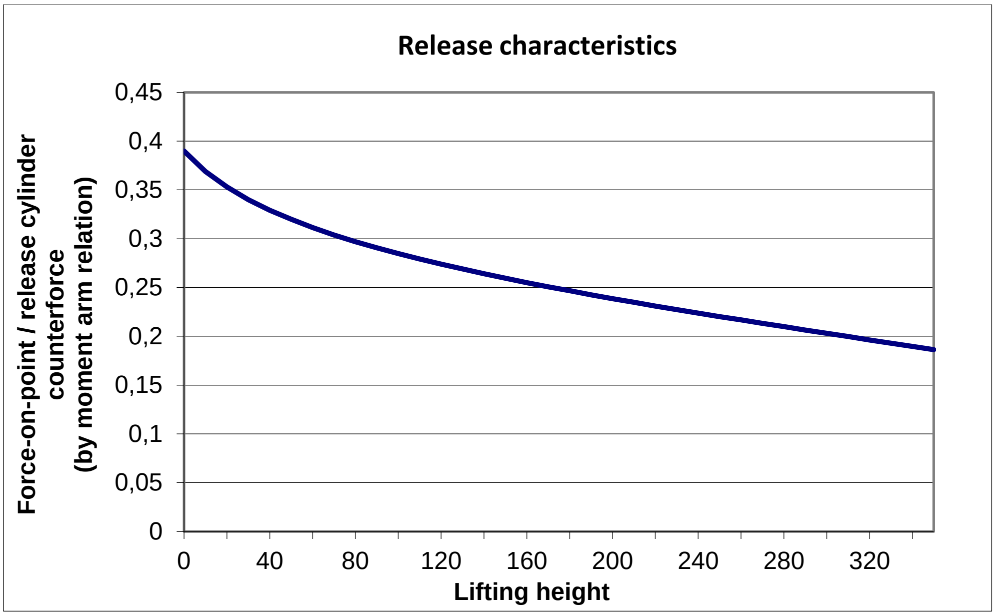


Figure 4.4: *The release characteristics re-calculated at this point for the mechanical set-up from pre-project.*

Release resistance along with initial forces and moments

With the plough body in nominal position in the soil, the stone release system should be able to withstand a certain force from the soil. This resistive force should be adjustable by the hydraulic pressure in the system. The rest is defined by the mechanical setup with the moment arm relations present between the plough point and the hydraulic cylinder, predefining the release characteristics, as previously described.

Mechanical setup from preliminary concept proposal

To withstand the force of maximum 16 kN at the plough point, which was defined in the preliminary project, a mechanical system has been designed as briefly described in the introductory section. The so far chosen design is described a bit further in the following section. However alterations on geometry must be performed to achieve acceptable characteristics.

Figure 4.5: *This illustration shows the mechanical setup and second moment arm relation in the concept proposal from preliminary project. Reworked from (Bjørås 2012).*

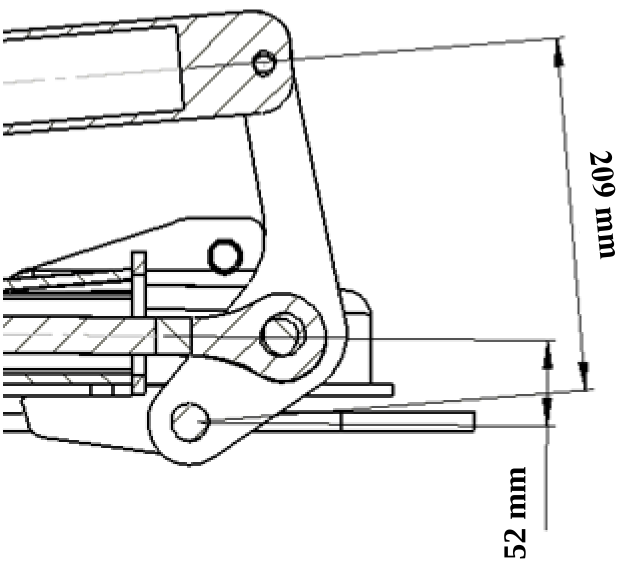


Table 4.1: *Calculated axial force required for cylinder to withstand a plough point force of 16 kN, with concept design from pre-project. Reworked from (Bjørås 2012)*

FB (Axial force on tie rod)	163.67	kN
z5	52	mm
z6	209	mm
FC	**40.72**	**kN**

Cylinder

Table 4.2: *Calculation of cylinder cross-sectional area and diameter. Reworked from (Bjørås 2012)*

Hydraulic pressure	16	MPa
Cylinder cross-sectional area	2545	mm^2
Cylinder piston diameter	57	mm

With a maximum system pressure of 160 bar (16 MPa) retained from the product specification in the pre-study, the needed cylinder diameter of 57 mm is found for the design so far chosen.

System pressure stability

An important product property in order to avoid that the hydraulic system interferes with the release characteristics, is the system ability to maintain a stable pressure during the entire release sequence. This naturally implies the right choice of hydraulic system and components with the required capacities. Special considerations with regards to components were outlined in the "Introduction" chapter and the relevant hydraulic systems are further investigated in the next concept phase.

Release height

In order to pass obstacles effectively and keep a general good plough quality during release it is important that the plough body can be lifted so high that it is able to pass over the obstacle without creating any upwards lift to the plough, which would create a momentary lack in plough quality in forms of too shallow ploughing.

Price

Obviously, price is an important factor when defining the system. This system is intended as an additional "high-end" option to the existing mechanical stone release system of Kverneland – the Auto Reset System, which allows for a bit more liberated pricing level. However, costs of components need to be kept down as much as possible. Meanwhile, the more critical product properties at this stage, like the release characteristics and pressure stability in the hydraulic system must be fully satisfied before taking the question regarding price further into consideration.

User-friendliness

The user-to-system interaction is also an important property of the final product, and is the type of product property that might give the product the final edge and completeness, provided that the right choices are made when dealing with the user interface, for instance.

For the time being the system should be manually operated by pre-setting the system pressure. However, a further system implementation in the longer run would be to use sensor that indirectly measures soil resistance. In that way the system will be able to facilitate automatic adjustment of its own pressure setting while ploughing.

System integration with plough design (both mechanical and hydraulic part of system)

Ideally, there should be as little alterations as possible on existing parts when fitting the new system onto the plough. This is, of course, due to the cost that new design revisions on existing parts will induce.

Mass and interference with center of mass of plough beam

The mass of the system should be as low as possible to avoid adding additional weight to the plough. Depending on the placement of the new components, like the hydraulic cylinder, the center of mass of the plough beam with components might deviate from the center of mass found on the Auto-Reset System that exists today. However the deviation in center of mass on plough beam should be as small as possible, but is not further specified in terms of metric limit specification at this point.

Aesthetic design

Aesthetic design is important in the way the product applies to the customer. A major goal within the aesthetic design of this product is to implement the aesthetics with the functional design and aim for an as organic design as possible. However, the primary goal of this product is its functionality. That means that the aesthetic design should be a minor factor that shall only be bared in mind, and seen as a final bonus in terms of achieving an appealing design on the final product.

Table 4.3: *Listing of the most important product properties. With regards to their influence on the final assessment on the overall product, each of the properties have been rated relative to their importance on a scale from 5 = Most important to 1 = Less important.*

Product property	Rating of importance (1-5)
▪ Release characteristics	5
▪ Release resistance (force on point allowance)	5
▪ System pressure stability	5
▪ Release height	4
▪ Price	4
▪ Ease of system adjustment and user-friendliness	3
▪ Integration of system with the plough design	3
▪ Mass	2
▪ Aesthetic design	1

4.2 Metric limit specifications

With regards to the relevant product properties at this point, metric limit specifications are outlined below, where the functional model shall be evaluated against those metric limit specifications.

Table 4.4: *Metric limit specifications*

Product property	Metric limit specification
▪ Force-on-point resistance	10 - 16 kN
▪ Reduction in release resistance during release sequence (release characteristics)	Maximum 30% reduction in system release resistance, with regards to force-on-point
▪ Hydraulic system pressure	100 – 160 bar, adjustable
▪ Pressure deviation allowed within set system pressure	Maximum 10% pressure deviation from preset system pressure during a release sequence
▪ Release height	350 – 400 mm

5. PRODUCT CONCEPT

From the minor discussion in the introduction chapter regarding the design chosen in the pre-project, the release characteristics were not found ideal. The so far chosen hydraulic system has some issues regarding valve capacity. In this product conceptualization further design options will be worked out for both mechanical- and hydraulic system.

5.1 Hydraulic system

This concept generation shall consist of different options and varieties within the already chosen main functional principle. The so far hydraulic schematics, which can also be found in the introduction chapter of this report, are once again repeated in the following functional conceptualization phase. For each alternative both schematics and functional description of the hydraulic system is attached. For hydraulic symbol descriptions, see “*Methods”* chapter. As a matter of form; hydraulic symbol have once more been obtained from Microsoft Visio, whereupon schematics also have been created.

Alternative 1: As is

This is the same hydraulic schematics as in the existing proposal, and is alternative 1 in this concept generation and the basis for the other alternatives outlined. This alternative however, should use an optimized directional valve with regards to oil flow capacity, in order to make the system behave as desired.

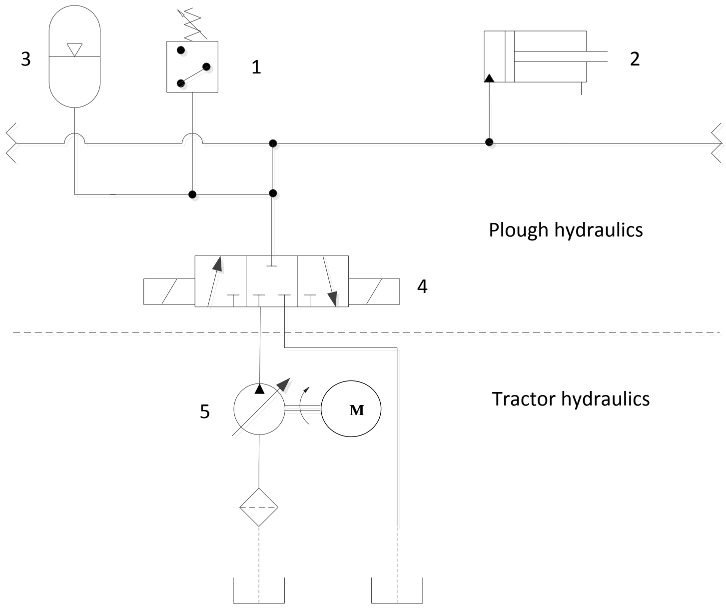


Figure 5.1: *Hydraulic schematics for alternative 1.*

Functional description; adjusting system pressure:

1. Required system pressure is fed into the electronic control and compared against actual system pressure logged by pressure switch (1).
2. Depending on whether it is a pressure increase or decrease that is required, the directional valve (4) either adds or evacuates oil to the hydraulic system, which simultaneously increases or decreases the gas pressure on the accumulator (3) along with the hydraulic pressure.

Functional description; release sequence:

1. An external force which exceeds the force from the hydraulic system pressure is acted upon the piston rod of the cylinder (2). This results in an increase in system pressure, which is the measured by pressure switch (1). The external force should only exceed the force from the system pressure when hitting an obstacle, like a stone, if the system pressure setting is correct.
2. In the next instance, input from the pressure switch is fed into the electronics, sending an output signal to the directional valve (4) with instructions to open. The accumulator (3) serve the purpose as a minor energy absorbing unit until the valve is fully open, in addition to absorb smaller mechanical vibrations from the soil during ploughing.
3. When the obstacle is passed, the external force is relieved, and the hydraulic pressure will start forcing the plough body back into the soil. The system pressure will drop, causing the directional valve to open and add oil to the system with use of the tractor pump (5) pressure.

Alternative 2: Servo system

The second alternative is based on alternative 1, but comes as a 2-stage system, with both a solenoid-operated valve and a hydraulic operated valve. This system can be defined as a type of servo- or master-slave system. The pressure adjustment is achieved using the solenoid operated valve, adding oil to the pilot circuit operating the hydraulic-operated valve. A small accumulator is attached, both as an energy absorber –and source in the pilot system. The other side of the hydraulic-operated spool is connected to the hydraulic release system itself, consisting of the cylinder and the main accumulator. Due to the pressure setting, the solenoid-operated valve is either adding or evacuating oil to the pilot circuit working with the set system pressure towards the one side of the spool. If the pressure setting is increased, the pressure on the right side of the spool is higher than the pressure on the left hand side connected to the main system. This is making the spool travel towards its left position, adding oil to the primary part of the circuit from the tractor pump until pressure on both sides of the spool are aligned. In that manner the final system pressure is adjusted. The exact same principle of adjustment is valid for relieving the system pressure, but the other way around, by evacuating through the two directional valves. When the plough body hits an obstacle, pressure rises in the main circuit. The left side of the spool is acted on by that increased pressure, which makes the spool go rightwards. Then oil can be evacuated through the hydraulic-operated valve to tank. On the right side of the spool the secondary accumulator will absorb some energy, while the solenoid-operated valve is directed with regards to maintaining the preset pressure. On the way to plough body reset, the pressure on the left side of the spool is below the pressure on the right side. Oil is then redeployed to the system through the hydraulic-operated valve, and the plough body is re-set into the soil. The functional principle is worked out in the hydraulic schematics in Figure 5.2.

Functional description; adjusting system pressure:

1. Required system pressure is fed into the electronic control and compared against actual system pressure logged by pressure switch (1) in the pilot pressure circuit.
2. Depending on whether it is a pressure increase or decrease that is required, the solenoid-operated directional valve (5) either add or evacuate oil to the hydraulic system, which increase/decrease the pressure on the right side of the spool of directional valve (4).
3. The directional valve (4) is then either adding or evacuating oil from the primary circuit until the pressure on both sides of the spool of valve (4) is equalized.

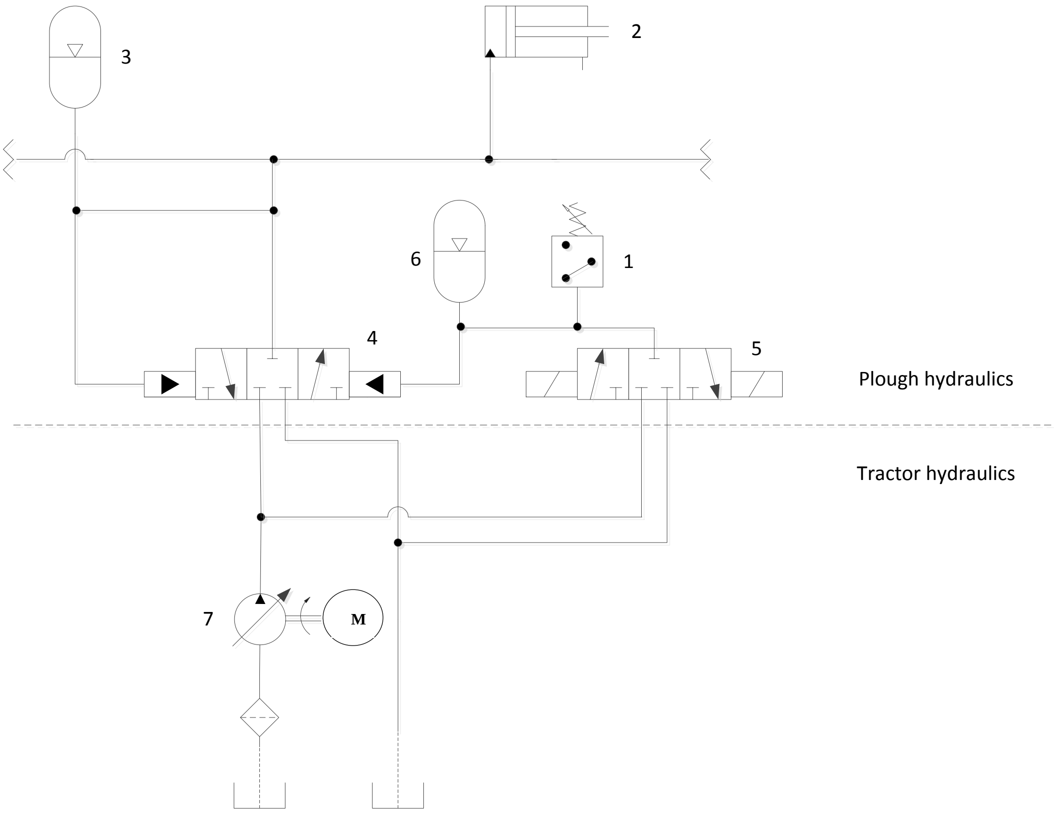


Figure 5.2: *Hydraulic schematics of the servo system.*

Functional description; release sequence:

1. Due to a stone or another obstacle an extra external force is acted upon the plough. This external force which exceeds the force from the hydraulic system pressure is then acted upon the piston rod of the cylinder (2). This result in an increase in system pressure exceeding the pilot pressure, which makes the spool of valve (4) travel rightwards. This make valve (4) evacuate oil to tank during the release sequence.
2. When the obstacle is passed by, the external force is decreased to a minimum, and the system adjusted pressure on the right side of the spool will make sure the valve (4) adds oil back to the system by moving the spool leftwards, while the plough body is being deployed back into the soil.

Alternative 3: Regenerative hydraulic loop

With this alternative, some of the flow from the hydraulic cylinder (2) during release is directed towards the other side of the piston. This lowers the demand for oil flow capacity at the directional valve (5), while at the same time decreasing the transport distance of the oil. At the rod side of the cylinder piston there is a minor system pressure set by the pressure relief valve (6). During release, oil flow is directed through directional valves (4, 5) to tank and to cylinder piston side respectively. Some of the flow on the cylinder piston side of the circuit is also directed to tank through relief valve (6). Hydraulic schematics can be found in Figure 5.3.

Functional description; adjusting system pressure:

1. Required system pressure is fed into the electronic control and compared against actual system pressure logged by pressure switch (1).
2. Depending on whether it is a pressure increase or decrease that is required, the solenoid-operated directional valve (5) either adds or evacuates oil to the hydraulic system. Valve (4) is kept closed.

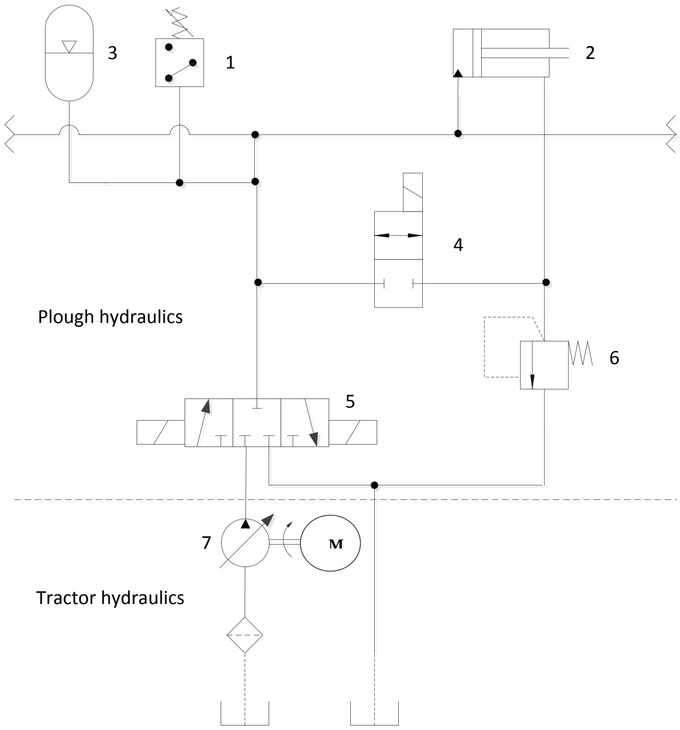


Figure 5.3: *Hydraulic schematics regenerative loop.*

Functional description; release sequence:

1. An external force which exceeds the force from the hydraulic system pressure is acted upon the piston rod of the cylinder (2). This result in an increase in system pressure which is sensed by pressure switch, where the directional valves (4, 5) are directed to evacuate oil.
2. Secondly; oil which is evacuated through valve (4) is led towards the piston side of the cylinder and additionally through pressure relief valve (6) to tank. Note: Pressure relief valve (6) will have a very low pressure setting.
3. When the obstacle is passed by, the external force is decreased to a minimum, and the directional valve (5) is instructed to open in order to add oil back to the system by use of the signal from the pressure switch. To exploit the possible short distance supply of oil from the cylinder rod side, valve (4) should by principle be open at this stage. But due to the low pressure in this part of the circuit which complicates redistribution of oil towards the high pressure side of the cylinder piston, valve (4) must be closed.

Alternative 4: Regenerative loop with cartridge valve

This system basically has the same functionality as alternative 3. But instead of using a 2-way solenoid-operated directional valve, the use of a cartridge valve (4) is chosen. The advantage of the cartridge valves is that they are generally ideal in systems dealing with high oil flows. There is implicated a pilot pressure on the cartridge valve which is made common with the system pressure, except during release, where those to are seperated by a check valve (5). Between the pilot side and the A-port side of the cartridge valve the area relation should be

1:1, and only with a very small spring force biasing the pilot side, in order to make the cartridge valve balanced.

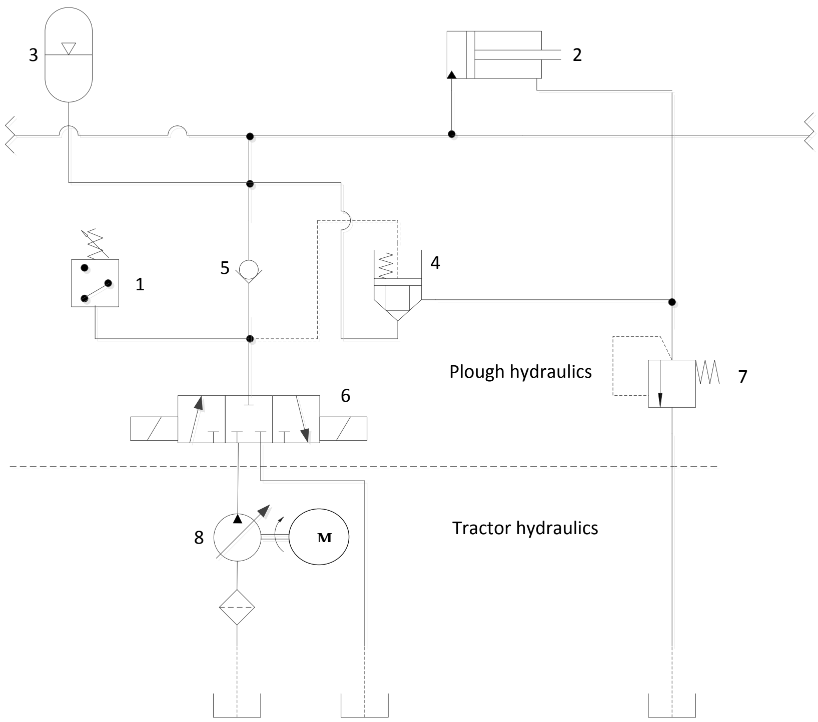


Figure 5.4: *Hydraulic schematics for re-generative loop with cartridge valve.*

Functional description; adjusting system pressure:

1. Required system pressure is fed into the electronic control and compared against the pilot pressure measured by pressure switch (1).

2a. If a pressure increase is required, the solenoid-operated directional valve (6) adds oil to the hydraulic system, both to the pilot circuit and to the main system circuit through check valve (5). In this way the pilot pressure and system pressure corresponds to each other, making the cartridge valve (4) closed when the system is balanced, due to the minor additional spring force biasing the pilot side of the cartridge valve.

2b. If a pressure decrease is required, the solenoid-operated valve (6) evacuates oil from the pilot side of the cartridge valve, instructed by a new pressure setting in the electronics. When the pilot pressure on the cartridge valve is decreased, the system pressure acting on port A, forces the valve to open and drain oil through relief valve (7) to tank, until the system pressure is equalized with the pilot pressure.

Functional description; release sequence:

1. An external force which exceeds the force from the hydraulic system pressure is acted upon the piston rod of the cylinder (2). This result in an increase in system pressure, differentiated from the pilot pressure on the cartridge valve, forcing the valve to open. Note: Main circuit and pilot circuit do not share oil during a release sequence due to the check valve (5). Secondly; oil which is evacuated through cartridge valve is led towards the piston side of the cylinder and through pressure relief valve (7) to tank.
2. When the obstacle is passed by, the external force is relieved. The system pressure decreases, and the pilot pressure makes the cartridge valve close, while finally equalizing the pressure with the main system pressure, by naturally forcing the check

valve to open. Meanwhile the plough body is retracting back to the soil; directional valve (6) is instructed to open in order to add oil back to both the main system and pilot system to pursue constant system pressure.

Alternative 5: Cartridge valve system with common pilot- and system pressure

This system is based on the same use of the cartridge valve (4) with the same functional principle as the system within alternative 4. The only difference is that this system is not exploiting the opportunity of feeding oil to the opposite side of the cylinder piston in a regenerative loop. Instead oil is led through the cartridge valve directly to tank during a release sequence. Hydraulic schematics are shown below, and for the functional description it can be referred to the same description as for alternative 4, due to the similarity described between those two.

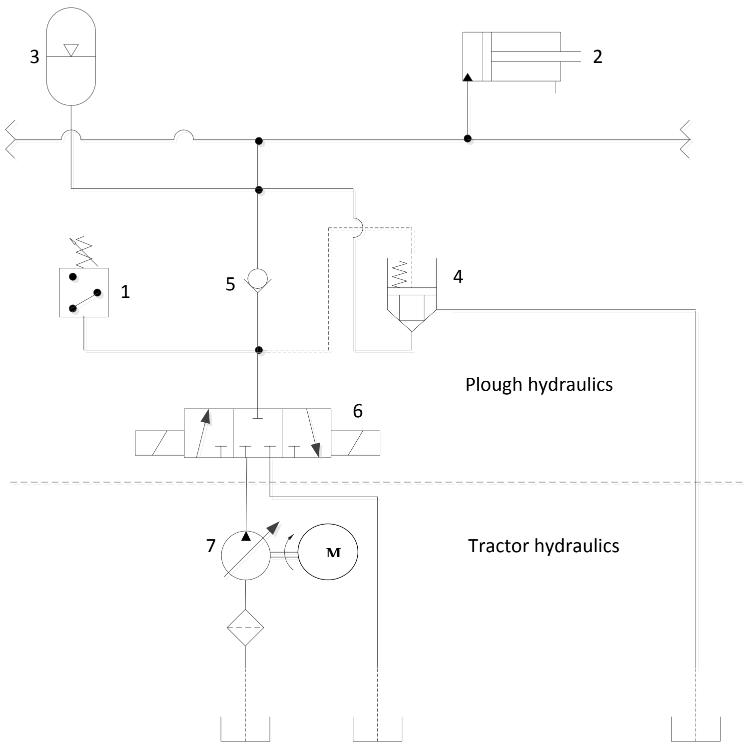


Figure 5.5: *Hydraulic schematics.*

Alternative 6: Cartridge valve system with separate pilot- and system pressure

This system is another variant of alternative 5. But the pilot oil supply is totally separated from the main system with its own directional valve (7) and pressure switch (4). This example can be an option in the situation where there is a differentiation between the pressurized area on the pilot port and the A-port of the cartridge valve. In this situation different pressures will be needed on each side of the valve to maintain an even force between the two sides. The two systems will in terms of different pressure requirement need separate control with a separate directional valve and pressure switch.

Functional description; adjusting system pressure:

1. Required system pressure is fed into the electronic control and compared against the pressure measured by pressure switch (1). Simultaneously, the required pressure in pilot circuit is calculated and compared to the pressure logged by pressure switch (4).

2a. If a pressure increase is required, the solenoid-operated directional valve (7) adds oil to the pilot circuit in order to keep valve (5) closed with the new intended pressure setting in main circuit. At the same time oil is added to the main system through valve (6) until reaching that new system pressure setting.
2b. If a pressure decrease is required, the solenoid-operated valve (6) evacuates oil from the main circuit, instructed by a new pressure setting from the electronics. At the same time oil is drained from the pilot side through directional valve (7) until correct pilot pressure is achieved.

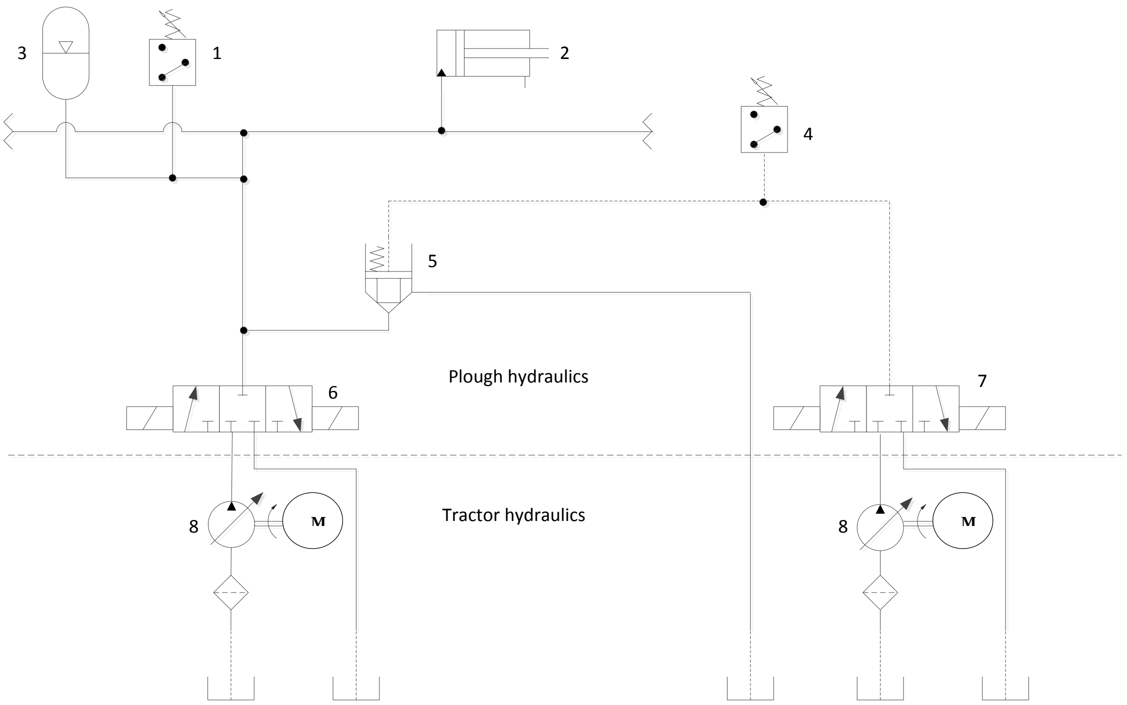


Figure 5.6: *Hydraulic schematics*

Functional description; release sequence:

1. An external force which exceeds the force from the hydraulic system pressure is acted upon the piston rod of the cylinder (2). This result in an increase in system pressure differentiated from the pilot pressure on the cartridge valve (5), forcing it to open, relieving main system.
2. When the system pressure drops at the end of the system release, oil is then fed back to the system through directional valve (6). Meanwhile the pilot pressure is corrected in accordance with the system pressure at all times.

5.2 Mechanical design

To try to keep a design that still implies a cylinder mounted beside the beam, the possibilities of altering the position of the cylinder end fix and its connection point with the tie plates will be looked into. This redesign is due to the fact that the calculated release characteristics of the mechanical design proposed so far, is not ideal, as previously mentioned. This will imply a change in moment arm relation 2. Changes to the beam itself or existing parts connected to it is not an option. That means that moment arm relation 1 is fixed. Note that the release characteristics have so far been calculated as a function of release height. Since oil flow and velocity of the cylinder becomes of greater importance in further calculations, the release characteristics is from now on calculated relative to time, provided a driving velocity of 8 km/h. The method of obtaining the release characteristics can once more be found in appendix A1, and related calculations on the release characteristics can be found in Excel file: "Calculated mechanical release characteristics concept designs.xlsx". Graphical output and comparison between release characteristics is located at the end of this chapter for the 3 design alternatives presented.

The plough beam- and tie rod design used as part of the mechanical design for this product is the result from earlier work performed at the Research and Development Department at Kverneland Group Klepp, obtained from (Kverneland Group 2012). However tie plate design and the specification of "moment arm relation 2", along with all calculations on release characteristics for this product are result of own work.

In figure 4.5 in the previous chapter, the design base for moment arm relation 2 is illustrated. Two new proposals similar to that design have been worked out. Both designs are intended to give more acceptable release characteristics than the previous proposal. Furthermore, design alternative 1 and 2, as they are referred to later, will still consist of the same type of components. The required change in design is achieved by altering the end point of the cylinder towards the tie plate. Another detail is that the cylinder fixes are interchanged, leaving the cylinder tube in front of the beam. This is in order to position the hydraulic hosing closer to the plough, saving consumption of hoses while serving the purpose of a cleaner design. As a third solution outlined, is the option of placing the cylinder at the rear of the beam.

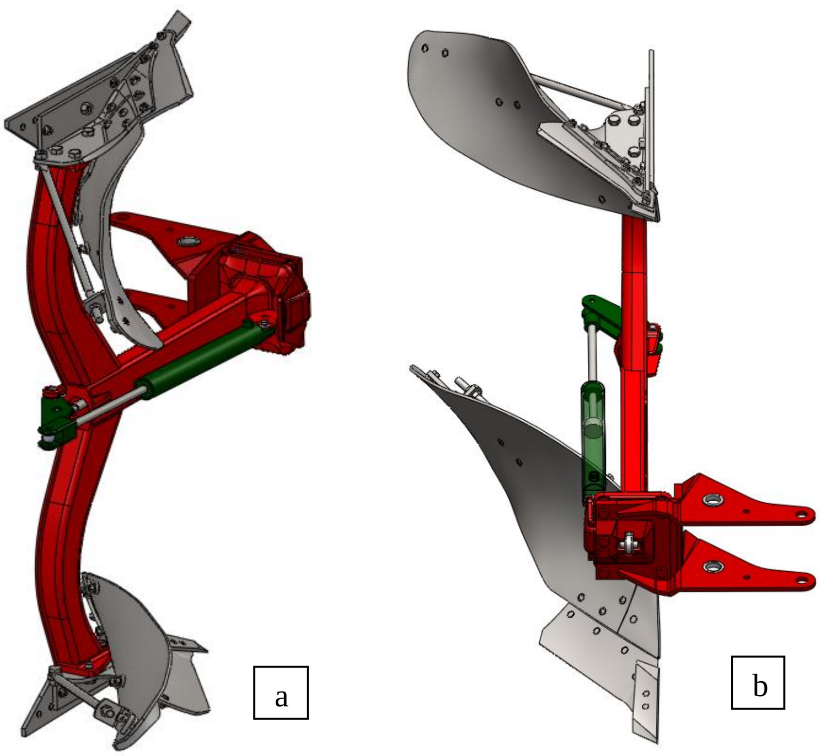


Figure 5.7 a, b: *This is the how the design of alternatives 1 and 2 will basically look like from a more rear view (a) and up-front view (b). Illustration is part of own work on the basis of design obtained from (Kverneland Group 2012).*

Alternative 1

Figure 5.8: *The design is made a bit more extreme by placing the cylinder/tie plate interconnection further behind the beam. This results in an increase in moment arm z6 during the whole release sequence.*

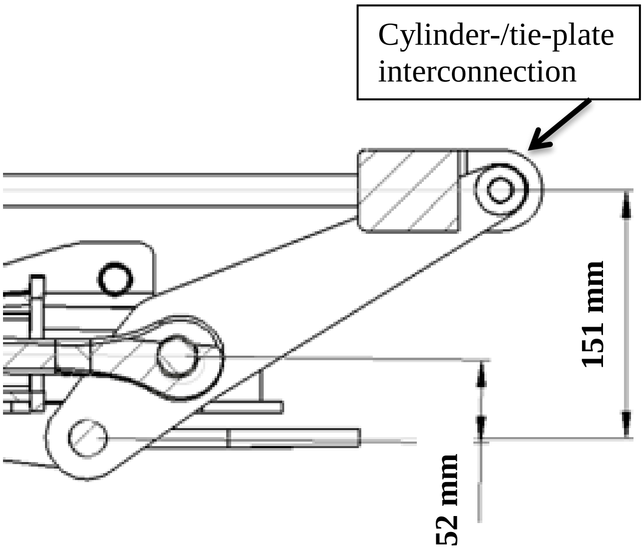


Table 5.1: *Initial moment arm relation, force-on-point and hydraulic pressure for use in further calculations for alternative 1.*

Properties	Measurement
Moment arm relation	0.3
Force-on-point	16 kN
Hydraulic system pressure	160 bar (16 MPa)

Calculate the initial axial force on the cylinder:

$$F_{cyl} = \frac{16\ kN}{0.3} = 53.33\ kN \quad (Eq\ 1)$$

Calculate the needed piston pressure area and diameter of the cylinder in order to withstand that force with the 160 bar pressure setting:

$$A_{cyl} = \frac{53\ 330\ N}{16\ MPa} = 3333\ mm^2 \quad (Eq\ 2)$$

$$d_{cyl} = \sqrt{\frac{4 * 3540\ mm^2}{\pi}} = 65\ mm \quad (Eq\ 3)$$

Table 5.2: *Brief calculation of mass of new components. Mass is calculated by the Solid Works CAD software.*

COMPONENT	MASS
Connecting plate	1.31 kg * 2
Cylinder tube	3.65 kg
Piston rod	1.61 kg
Rod end	1.29 kg
SUM	**9.17 kg**

Alterative 2

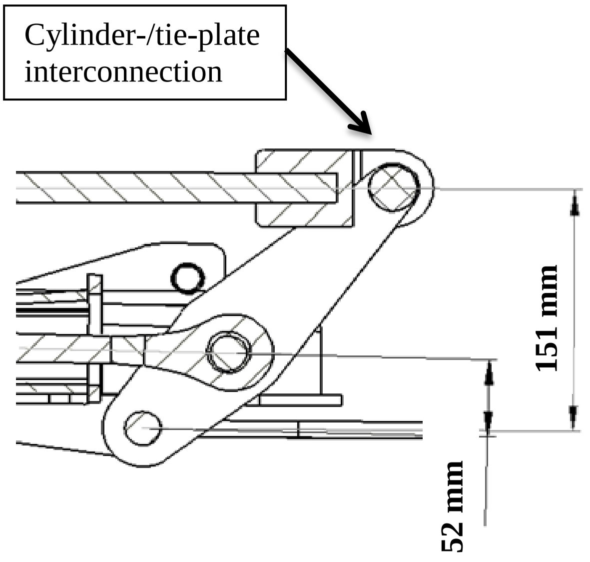


Figure 5.9: *This design is based on alternative 2, with the same moment arms before release, but the interconnection between the cylinder rod end and the connection plate is brought closer to the plough beam in order to get a more compact design.*

The initial moment arm relation by 0,3 implies the same axial force on the cylinder upon release as for alternative 1, and subsequently this implies the same cylinder piston diameter as well. So for the calculation of that cylinder piston diameter it can be referred to the previous calculations performed for alternative 1.

Table 5.3: *Brief calculation of mass of new components. Mass is calculated by the Solid Works CAD software.*

COMPONENT	MASS
Connecting plate	1 kg * 2
Cylinder tube	3.65 kg
Piston rod	1.38 kg
Rod end	1.29 kg
SUM	**8.32 kg**

Alternative 3

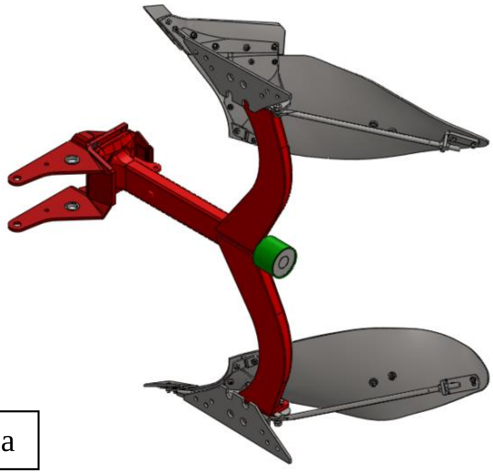
An alternative might be to reconsider the option of locating the cylinder at the rear of the beam, which was one of the other design alternatives outlined in the pre-project as well. The release characteristics will then be defined of the first moment arm relation only.

Table 5.4: *Initial moment arm relation, force-on-point and hydraulic pressure for use in further calculations on alternative 3.*

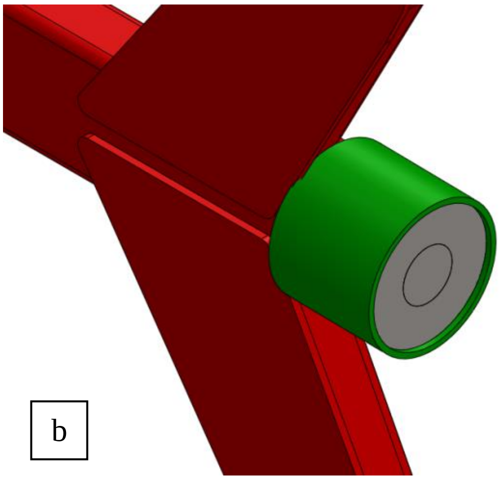
Properties	Measurement
Moment arm relation	0.1
Force-on-point	16 kN
Hydraulic system pressure	160 bar (16 MPa)

Calculate the initial axial force on the cylinder:

$$F_{cyl} = \frac{16\ kN}{0.1} = 160\ kN \quad (Eq\ 1)$$



a



b

Figure 5.10 a, b: *Illustration of the design of alternative 4, with the cylinder mounted directly onto the rear part of the plough beam. Shows own design proposal based on design obtained from (Kverneland Group 2012).*

Calculate the required piston pressure area and diameter of the cylinder in order to withstand that force with the 160 bar pressure setting:

$$A_{cyl} = \frac{160\ 000\ N}{16\ MPa} = 10\ 000\ mm^2 \quad (Eq\ 2)$$

When putting the cylinder right behind the beam and connecting the piston rod of the cylinder in direct connection with the tie rod, the rod side of the cylinder piston will have to be the pressure side of the cylinder. This naturally implies subtraction of the cross-sectional area of the piston rod to obtain the effective pressurized area and subsequently the required piston diameter. Dimensions of the piston rod has to be investigated more carefully in the case of choosing this design alternative, but for use in this basic calculation, d = 40 mm as piston-rod diameter is chosen for the further calculation of piston diameter:

$$A_{rod} = \frac{\pi * 40^2}{4} = 1257\ mm^2 \quad (Reworked\ from\ Eq\ 3)$$

$$A_{piston} = 10\ 000 + 1257 = 11\ 257\ mm^2$$

$$d_{piston} = \sqrt{\frac{11\ 257 * 4}{\pi}} = \mathbf{119.72\ mm} \quad (Eq\ 3)$$

Minimum required piston diameter is 120 mm.

Table 5.5: *Brief calculation of mass of new components. Mass is calculated by the Solid Works CAD software.*

COMPONENT	MASS
Cylinder tube	2.03 kg
Piston rod	4.32 kg
Inside rod	0.64 kg
SUM	**6.99 kg**

Release characteristics overview

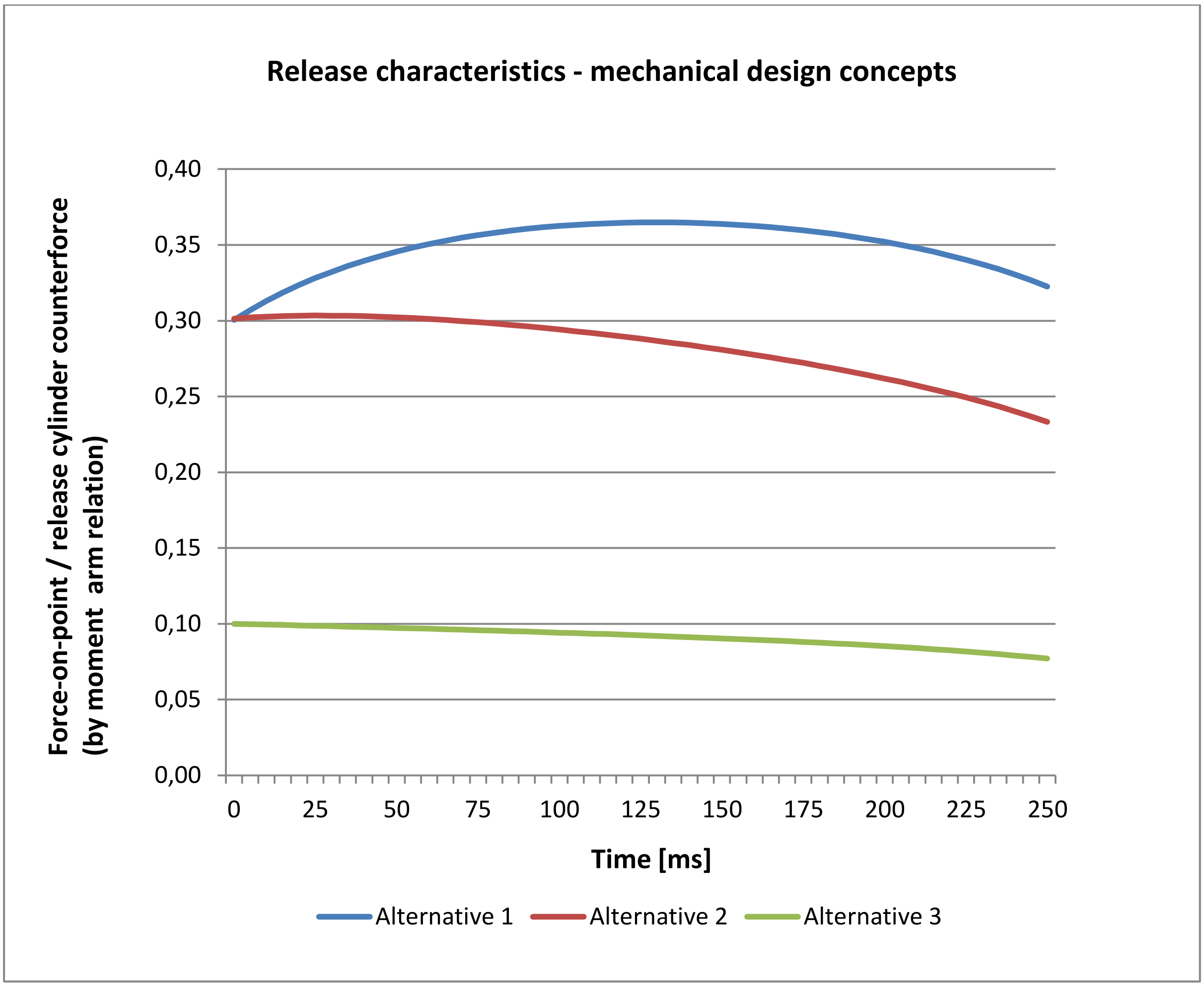


Figure 5.11: *This graphical output of the release characteristics for the three designs presented, reveal some of the differences in initial moment arm relation and decrease in release resistance between the designs. Alternative 1 has characteristics that are increasing by 20% at the most. Alternatives 2 and 3 show the same tendency in terms of decrease in characteristics, but with different initial moment arm relations.*

The release characteristics revealed for alternative 1 shows a major increase in system resistance for the first half of the calculated sequence. It drops off at the end of the sequence, but the system release resistance is still higher at the end than in the beginning of the sequence. We can already now say that design alternative won't work out in order to follow the product specification, where it is specified that the characteristics should imply a decrease in release resistance for the sequence in total.

Alternatives 2 and 3 have much more acceptable characteristics, by gradually decreasing throughout the sequence. We can see that alternative 3 that has the cylinder mounted directly to the rear part of the plough beam and exploit only the first moment arm relation, has an initial moment arm relation at one third of alternatives 1 and 2. This is compensated by using a cylinder with larger piston diameter, as shown in previous calculations on cylinder piston diameter, in order to achieve same initial system resistance.

To get an even clearer picture of the differences between the characteristics of the three alternatives, the characteristics are obtained and displayed in terms of decrease/increase in characteristics by percentage of initial resistance that the design alternatives imply.

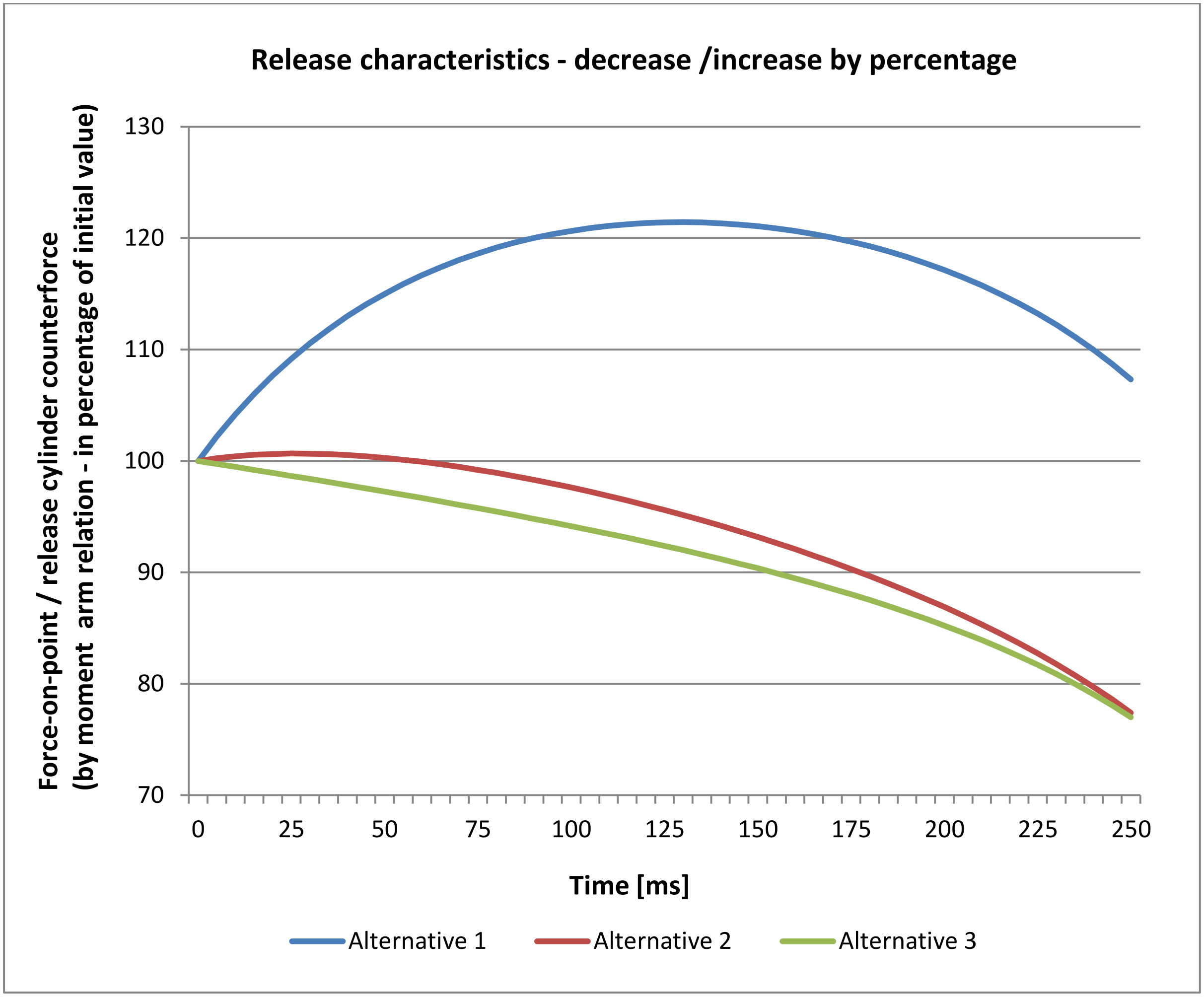


Figure 5.12: *When obtaining graphic output on release characteristics in terms of decrease/increase by percentage, we get a clearer picture of the differences in the characteristics, especially in terms of comparing alternative 2 and 3. The total decrease is almost the same for those two alternatives (22.6 % decrease with alternative 2 and 23 % decrease with alternative 3). With alternative 2 we get characteristics that maintain system resistance with even a slight increase for the first 25 milliseconds, before decreasing. The decrease in characteristics with alternative 3 is close to linear for the first two thirds of the calculated sequence.*

6. CONCEPT SCREENING AND SELECTION

From the conceptualization of both the hydraulic and the mechanical system, a selection of the alternatives outlined in the previous section is performed. This will lead to a final product concept, which will be the design basis for the further design analysis on the functional model. The concept screening consists of an internal screening, where the different design alternatives are rated with regards to their completeness in terms of the product properties outlined in the product specification.

6.1 Development of selection matrix

The concept selection will be carried out using Stuart Pugh's method, by quantifying product properties in a selection matrix. Each of the product properties is given their "rating of importance" which spans from 1-5, where 5 is most important and 1 is still of significance, but not so important. The "rating of importance" shall reflect significance of the product property related to the system performance, which is based on property listing in Table 4.3.

Next is the rating for each alternative relative to the product properties. The rating interval varies from the values of 1 to 6, where 6 fully satisfy the product property, and 1 does not at all. The score that each design alternative get on each product property is made out by the "rating of importance" for that product property multiplied with the "rating" that the specific alternative get for the respective product property.

The screening will be split into two parts; one for the hydraulic system and one for the mechanical system.

6.2 Screening

6.2.1 Hydraulic system design

Table 6.1: *Selection matrix for the hydraulic system concept selection.*

System alternatives		1		2		3		4		5		6	
Product property	**Rating of importance (1-5)**	Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score
▪ System pressure stability	5	2	*10*	3	*15*	4	*20*	5	*25*	5	*25*	6	*30*
▪ Price	4	5	*20*	4	*16*	3	*12*	3	*12*	4	*16*	2	*8*
▪ Integration of system with existing hydraulics	3	4	*12*	4	*12*	4	*12*	4	*12*	4	*12*	4	*12*
SUM			**42**		**43**		**44**		**49**		**53**		**50**

Selection, discussion and comments to screening
The system pressure stability is based on how well the system is expected to maintain a stable system pressure during the release sequence. This is given by the system capacity with regards to draining oil. Additionally; the reaction time of the valve(s) in direct interaction with the release system plays a role as well. Generally there would be expected that valves which is hydraulically operated act faster than those which are operated by electronics. However, the valve design itself also matters (e.g. whether it is a poppet or spool valve). In the selection with regards to pressure stability, the systems which have cartridge valves integrated are given the best rating/scores. This is based on the fact that they are controlled by an adjustable hydraulic pilot pressure working against the system pressure, where the hydraulic system pressure can act on and open the valve directly, without use of electronic actuation. Additionally, cartridge valves have in general better oil flow capacity than directional spool valves.

When it comes to the pricing it is more or less about making an acceptable compromise between price and functionality. Between all the alternative systems, either directional valves or cartridge valves are implemented. The directional valves are most likely to be delivered as a complete valve, while the cartridge valves need a housing to be built into. So the biggest uncertainty when it comes to the cartridge valves is how much the housing is going to cost, since a capable cartridge valve itself doesn’t need to be that expensive. But it is important to bear in mind that the price is not calculated for the different alternatives, so at this point it is only based on qualified guessing.

The system integration with existing hydraulics is a bit hard to differentiate between the 6 alternative designs. All of the systems are intended to use a directional valve that is already in use for on one of the Kverneland ploughs today, in one way or another. Except from that, there is nothing else present in the different systems that use existing components. All of the alternatives are by that reason given the same average score on that point.

Alternative 4, 5 and 6 gets the best overall scores, with alternative 5 as the very best. All three are highly rewarded by the fact that they have integrated cartridge valves to the respective systems due to the expectation of high oil flow capacity and minimum opening time of the cartridge valves. System 5 gets the final top score, due to the fact that it is a bit more cost saving than alternative 4 and 6. Alternative 4 uses a regenerative loop which means that a double-acting cylinder in addition to a pressure relief valve to tank is required. Alternative 6 comes out a bit more expensive because it uses a separate control for the pilot pressure, which implies another solenoid operated directional valve, and another pressure sensor for monitoring the pilot pressure.

The chosen hydraulic system is alternative 5, but alternative 4 and 6 is kept as optional, since there is still some uncertainty with regards to the final hydraulic system, especially with consideration to which valves that is found suitable and that can be delivered by hydraulic supplier.

Chosen hydraulic system: Alternative 5, with alternative 4 and 6 as optional

6.2.2 Mechanical design

Just as for the selection of the hydraulic system design, a selection will be made on which mechanical design alternative that shall be taken further. The selection matrix works in the exact same way as before, with a rating from 1 to 6 for the different design alternatives relative to how they fulfill the respective product properties. Each product property has a "rating of importance" ranging from 1-5.

Table 6.2: *Selection matrix for the mechanical design concept selection.*

System alternatives		1		2		3	
Product property	**Rating of importance (1-5)**	Rating	*Score*	Rating	*Score*	Rating	Score
▪ Release characteristics	5	3	*15*	5	*25*	5	25
▪ Price	4	4	*16*	4	*16*	4	16
▪ Integration of system with the existing plough design	3	6	*18*	6	*18*	5	15
▪ Mass	2	3	*6*	4	*8*	4	8
▪ Aesthetic design and compactness	1	2	*2*	4	*4*	5	5
SUM			**57**		**71**		**69**

Discussion and comments to screening

At this point of the process there are only the release characteristics and the mass of the mechanical components that can be truly quantified. How well the different alternatives score on price is not made out of any estimates, but on more or less qualified guessing. How well the system fits integration with the existing design and how well an aesthetic design is accomplished, is rather difficult to quantify in the first place. The assessment of the design is completed in an as objective manner as possible, but also reflects some subjective assessment. Since some of the product properties are a bit hard to specify in terms of a quantitative measure, it has been chosen to look a bit beyond those criteria, by focusing even more on the quantitative measurements like release characteristics particularly. This is also the most important product property, and essential for the functionality of the product and the later testing in this project. Therefore the functionality requirement will overrule the consideration of design, which is not really essential to the product and the impending testing of this project. It can rather be dealt with more in detail at a later point, if the functional test proves successful.

There is a close final score between alternative 2 and 3 on the mechanical design, while alternative 1 get a lower score, which basically means that it can be ruled out. Alternative 1 is suffering from too much increase in the release characteristics just after release, and is therefore given a low score on that particular point. Due to the close score, it is hard to separate alternative 2 and 3, especially since the release characteristics are pretty much similar, implying practically the same final decrease in force-on-point at the end of the sequence. But since alternative 2 is a bit easier to integrate in an acceptable manner into the plough beam than alternative 3, as the plough beam design is today, the final choice of mechanical design is alternative 2. However alternative 3 is still an alternative to keep in mind in the longer run, especially if there is possible to implement changes to the plough beam to make the solution fit in better.

Chosen mechanical design: Alternative 2

6.3 External feedback on hydraulics

The chosen hydraulic system (alterative 5) along with the two optional solutions (alternative 4 and 6) with their hydraulic schematics has been distributed to hydraulic supplier Sauer-Danfoss and their service department in Norway. The aim has been to get an assessment on which hydraulic valves that might be possible to implement into the system, holding the adequate capacities and functionality.

Through e-mail and telephone contact their evaluation and further proposals for the system and use of components has been received.

There were some challenges finding suitable valves due to the high system oil flow and the short duration of each single release sequence. Many cartridge valves had the sufficient oil flow capacity, but other issues regarding their performance were raised. Some concern was especially directed towards the question whether some valves would actually be capable of empty the system of pressure due to high oil flow capacity, if not closing in time.

After a period of research on valve options in cooperation with the Sauer-Danfoss service contact, suitable solutions were found on two types of cartridge valves implemented into hydraulic system alternative 5. The process of finding suitable valves has been an ongoing part of the "*design analysis and product architecture"*, chapter 7. Valves and choice of them are described more closely in that chapter.

7. DESIGN ANALYSIS AND PRODUCT ARCHITECTURE

From the conceptualization phase a final design has been chosen, both for the hydraulic and mechanical system. In this section, the more detailed specifications of the hydraulic components that are going to be a part of the system will be defined. Fluid mechanics calculations on the system behavior will be performed. For the mechanical system design detailing on new components specified within the system will be carried out. As specified in the project limitations, this regard the tie plate in its entirety.

Following the design phase, a functional test will be performed, as already mentioned. The test results will be compared against the calculations performed in this chapter, as far as it is possible.

The characteristics and the basis for the calculations performed in this chapter are mainly based on calculations and graphical outputs from Excel-document named "Calculation basis chapter 7", which is referred to in appendix A1.1.

For reference to equations used in calculations throughout the report, see "Methods" and Table 3.4. For literature references on equations used and listed in Table 3.4, see table text in same table. For special equations and terms used in calculations, not listed in Table 3.4 and Table 3.5, literature references are listed through calculations, both in this chapter and in further calculations throughout report.

Calculations regarding the hydraulics throughout the report partly differ from SI measurement by using pressure units in "bar" ($=10^5$ Pa) and volumetric flow converted from SI unit (m^3/s) by using "liters/minute" (m^3/s = 60 000 L/min). Those units are used, caused by the fact that they are the most common with relevance to indicated capacities used on hydraulic components. For reference to units of measure and use of converted unit of measure, see Table 3.1, 3.4, and 3.5.

7.1 Release characteristics

Once more the release characteristics are essential to the system. The release characteristics are basically defined by the mechanical part of the system, but are highly essential to the hydraulic design, since it has a significant impact on the oil flow characteristics of the hydraulic system. For that reason, the design phase is commenced by once more specifying the release characteristics of the chosen design.

As mentioned and seen earlier in the report, the release characteristics is defined with regards to time in order to use these calculations further. This will make it possible to define system properties like oil flow during the sequence relative to time.

When we know the time duration and the plough point horizontal (x-directional) displacement through the sequence, each time interval within that sequence can be seen in relation with horizontal travel of the plough point for that interval. Vertical travel of the plough point can be related to the horizontal travel by the use of the Pythagoras' dogma, as illustrated below. Furthermore; combining this relation with the two moment arm relations in the mechanical set-up, the final characteristics can be obtained, once more using the method shown in appendix A1.

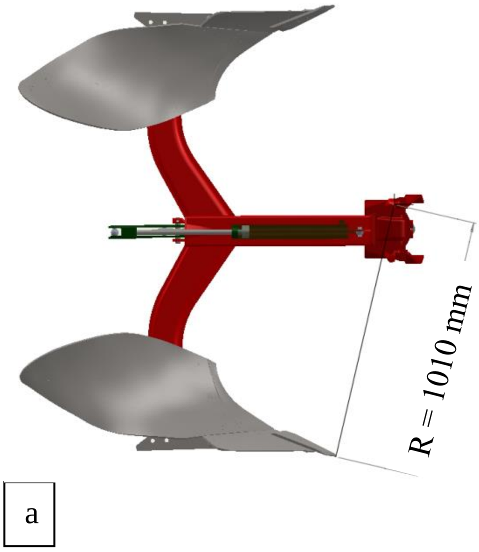


Figure 7.1a: *First of all we do have a fixed distance between the upper ball link on the beam holder and the plough point, respectively. We can exploit this fact and get the displacements in x- and y-direction by simply using Pythagoras' dogma. Illustration reworked from (Kverneland Group 2012).*

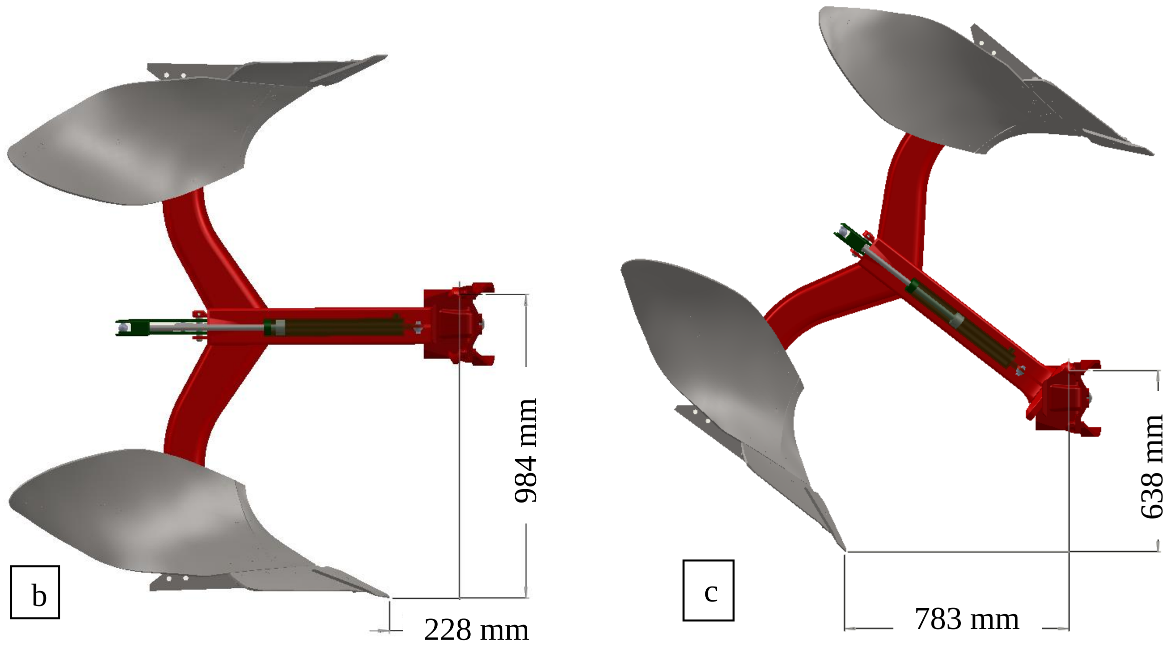


Figure 7.1 b, c: *These illustrations show the x- and y-values in nominal body position, and at a release height of 350 mm. As mentioned, the hypotenuse illustrated above is fixed between these two variables. Illustrations reworked from (Kverneland Group 2012).*

The release sequence is defined with a release height of 350 mm. The time duration of the sequence is obtained by calculations below.

Define the travel velocity in x-direction:

$$v_x = 8\frac{km}{h} = 2.22\ m/s$$

From the x/y relation in displacement of the plough point during the sequence, the displacement of the plough point in x-direction is obtained (see figure 7.1 and calculation details in appendix A1):

$$\Delta x = 783\ mm - 228\ mm = 555\ mm = 0.555\ m$$

Calculate the time duration of the sequence:

$$\frac{\Delta x}{v} = \frac{0.555\ m}{2.22\frac{m}{s}} = 0.25\ s = 250\ ms$$

When calculating the release characteristics, weight of plough beam and mechanical friction has not been regarded. There are basically two types of friction that have been neglected in the mechanical part of the system:

- External friction, which is the friction between surfaces of mechanical parts connected together
- Internal friction, which is the friction present within the solid materials while absorbing energy

Consequently, the energy absorption in the mechanical part of the system has not been regarded.

When it comes to the hydraulic system, energy loss will be calculated, which also implies calculation of pressure loss throughout the system. This is however based on a theoretical calculation and will not be fully comparable to the functional test, since the calculation only deals with loss in pipeline system, without regarding parts of the hydraulic circuit, like cylinder and valves.

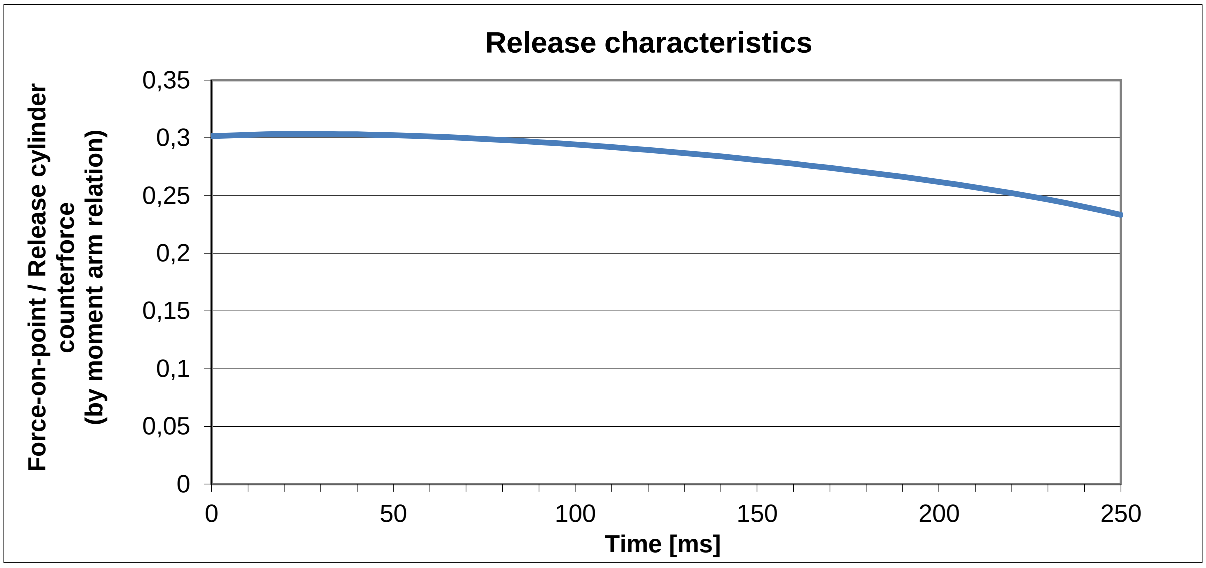


Figure 7.2: *Release characteristics as a function of time, x(t) in phase 1 of release sequence for the chosen mechanical design. Graphical output and calculations can be found in digital copy of Excel document related to calculations performed in chapter 7, as previously mentioned. See appendix A1.1 for more information.*

7.2 Hydraulic system design

7.2.1 Calculation basis

In order to make the final decisions when it comes to the hydraulic components, basic calculations in order to obtain system characteristics must be carried out. Listed below are the aspects that are important to get hold of in terms of system behavior, its characteristics and consequently the needed capacity of the hydraulic components.

System quantities to calculate:

- Oil flow
- Energy and mass in system

From now on and further down the calculations, what is meant by the "system" when calculating its mass and energy, is the oil volume displaced by the cylinder travel during the release sequence – not the entire volume and mass within hydraulic circuit.

System- oil flow and pressure for chosen release cylinder

From the information given in chapter 5 about the chosen concept design, a cylinder piston diameter by 65 mm is proposed. When selecting a suited cylinder for the functional test, using a cylinder within cylinder range of Kverneland Group Klepp will be pursued. The closest to the given specification is a cylinder with piston diameter of 63 mm, which is close enough to the cylinder specification for this test. This is just below the previously calculated piston diameter. However, the system pressure needs to be increased in order to hold the same maximum release resistance in terms of the 16 kN on the plough point, when accounting for the new piston diameter. Note that the intension in the long run, is still to develop a system as earlier described with a maximum pressure setting by 160 bar, according to the product specification. But to develop a new cylinder only for the purpose of this test is too expensive. So for the further calculations and the testing, cylinder diameter by 63 mm will be used.

Calculate new piston cross-sectional area:

$$A = \frac{\pi * (63\,mm)^2}{4} = 3117\,mm^2 \quad (Reworked\ from\ Eq\ 3)$$

Recalculate maximum system pressure (for use in calculations):

$$p = \frac{53\,000\,kN}{3117\,mm^2} = \approx 17\,MPa = 170\,bar \quad (Reworked\ from\ Eq\ 2)$$

The average oil flow present can be calculated by following equation:

$$Q_{avg} = v_{avg} * A \quad (Reworked\ from\ eq\ 9)$$

From the calculations performed as in appendix A1 and furthermore obtained from documentation in Excel for calculations in chapter 7, the average cylinder velocity is:

$$v_{avg} = 0.6\frac{m}{s}$$

$$Q_{avg} = 600\frac{mm}{s} * 3117\,mm^2 = \ 1.87 * 10^{-3}\frac{m^3}{s} = 112\ L/min$$

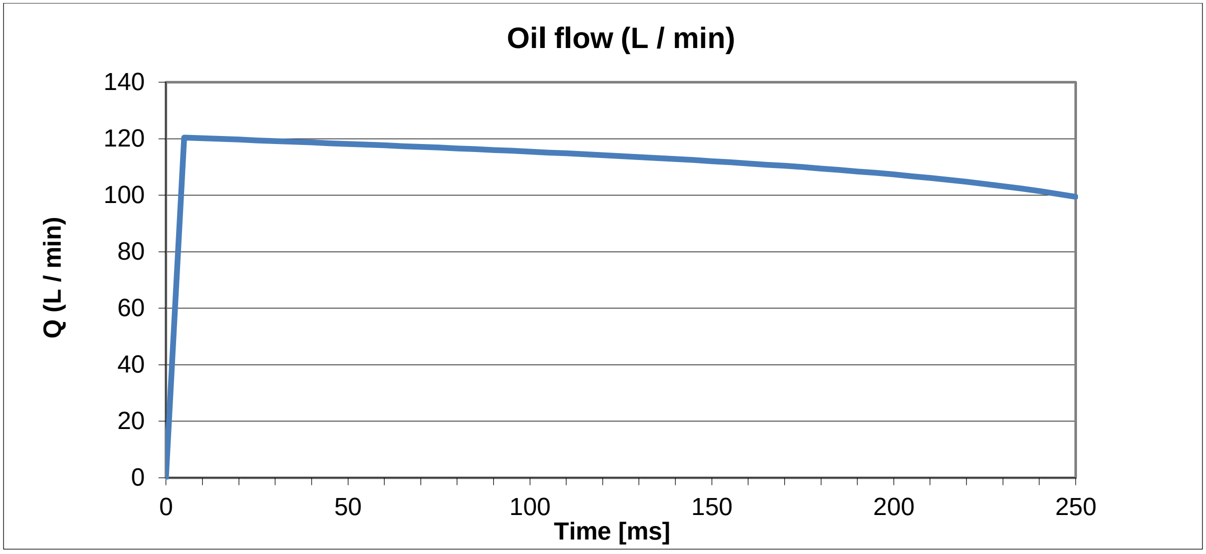


Figure 7.3: *The oil flow obtained from Excel sheet: "Calculation basis for chapter 7". Q_{max} = 120 L/min.*

Mass in system

When calculating the mass of the system, we use the following equation:

$$m = \rho * V \quad (Eq\ 4)$$

In order to calculate the system volume, oil flow is regarded in relation with time. The system volume is obtained by the following integral:

$$V = \frac{1}{2}\int_{t_0}^{t_n} Q(t)\,dt \quad (Eq\ 5a)$$

By separating into 5 ms time intervals as done in Excel, gives the following expression:

$$V = \frac{1}{2}\left[\int_{t=0}^{t=5} Q(t)\,dt + \cdots + \int_{t=200}^{t=250} Q(t)\,dt\right]$$

By obtaining average oil flow as already calculated to 112 l/min, we get the following system oil volume over the calculated release sequence by 250 ms:

$$Q = 112\frac{L}{min} = 0.00187\ \frac{m^3}{s}$$

$$V = 0.00187\,{}^{m^3}/_{s} * 0.25\ s = 0.000468\ m^3 = 0.468\ L \approx 468\ 000\ mm^3 \quad (Eq\ 5b)$$

Taking the density of hydraulic oil used for the test; 865kg/m^3 (Shell 2013), gives the system mass:

$$m = 865\ {}^{kg}/_{m^3} * 0.000468\ m^3 \approx 0.405\ kg \quad (Eq\ 4)$$

Energy in system

The energy in the system can be regarded in terms of either the potential energy in the system upon release, or the total kinetic energy elapsed for a complete release sequence. The energy in this case, however, is calculated in terms of the potential energy.

<u>Potential energy</u>

The volume displacement in the system during a release sequence can be regarded as the system volume, as already mentioned. This forms the basis for calculating the potential energy in the system. Given the maximum recalculated system pressure of 17 MPa and the specified volume from the volume displacement, we can specify the amount of potential energy upon release:

$$E_p = 17\,MPa * \ 468\,000\,mm^3 = 7\,956\,000\,Nmm \ \approx \ 8\,kJ \qquad (Eq\ 6)$$

Throughout a release sequence, 8 kJ of energy needs to be absorbed by the system at maximum system resistance.

7.2.2 Hydraulic components

To serve the functionality of the system, specific hydraulic components are needed. Based on the chosen hydraulic system from the conceptualization phase, the components can now be specified in more detail. The bottleneck in the system, as experienced from preliminary project, is the system ability to drain oil to tank during the release sequence. In the chosen hydraulic system, as once more illustrated below, the cartridge valve (4) is going to serve that purpose. As already mentioned, contact has been established towards one of the Kverneland hydraulics suppliers - Sauer-Danfoss and their service department in Norway in order to be supplied with a suitable hydraulic valve to the system.

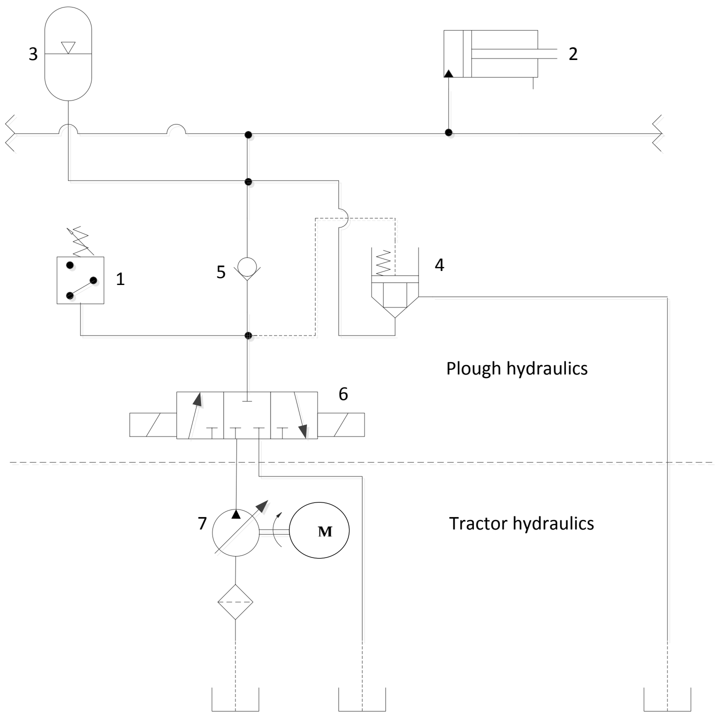


Figure 7.4: *The hydraulic schematics of the system, with all hydraulic components integrated to the system on plough. This includes: cartridge valve (4), directional valve (6), hydraulic cylinder (2) gas accumulator (3) and pressure switch (1) that need to be specified.*

Hydraulic cylinder

As already specified, the applicable cylinder piston diameter is now 63 mm. Furthermore, the necessary cylinder displacement can be calculated from the average cylinder travel velocity and the time duration of the release sequence. We can regard the calculation of cylinder travel in terms of following integral:

$$s_{cyl} = \frac{1}{2}\int_{t_0}^{t_n} v(t)\,dt \qquad (Eq\ 7a)$$

By separating into 5 ms time intervals, gives the following expression for the calculation of cylinder travel distance throughout sequence:

$$s_{cyl} = \frac{1}{2}\left[\int_{t=0}^{t=5} v(t)\,dt + \cdots + \int_{t=200}^{t=250} v(t)\,dt\right]$$

Once more, from calculation according to method applied in appendix A1, the average cylinder velocity can already be obtained from Excel document in relation to calculations in chapter 7:

$$v_{avg} = 0.6\frac{m}{s}$$

Hence, the cylinder displacement within the 250 ms interval can be calculated:

$$s_{cyl} = \ 0.6\frac{m}{s} * 0.25 = 0.15\,m = 150\,mm$$

The stroke length of the cylinder might just as well be a bit more, but 150 mm of stroke length is a minimum.

Cartridge valve

For the task of evacuating oil during release sequence, a proper cartridge valve shall be chosen prior to testing.

Table 7.1: *The vital properties of the cartridge valve are listed in this table, and form the basis for the selection of the specific valve.*

Properties	Measurement
Minimum oil flow capacity	120 L/min
Area differentiation; pilot port / pressure port	1:1

As seen from the table, the needed oil capacity of the valve for the test is 120 l/min, by means of the previous calculations. Additionally it should have a 1:1 area ratio between pilot port (x) and the pressure port (A). This is a prerequisite to the system, since the pilot system and the main system will share a common pressure, except during release, when check valve (5) from Figure 7.4 block the connection between the two systems. The weak spring in the valve will make the pilot side of the valve biased, so that the valve doesn't open until pre-set system pressure is exceeded. However, the additional spring force is so weak that it doesn't interfere with the release resistance in any way worth mentioning.

Chosen valve

With regards to the valve subject for the testing, there are two alternatives that are going to be tested; one sequence valve and one pressure relief type of valve. This is a result of the research on choice of valve and technical advice from Sauer Danfoss representative on basis of hydraulic system specified from concept generation -and selection. The sequence valve has the desired functionality in terms of a pilot pressure port at the spring chamber, facilitating the hydraulic adjustability the system should have. However, there is a minor uncertainty with regards to the oil flow capacity, so a secondary valve with the sufficient oil flow capacity has been included to the test as an optional valve. The downside to this secondary valve, is that it only facilitating manual pressure adjustment, so it will not be an alternative in the long run, but sufficient for testing system capacity at this stage.

Table 7.2: *Chosen cartridge valve 1. (Comatrol-1 2010)*

Valve description	Comatrol CP241-21 – Pilot-operated sequence valve
Oil flow capacity	190 L/min (75 L/min at 7 bar rated pressure drop)
Pressure adjustment	Hydraulic pilot + compressible mechanical spring

The sequence valve have the required 1:1 pressure area ratio between piloted spring chamber and pressure port, which means that pilot-pressure source towards spring chamber can share pressure adjustment oil supply with the main system, as long as it is separated by a check valve during release.

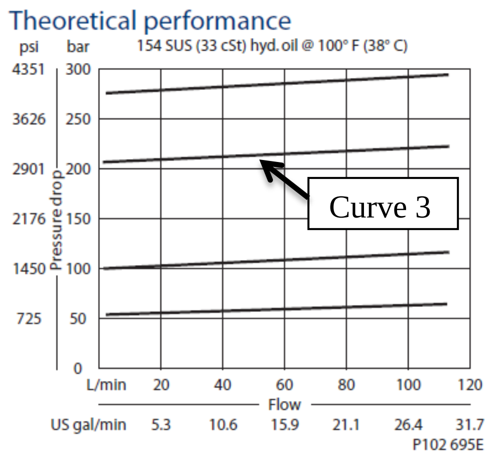


Figure 7.5: *This shows the theoretical performance of the CP241-21 valve rated for flows just above 110 L/min assuming hydraulic oil with 33 cSt (centi-Stokes kinematic viscosity at 38°C. We can read out an approximate pressure drop about 20 bar at a preset pressure just above 200 bar looking at "curve 3". Reworked from (Comatrol-1 2010)*

With technical information retained from (Forbord 2013) we should expect the CP241-21 to have a maximum flow capacity equal to the CP211-2 described below, which is rated at 190 L/min, despite previous uncertainties.

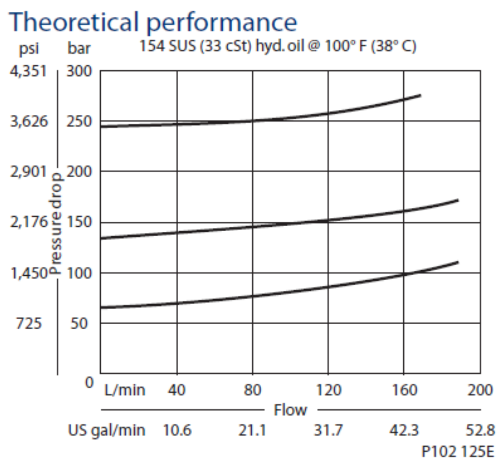


Figure 7.6: *Theoretical performance of CP211-2 valve assuming hydraulic oil at 33 cSt at 38° C. The pressure drop increases in an exponential manner with increase in flow. (Comatrol-2 2010)*

The mechanical adjustable valve has been chosen in order to have a "back-up" valve with the documented capacity in case the sequence valve should fail.

Table 7.3: *Cartridge valve, alternative 2. (Comatrol-2 2010)*

Valve description	Comatrol CP211-2 – Pilot-operated relief valve
Oil flow capacity	190 L/min
Pressure adjustment	Compressible mechanical spring

Directional valve

A solenoid-operated directional valve is needed in order to adjust system pressure. Additionally it will at a later point be implemented in order to feed oil back into the system in the second phase of the release sequence. However, the functional test model in this project will only have the functionality of plough body release (release phase 1) and will not be able to make the body retract automatically. The valve shall be operated by an electronic control system. The electronic control system is described more thoroughly later in this chapter.

Table 7.4: *Chosen directional valve for system. (Sauer-Danfoss 2012)*

Valve description	Sauer Danfoss PVG32 directional valve
Oil flow capacity	100-130 L/min
Electronic actuation	PVEO – On/Off

When looking at the oil flow capacity of the directional valve, it would actually be just enough to meet system requirement at highest rated oil flow for the system as it is designed at this point. However, the reaction time of the valve is still not sufficient in order to serve as main valve by oil evacuation during release.

Gas accumulator

According to metric limit specification, the pressure deviation should be maximum 10% from the nominal pressure setting at any time during the release sequence. The lack of capacity in valve system must alternatively be covered by the capacity of the accumulator. However, when looking at the flow capacities of the chosen test valves, the oil flow capacity is sufficient to drain the oil flow present during the release sequence. If assuming no dead time between reaching relief pressure and valve activation, and neglecting reduced oil flow capacity between valve activation and until it is fully open, there is no need for extra accumulator volume capacity other than for absorbing minor shocks to the system. Therefore, no further calculations on accumulator volume are performed, since it is not proven to be required at this point. It might actually be rather beneficial for the test and its output to have a smaller accumulator volume. In that way, system characteristics and possible lacks in capacity characterized by pressure peaks can be obtained more easily. Alternatively a bigger accumulator can be chosen for a re-test, if the system fails to meet the system requirements.

Table 7.5: *Chosen gas accumulator for the testing.*

Volume capacity	0.5 L

Hydraulic hoses

Another essential part of the system is the hosing, in order to assure that the hosing and pipeline between the cylinder and the valve system is designed for the designated oil flow. In the later section, fluid dynamic calculations are performed with focus on doing an analysis with regards to the hydraulic hosing and the flow in the pipeline.

7.2.3 Electronic control system

The electronic control system shall consist of a pressure switch, plough ECU (Electronic Control Unit) consisting of a PCB (printed circuit board), user interface including the required software, actuators for the hydraulic directional valve, along with the necessary wiring.

Pressure sensor/switch

In the prospective final system, the pressure switch shall serve as the interface between the hydraulic and the electronic system, using the measured pressure to control the direction of oil in and out of the system by the directional valve. For this first functional test model, the pressure sensor will only be used to measure pressure, whereas the directional valve will be manually controlled by operator through electronic system interface.

Table 7.6: *Chosen pressure sensor. (Hydac International)*

Pressure transmitter type description	Hydac HDA 4445-A-250-000
Pressure range	0 - 250 bar

Plough ECU with printed circuit board

For this system, the existing ECU used on the Kverneland PW plough shall be used.

User Interface

Selection of hydraulic pressure as system input and monitoring system output will be achieved by means of the Kverneland Isomatch Tellus terminal (or other compatible terminal) that shall be used as the system user interface. This includes the required software, which will be taken care of by Kverneland Mechatronics. The user interface and the basic functionality of the electronic software will be described more closely in the later specification of the functional test.

7.2.4 Fluid dynamics and pipeline flow

Fluid dynamic calculations are carried out in order to assure a pipeline design that can deal with the present oil flow, as far as possible. Furthermore these calculations will be used to estimate pressure- and energy losses through pipeline system. The calculations in this section will as far as possible, be compared to the results obtained in the later testing.

The pipeline forms the basis of the further calculations and the functional testing is designed as illustrated below. The pipeline system can be looked upon as a grid of nodes and elements, where nodes represents interconnection points between pipeline elements, and the elements are simply the different parts of the pipeline, as shown in Figure 7.9.

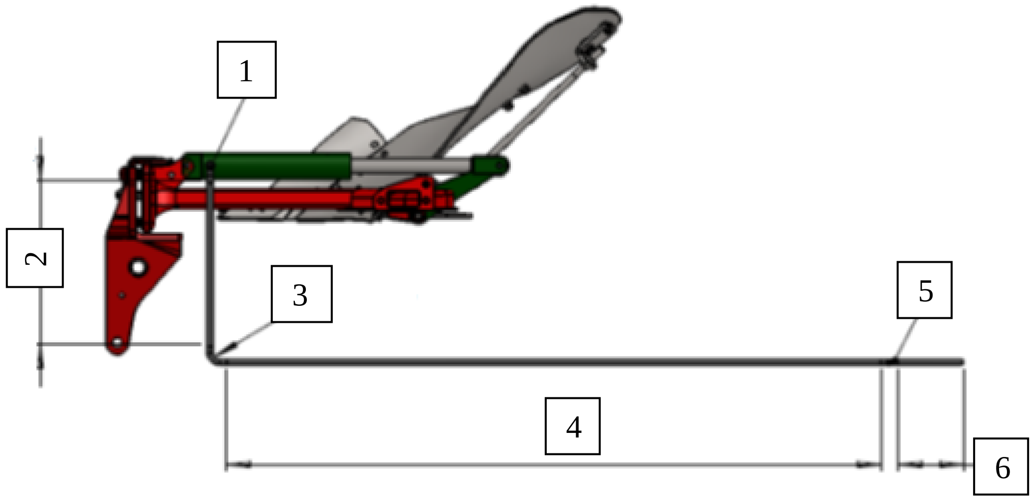


Figure 7.7: *The illustration shows the pipeline system connected to the cylinder on the plough beam. The numbered boxes in this illustration are referring to elements listed in Table 7.7. Pipeline system performed by own design. Plough beam design based on 3-dimesional computer model obtained from (Kverneland Group 2012).*

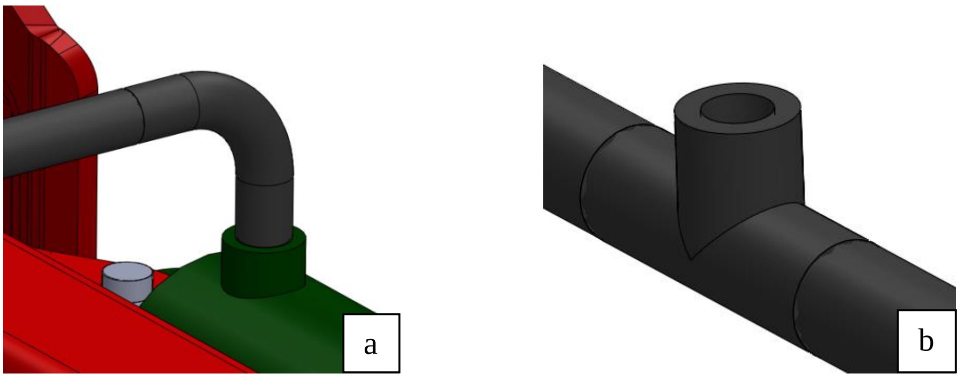


Figure 7.8 a: *90° bend elbow going from the cylinder, which is further connected to the 500 mm pipe.*

Figure 7.8 b: *Shows the T-joint at the end part of the pipeline at which hydraulic valve and accumulator are connected at their respective connections.*

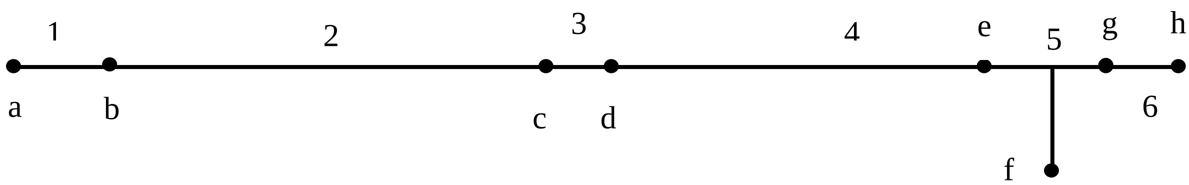


Figure 7.9: *The hydraulic network illustrated as nodes and elements. Element numbers are once more referring to Table 7.7, while the nodes (a-h) is referring to nodes listed in Table 7.8*

Table 7.7: *System elements with descriptions.*

Element	Description
1	90° elbow, D60 bend radius
2	500 mm hose pipeline

Table 7.7 continued.

3	90° elbow, D60 bend radius
4	2000 mm hose pipeline
5	T-joint
6	200 mm hose pipeline

Table 7.8: *Element inlets and outlets with the "nodal denotation" from figure 7.10.*

Node	Description
a	Cylinder outlet / 90° elbow inlet
b	90° elbow outlet / 500 mm hose inlet
c	500 mm hose oulet / 90° elbow inlet
d	90° elbow outlet / 2000 mm hose inlet
e	2000 mm hose outlet / T-joint inlet
f	T-joint outlet 1 / Accumulator inlet
g	T-joint outlet 2 / 200 mm hose inlet
h	200 mm hose outlet / Valve system inlet

To be able to do the further calculations, the pipeline between the hydraulic cylinder and the valve needs to be specified in terms of pipeline cross-sectional diameter. This design will also form the basis for the pipeline set up in the later functional test.

As a general rule obtained from (Kjølle 1995), the flow velocity, v_{oil}, should be around 4-5 m/s for pressures above 50 bar. On the basis of this information and making use of the maximum oil flow already obtained, cross sectional area and inner diameter of the pipeline can be calculated. Convert maximum flow by 120 l/min:

$$120\,\frac{L}{min} = 2 * 10^{-3}\ \frac{m^3}{s}$$

Calculate pipeline cross-sectional area:

$$A_{pipeline} = \frac{2 * 10^{-3}\ m^3/_s}{5\,\frac{m}{s}} = 4 * 10^{-4}\ m^2 = 400\ mm^2 \quad (Reworked\ from\ Eq\ 9)$$

Calculate pipeline cross-sectional diameter:

$$d_{pipeline} = \sqrt{\frac{4 * 400\ mm^2}{\pi}} = 22.57\ mm \quad (Eq\ 3)$$

, which is a rather big pipeline diameter.

Worth mentioning is that this system is supposed to operate between 100-160 bar, and 110-170 bar intended for the further testing. That is well over the rated pressure of 50 bar for this calculation, and for that reason a lower pipeline diameter will be allowed, consequently going down to an inner hose diameter of ½", which equate approximately 13 mm. Since no proper data on duct diameter in high pressure systems are found, this duct diameter will be used, both for the further calculations and during functional test.

Table 7.9: *Calculation basis for pipeline flow analysis. Oil data retrieved from (Shell 2013), for Shell Tellus S3 M 46 hydraulic oil, which is the selected oil type that will be use during the test.*

Properties	Symbol	Measure
Oil density (at 15 °C)	ρ	865 kg/m^3
Oil kinematic viscosity (at 40 °C)	ϑ	46 mm^2/s
Average oil flow	Q	112 L/min = 1.87*10^6 mm^3/s
Hose diameter	d	13 mm (1/2“)

What to obtain from calculations:

- Reynolds number to define the flow
- Velocity profile
- Pressure loss along pipeline
- Energy loss through pipeline (head loss)

In addition to pressure loss calculated along the pipeline, the pressure at the given nodes has been calculated relative to time throughout a release sequence. But since these more detailed calculations doesn’t give much more information about the pipeline capacity than the calculation of simply the pressure loss along the pipeline, it is regarded as secondary data, and can be found in appendix A2.

As an additional comparison to the hand calculations on pressure- and energy loss through pipeline, a CFD analysis has been performed later in this chapter.

Reynolds number

In order to verify whether we are dealing with a laminar or a turbulent flow, the Reynolds number must be calculated. The Reynolds number of the transition between laminar and turbulent flow are the following, according to (Kjølle 1995):

$$Laminar\ flow < Re \approx 2300 < Turbulent\ flow$$

Calculate Reynolds number, by equation obtained from (Kjølle 1995):

$$Re = \frac{Q * d}{\vartheta * A} = \frac{1.87 * 10^6 \frac{mm^3}{s} * 13\ mm}{46 \frac{mm^2}{s} * (\frac{\pi * (13\ mm)^2}{4})} = 3992 \quad (Eq\ 8)$$

$$\rightarrow Turbulent\ flow$$

Oil flow velocity and velocity profile

Cross sectional area of ½” hose:

$$A = \frac{\pi * (13\ mm)^2}{4} = 132.73\ mm^2 = 1.327 * 10^{-4}\ m^2 \quad (Reworked\ from\ Eq\ 3)$$

Calculate the oil flow velocity that is present at the pipeline inlet at the average rated oil flow:

$$v_{oil} = \frac{1.87 * 10^{-3}\, m^3/s}{1.327 * 10^{-4}\, m^2} = 14.13 \frac{m}{s} \quad (Eq\ 9)$$

Furthermore the velocity profile can be determined. For turbulent flows with $R_e < 150\,000$ the following equation can be used, obtained from (Kjølle 1995):

$$v = v_{max} \left(\frac{y}{r}\right)^{\frac{1}{7}} \quad (Eq\ 10)$$

, where the previous calculated oil flow now is denoted as the maximum flow velocity that is assumed to be present along pipe centerline.

At y = 0, which is along the inner diameter solid surface of the pipeline, the velocity is zero. However, it is important to note that it takes some distance from the entrance of the pipeline system before the velocity profile is fully developed. According to (Fox et al. 2010), a fully developed velocity profile can be expected within 25-40 times pipeline diameter from pipeline entrance, as a general rule.

By calculate approximate length of the entrance region, we get:

$$x_{entrance} = 40 * d = 40 * 13 = 520\ mm$$

By calculating the velocity for 1 mm intervals from pipeline inner solid surface towards the center of the flow line, we obtain a theoretical velocity profile:

$$v_1 = 14.13 \frac{m}{s} * \left(\frac{1\ mm}{6.5\ mm}\right)^{\frac{1}{7}} = 10.82\ \frac{m}{s} \quad (Eq\ 10)$$

$$v_2 = 14.13 \frac{m}{s} * \left(\frac{2\ mm}{6.5\ mm}\right)^{\frac{1}{7}} = 11.94\ \frac{m}{s}$$

$$v_3 = 14.13 \frac{m}{s} * \left(\frac{3\ mm}{6.5\ mm}\right)^{\frac{1}{7}} = 12.65\ \frac{m}{s}$$

$$v_4 = 14.13 \frac{m}{s} * \left(\frac{4\ mm}{6.5\ mm}\right)^{\frac{1}{7}} = 13.18\ \frac{m}{s}$$

$$v_5 = 14.13 \frac{m}{s} * \left(\frac{5\ mm}{6.5\ mm}\right)^{\frac{1}{7}} = 13.61\ \frac{m}{s}$$

$$v_6 = 14.13 \frac{m}{s} * \left(\frac{6\ mm}{6.5\ mm}\right)^{\frac{1}{7}} = 13.96\ \frac{m}{s}$$

, whereas:

$$v_7 = v_{max} = 14.13 \frac{m}{s}$$

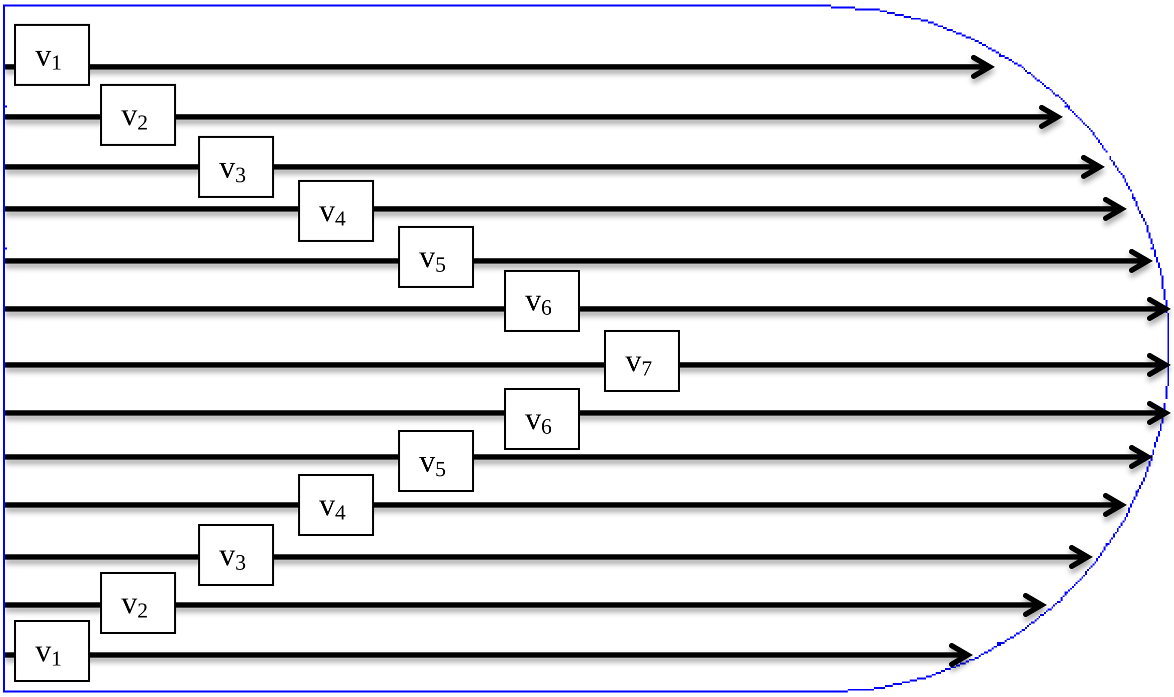


Figure 7.10: *Illustration of the calculated velocity profile of the fully developed turbulent flow in the pipeline.*

Pressure drop along pipeline

To keep track of expected pressure loss in the system, the pressure drop along the pipeline is calculated. To get the total system loss, losses in valve and cylinder must be included. However, for this calculation it is only with regards to the pipeline system between cylinder and valve that pressure loss is calculated. Calculation is done for the 6 pipeline elements isolated, where total loss is summarized for the entire pipeline system in the end.

The calculation is done with regards to the Bernoulli equation, obtained from (Brautaset 1983), taking friction effects within the fluid into account:

$$P_{tot} = p + \lambda * \frac{l}{d} * \rho(rho) * \frac{v_m^2}{2} \quad (Eq\ 11)$$

, where:

$$\mathrm{p} = \text{static pressure}$$

, which is the real measured system pressure at each point of the system, and that relates to the potential energy in the system.

Furthermore:

$$\lambda * \frac{l}{d} * \rho(rho) * \frac{v_m^2}{2} = dynamic\ pressure$$

, which is relative to fluid velocity and kinetic energy in system.

Finally:

$$p_{tot} = total\ pressure$$

, including static and dynamic pressure, where the sum is constant along the whole pipeline.

It is important to note that the dynamic pressure is not a measurable quantity like the static pressure, whereas the static pressure decrease represents the real system pressure loss. The pressure loss is caused by an energy transition from mechanical to thermal energy in the system in terms of internal friction effects in the fluid.

Before doing the calculation there is made some assumptions and approximations about the system:

- assume a constant oil flow for the simplicity of this calculation, where the oil flow is the rated average of the oil flow recently calculated to 112 L/min, which is equal to $1.87*10^{-3}$ m^3/s.
- Possible disturbance from valves and accumulator is not taken into account
- Assume smooth solid surface inside pipeline and subsequently frictionless interaction between solid and fluid.
- Assume fully developed turbulent flow along the entire pipeline with the respective internal fluid friction taken into account for the specified flow.

<u>Element 1: 90° elbow, D60 bend radius</u>

The calculation of pressure loss through bends is a bit more complex than for simple straight pipelines. In addition to the pure pressure loss due to friction like for a straight line, a second pressure loss due to resistance from the sudden change in directory must be included.

To calculate the pressure drop along a straight pipe the static-/dynamic pressure relation including friction effects from the Bernoulli equation, obtained from (Brautaset 1983) is used:

$$\Delta p_{loss} = \lambda * \frac{l}{d} * \rho(rho) * \frac{v_m^2}{2} \quad (Reworked\ from\ Eq\ 11)$$

Note that the right hand of the equation should have a negative sign, but turn it to positive, since the static pressure is denoted with pressure “loss”.

By regarding the inner surface of the tube as smooth, and taking the turbulent flow into account, the friction coefficient can be calculated by the following formula (Brautaset 1983):

$$\lambda = \frac{0.316}{R_e^{0.25}} = \frac{0.316}{3992^{0.25}} = 0.0398 \quad (Eq\ 13)$$

Calculate the length of the bended pipe under the assumption of a bend radius equal to 60 mm:

$$l = \frac{\pi * d}{4} = \frac{\pi * 60\ mm}{4} = 47.125\ mm$$

Calculate pressure loss from pipe length:

$$\Delta p_{loss,1a} = 0.0398 * \frac{0.047125\ m}{0.013\ m} * \left(865\frac{kg}{m^3}\right) * \frac{\left(14.13\frac{m}{s}\right)^2}{2} \quad (Reworked\ from\ Eq\ 11)$$

$$\Delta p_{loss,1a} = 12\ 458\ Pa = 0.0125\ MPa = 0.13\ bar$$

Calculate pressure loss due to bend (Brautaset 1983):

$$\Delta p_{loss,1b} = \delta\rho(rho)\frac{v_m^2}{2} \quad (Eq\ 14)$$

, where coefficient δ = 1.2 applies for 90° bends according to (Brautaset 1983).

$$\Delta p_{loss,1b} = 1.2 * \left(865\frac{kg}{m^3}\right) * \frac{\left(14.13\frac{m}{s}\right)^2}{2} \quad (Eq\ 14)$$

$$\Delta p_{loss,1b} = 103\ 622\ Pa = 0.104\ MPa = 1.04\ bar$$

$$\Delta p_{loss,1} = 0.13\ bar + 1.04\ bar = 1.17\ bar$$

Part 2: 500 mm hose pipeline

$$\Delta p_{loss,2} = 0.0398 * \frac{0.5\ m}{0.013\ m} * \left(865\frac{kg}{m^3}\right) * \frac{\left(14.13\frac{m}{s}\right)^2}{2} \quad (Reworked\ from\ Eq\ 11)$$

$$\Delta p_{loss,2} = 132\ 184\ Pa = 0.132\ MPa = 1.32\ bar$$

Part 3: 90° elbow, D60 bend radius

$$\Delta p_{loss,3} = \Delta p_{loss,1} = 1.17\ bar$$

Part 4: Pipe 2000 mm hose pipeline

Calculate the pressure loss along the 2000 mm pipeline:

$$\Delta p_{loss,4} = 0.0398 * \frac{2\ m}{0.013\ m} * \left(865\frac{kg}{m^3}\right) * \frac{\left(14.13\frac{m}{s}\right)^2}{2} \quad (Reworked\ from\ Eq\ 11)$$

$$\Delta p_{loss,4} = 528\ 737\ Pa = 0.53\ MPa = 5.3\ bar$$

Part 5: T-joint

The T-joint is mounted onto the accumulator and the valve system. As long as the valve system is able to drain the sufficient amount of oil, there will be no oil flow into the accumulator. In that case the T-joint can be considered as a straight tube leading oil in only one direction, while neglecting the shear resistance against neighbouring oil filling the pipe outlet towards the accumulator. In the case where there is flow into the accumulator, we need to calculate with the situation where flow is separated into the two ducts. This might just as likely be the case in the start of the release sequence, before the valve system is fully open.

Calculate for case 1; flow only towards valve system:

$$\Delta p_{loss,5-case1} = 0.0398 * \frac{0.05\ m}{0.013\ m} * \left(865\frac{kg}{m^3}\right) * \frac{\left(14.13\frac{m}{s}\right)^2}{2} \quad (Reworked\ from\ Eq\ 11)$$

$$\Delta p_{loss,5-case1} = 13\ 218\ Pa = 0.013\ MPa = 0.13\ bar$$

Calculate for case 2; flow into both ducts:

$$v = \frac{\left(14.13\frac{m}{s}\right)}{2} = 7.065\frac{m}{s}$$

, where the recalculated flow velocity is regarded for the entire T-joint. This also includes the entrance region before the flow separation into two ducts.

$$\Delta p_{loss,5a-case2} = 0.0398 * \frac{0.05\ m}{0.013\ m} * \left(865\frac{kg}{m^3}\right) * \frac{\left(7.065\frac{m}{s}\right)^2}{2} \quad (Reworked\ from\ Eq\ 11)$$

$$\Delta p_{loss,5a-case2} = 3305\ Pa = 0.033\ bar$$

Flow towards accumulator changes direction of 90°, so do the same calculation as for a 90° elbow:

$$\Delta p_{loss,5b-case2} = 1.2 * \left(865\frac{kg}{m^3}\right) * \frac{(7.065\ m/s)^2}{2} \quad (Eq\ 14)$$

$$\Delta p_{loss,5b-case2} = 25\ 906\ Pa = 0.026\ MPa = 0.26\ bar$$

$$\Delta p_{loss,5-case2} = 0.26\ bar + 0.033\ bar = 0.29\ bar$$

Part 6: 200 mm hose pipeline

$$\Delta p_{loss,6} = 0.0398 * \frac{0.2\ m}{0.013\ m} * \left(865\frac{kg}{m^3}\right) * \frac{\left(14.13\frac{m}{s}\right)^2}{2} \quad (Reworked\ from\ Eq\ 11)$$

$$\Delta p_{loss,6} = 52\ 874\ Pa = 0.053\ MPa = 0.53\ bar$$

Calculate total pressure loss along the pipeline, assuming flow to pass straight through T-joint (element 5):

$$\Delta p_{loss,tot} = \Delta p_{loss1-6} = 9.99\ bar$$

Calculation of energy loss

What is called "head loss" describes the loss in momentum of the fluid through the pipeline. The total head loss can be separated into major- and minor losses. The major losses are a result of frictional effects, while minor losses come from entrances, fittings, and area changes. The pipeline system in this case has a constant cross-sectional area, and though

there will be some additional friction from entrances and fittings, the head loss of the pipeline itself, will be calculated only with regards to the major losses.
Calculating head loss from the total pressure loss, by equation obtained from (Kjølle 1995):

$$h_l = \frac{\Delta p}{\rho} \quad (Eq\ 15)$$

$$h_l = \frac{990\,000\ Pa}{865\,\frac{kg}{m^3}} = 1144.51\,\frac{m^2}{s^2} \quad (Eq\ 15)$$

Calculate total energy loss through pipeline or more precisely; energy transition from mechanical to thermal energy, by equation obtained from (Kjølle 1995):

$$E_{loss} = h_l * m \quad (Eq\ 16)$$

Since the system mass has already been calculated to 0,405 kg, we can calculate energy loss directly:

$$E_{loss} = 1144.51\,\frac{m^2}{s^2} * 0.405\ kg = 464\ J \quad (Eq\ 16)$$

By taking the specific heat capacity of the oil into account, the temperature rise in the oil can be estimated. Since no detailed numbers on the specific heat of the hydraulic oil can be obtained, the heat capacity for petroleum from (Johannessen 2002) is used, which is assumed to be close enough to the properties of the hydraulic oil in this calculation. Furthermore, the number is obtained at 1 atmosphere of pressure and 293 Kelvin of temperature.

$$c_{oil} = \frac{2100\ J}{kg * K}$$

By taking the density from the chosen hydraulic oil, we get the approximate heat capacity of the oil, by rewriting equation obtained from (Tipler et al 2008):

$$C_{oil} = m * c_{oil} \quad (Eq\ 17)$$

Calculate system mass:

$$m = \rho_{oil} * V_{pipeline}$$

$$m = 865\,\frac{kg}{m^3} * 0.000468\ m^3 = 0.405\ kg$$

$$C_{oil} = 0.405\ kg * \frac{2100\ J}{kg * K} = 850\ J/K$$

Calculate temperature increase:

$$\Delta T = \frac{E_{loss}}{C_{oil}} = \frac{464\ J}{850\,\frac{J}{K}} = 0.55\ K \quad (Eq\ 18)$$

FEA-/CFD analysis – Flow Simulation

A flow analysis using computational fluid dynamics have been completed using the Solid Works Flow Simulation software. These results are used as a comparison against the manual calculations just shown, with regards to pressure drop and the mechanical-to-thermal energy transition and consequent temperature increase.

Fluid input data

Table 7.10: *Overview of the fluid input with regards to the environmental conditions specified for the pipeline.*

Parameter	Value
Thermodynamic system pressure	$1.7*10^7$ Pa (170 bar)
Thermodynamic temperature setting	313 K
Turbulence intensity	0.057 %
Fluid flow cross sectional diameter (Turbulence length)	0.013 m (13 mm)
Turbulence energy	0.965 J/kg
Turbulence dissipation	91.36 W/kg
Type of flow	Turbulent only

Calculation of fluid input

Equations for calculating the following input is obtained from (CFD-Online).
Turbulence intensity with regards to the previously calculated Reynolds number:

$$I_t = 0.16 * Re^{-(1/8)} = 0.16 * 3992^{-(1/8)} = 0.057\ (\%) \quad (Eq\ 19)$$

Calculate the turbulence energy, which is the kinetic energy of the system inlet flow combined with the turbulence intensity:

$$k = \frac{3}{2} * (v_{oil} * I_t)^2 = \frac{3}{2} * \left(14.13\frac{m}{s} * 0.057\right)^2 = 0.965\frac{J}{kg} \quad (Eq\ 20)$$

Calculate turbulence dissipation by the following equation, obtained from (cfdesign.com):

$$\epsilon = \frac{c_\mu^{(3/4)} * k^{(3/2)}}{0.07 * l} \quad (Eq\ 21)$$

,where c_μ = 0,09 (coefficient, empirical value) and l = turbulence length.

$$\epsilon = \frac{0.09^{(3/4)} * \left(0.965\frac{J}{kg}\right)^{(3/2)}}{0.07 * 0{,}013\ m} = 91.36\frac{W}{kg} \quad (Eq\ 21)$$

Boundary conditions

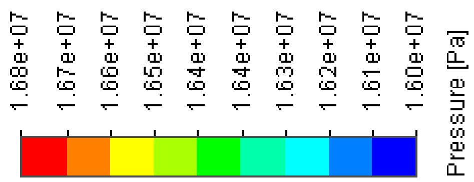
There are three system boundaries; at inlet (node a), at main outlet towards valve system (node f) and at secondary outlet towards accumulator.

__Table 7.11:__ System boundary conditions.

Boundary condition	Value
Inlet flow (at node a)	$1.87*10^{-3}$ m^3/s (112 L/min)
Static pressure (at node f)	$1.6*10^{7}$ (160 bar)
Static pressure (at accumulator inlet)	$1.6*10^{7}$ (160 bar)

Note that the static pressure at both accumulator inlet and at node f is set at 160 bar. While the hydrostatic pressure in the system in its initial state is 170 bar, the calculated pressure drop from the hand calculation is used to estimate the pressure conditions at the end of the pipeline, where boundary conditions regarding pressure is set. The pressure at the end of the pipeline is therefore estimated to 160 bar under the assumption of constant oil flow and fully developed turbulent flow along pipeline.

Pressure



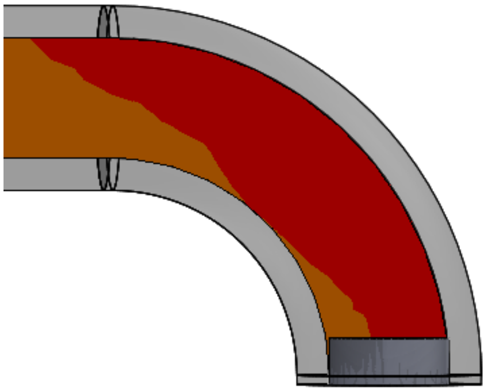


***Figure 7.11 (left):** The pressure scale shown in Pascal with 10^5 Pa intervals (equal to 1 bar).*

***Figure 7.12 (right):** Illustration of how the oil is being pressurized against the outer wall of the 90° elbow.*

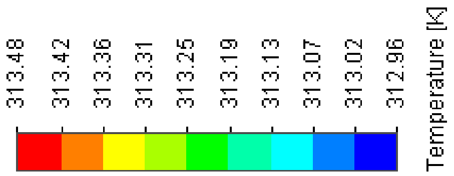
***Figure 7.13:** As seen from this illustration the results output indicates a pressure deviation or rather pressure loss through the pipeline of 8 bar. The (red) 168 bar pressure indication at 90° elbow at cylinder outlet, shows the pressure in the beginning of the pipeline, while the (blue) 160 bar pressure indication at valve connection shows the pressure at the end of the pipeline.*

The pressure loss of 8 bar is close to the manual calculation, which gave a pressure loss of 10 bar. However, it is not spot-on-target. One of the reasons for the smaller deviation between the hand calculation and the computer analysis might be caused by the fact that the hand calculations are carried out under the assumption of a constant and uniform oil flow velocity without accounting for local changes to the velocity along the direction of the pipeline caused by flow acceleration out of elbows, for instance.

Temperature

Figure 7.14: *As seen from this overview of the pipeline, the temperature is rising along the pipeline in accordance with the pressure loss. The temperature at the inlet of the pipeline is 312.96 K more or less in accordance with the 313 K system input. The temperature at the end is 313.48 K.*

Figure 7.15: *The temperature scale shown in Kelvin.*



The temperature difference throughout the pipeline from the CFD analysis is as follows:

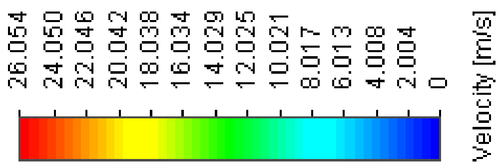
$$\Delta T = 313.48 - 312.96 = 0.52\ K$$

Compared to the hand calculation, which gave ΔT = 0.55 K, the CFD analysis shows a good consistency against the hand calculation, with a difference against hand calculations by 0.03 K.

Fluid velocity and flow trajectories

When studying the flow trajectory the already mentioned flow accelerations out of the elbows can be easily spotted. Just from the fact that these accelerations are present, there is on the other hand difficult to catch sight of any fully developed velocity profile similar to the expected velocity profile outlined in figure 7.10. But assuming no friction between pipe wall surface and fluid we should not expect any developed velocity profile in the first place.

Figure 7.16: *Velocity scale from results output.*



The flow throughout the pipeline is relatively consistent, but with local variations, especially along the two 90° elbows and the T-split.

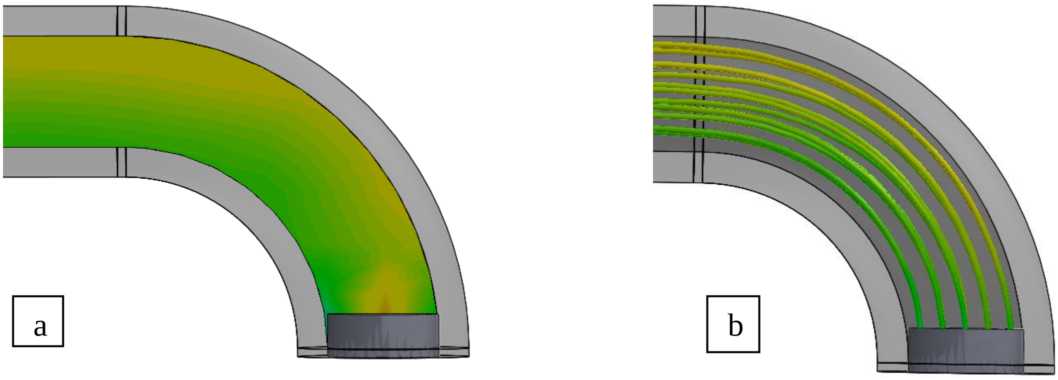


Figure 7.17 a, b: *As expected the maximum velocity around the 90° elbow is in the outer radius as illustrated by both the velocity- (a) and flow trajectory (b) profile.*

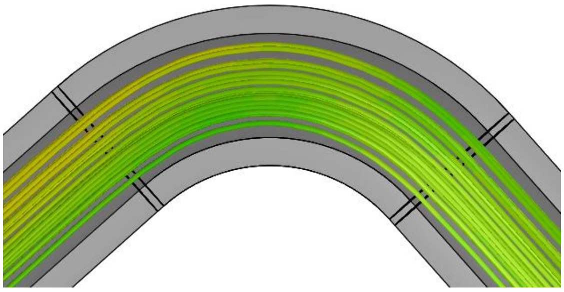


Figure 7.18: *Illustration of the flow through the 2nd 90° elbow which illustrates how the oil is accelerated along the outer radius on the way out of the elbow.*

However, what is surprising when studying the flow trajectories is that turbulent flow doesn't seem to be present. From previous calculation of Reynolds number we would expect some turbulent flow to occur. On the other hand the calculated Reynolds number is still not more above the transition to laminar flow that laminar flow could still be present in this case.

7.3 Mechanical design

With regards to the mechanical design most components from the existing Kverneland Auto Reset system can be used. However, a design analysis of the tie plate that serves as the new connection between the tie rod and the cylinder needs to be performed. Tie plate is once more result of work performed during this project. The cylinder chosen is within the range of cylinders found at Kverneland Klepp today, with the best possible fit to cylinder specification with regards to calculated piston diameter and stroke length. Further detail design at later point will be performed by a hydraulic cylinder manufacturer/supplier.

7.3.1 Tie plate

The tie plate will be designed according to Eurocode 3 as a bolted connection. To verify that the tie plate can withstand the maximum static forces, both hand calculations and a FEM-analysis will be carried out. The FEM analysis is performed as a verification of hand calculations. Additionally, it will be used for possible design optimization in order to pursue the possibilities of reducing material.

Tie plate design

The design calculations starts off by identifying the static loads and their direction. According to the release characteristics, the greatest resistance in the system is at the time upon release.

Therefore initial values regarding the axial forces on tie rod and cylinder which acts on the tie plate are retained for use in this calculation.

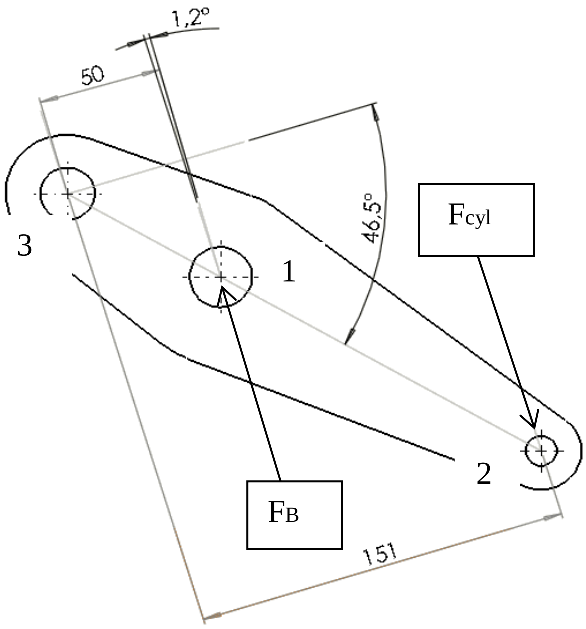


Figure 7.19: *The applied force from the cylinder and the counterforce, F_B along tie rod center axis both acting on the tie plate and its respective bolted connections 1 and 2.*

The following calculations performed on tie plate is based on NS-EN 1993-1-8:2005+NA:2009 and Eurocode 3, obtained from (Terjesen 2012).

The nominal diameter of hole 1 is predefined from the hole diameter on the tie rod, obtained from (Kverneland Group 2004):

$$d_0 = 25.25\,mm$$

The nominal diameter of hole 2 is predefined from the chosen hole-diameter on the cylinder rod end:

$$d_0 = 15.25\,mm$$

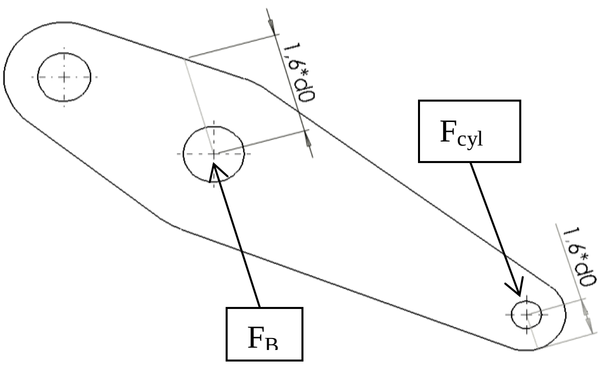


Figure 7.20: *As seen from this illustration, the critical part on the plate is naturally the length by which there exists material outside the hole in a parallel line to the force. From the instructions in Eurocode 3 the qualitative lengths on both holes are illustrated.*

Calculate the required material around hole 1, where F_B is acting:

$$1.6 * d_0 = 1.6 * 25.25\,mm = 40.4\,mm$$

Calculate the required material around hole 2, where F_{cyl} is acting:

$$1.6 * d_0 = 1.6 * 15.25\,mm = 24.4\,mm$$

Calculate the required material thickness:

$$t = 0.7\sqrt{\frac{F_{Ed} * \gamma_{M0}}{f_y}} \quad (Eq\ 22)$$

Calculate the design force, F_{Ed} which is equal to F_B:

$$F_{Ed} = F_B = \frac{F_A}{Moment\ arm\ relation\ 1} = \frac{16\,kN}{0.1} = 160\,000N$$

The material factor is the following according to Eurocode 3 (Terjesen 2012):

$$\gamma_{M0} = 1.05$$

Assume S355J0 structural steel:

$$f_y = 355\,MPa$$

Calculate the final thickness:

$$t_{tot} = 0.7\sqrt{\frac{160\,000\,N * 1.05}{355\,MPa}} = 15.22\,mm \quad (Eq\ 22)$$

Given the fact that the load is divided between two plates the thickness for each plate is the following:

$$t = \frac{15.22\,mm}{2} = 7.6\,mm$$

Choose t = 10 mm (x 2).
Do a control on the material thickness versus the hole diameters, according to the requirement of the given standard in Eurocode 3:

$$d_0 = 25.25\,mm \leq 2.5 * t = 2.5 * 20\,mm = 50\,mm$$

$$\rightarrow OK$$

Stress verifications

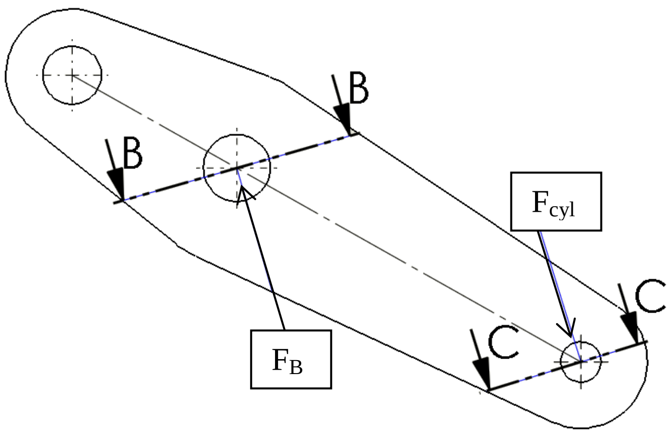
Further calculations are also done according to Eurocode 3.

<u>Tensile stresses</u>

The second test against material stress is done with regards to tensile stress caused by F_B and F_{cyl} at their respective perpendicular cross-sections.

Table 7.12: *Applied forces and minimum safety factor applying for static structural analysis of tie-plate.*

Boundary condition	Value
F_B	160 kN
F_C	53.33 kN
Material yield limit	355 MPa
n (safety factor minimum requirement)	1.5



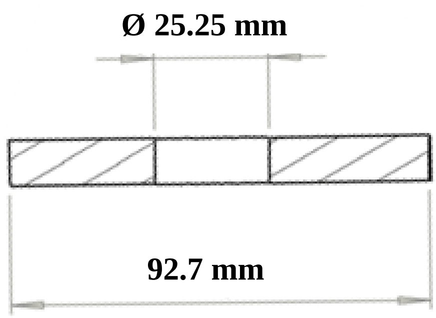
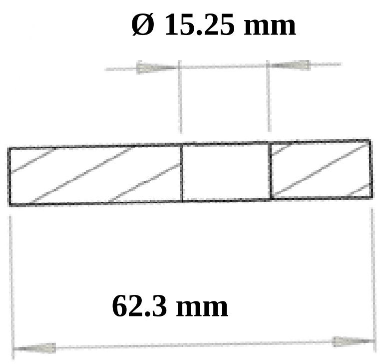


Figure 7.21 (left): *Cross section B-B is located in the perpendicular section of F_B while cross-section C-C is located perpendicular to F_{cyl}.*

Figure 7.22 (right): *Cross-section B-B.*

Figure 7.23: *Cross-section C-C.*



Design criteria:

$$N_{t,Rd} \geq F_{Ed}$$

Calculate the resistive force in the material at cross-section B-B:

$$N_{t,Rd} = \frac{0.9 * A_{net} * f_u}{\gamma_{M2}} \qquad (Eq\ 23)$$

$$N_{t,Rd} = \frac{0.9 * \left((92.7\ mm - 25.25\ mm) * 20\ mm\right) * 510\ MPa}{1.25} \qquad (Eq\ 23)$$

$$N_{t,Rd} = 495\ 352\ N$$

Control towards the design criteria:

$$495\ 352\ N \geq F_{Ed} = F_B = 160\ 000\ N \ \rightarrow OK$$

Design criteria:

$$N_{t,Rd} \geq F_{Ed}$$

Calculate the resistive force in the material at cross-section C-C:

$$N_{t,Rd} = \frac{0.9 * \big((62.3\ mm - 15.25\ mm) * 20\ mm\big) * 510\ MPa}{1.25} \qquad (Eq\ 23)$$

$$N_{t,Rd} = 345\ 535\ N$$

Control towards the design criteria:

$$345\ 535\ N \geq F_{Ed} = F_{cyl} = 53\ 300\ N \ \rightarrow OK$$

<u>Bending- moment and stress</u>

Furthermore, bending stress shall be verified. The tie plate can be regarded as a cantilever beam, where F_{cyl} is the force at the very end, and F_B can be the regarded as the support force that balance the force from the cylinder upon release, with no translational- nor rotational degrees of freedom at hole 1, referring to Figure 7.19. Hence no force at the pivot point is implemented into the calculation.

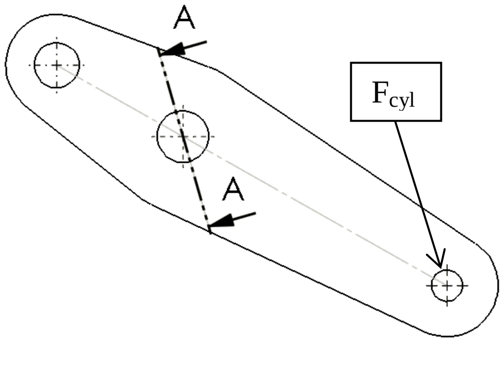


Figure 7.24: *To verify that the plate can withstand the forces with regards to bending stress a control at the cross-section A-A is done, where the maximum bending moment is located.*

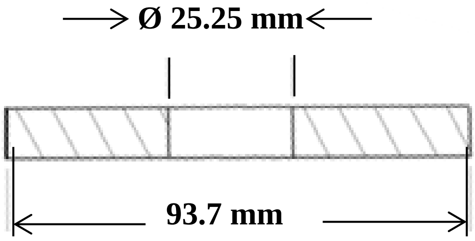


Figure 7.25: *Cross-sectional view A-A with its respective dimensions in order to calculate the section modulus.*

Calculate the bending stresses according to Eurocode 3 as referred to earlier.
Design criteria:

$$M_{el,Rd} \geq M_{Ed}$$

Calculate material resistance against moment:

$$M_{el,Rd} = \frac{W_{el} * f_y}{\gamma_{M0}} \quad (Eq\ 24)$$

(Calculate the section modulus):

$$W_{el,A-A} = \frac{(2 * 10\ mm) * ((93.7\ mm)^2 - (25.25\ mm)^2)}{6}\ (Eq\ 26)$$

$$W_{el,A-A} = 27\ 140\ mm^3$$

$$M_{el,Rd} = \frac{27\ 140\ mm^3 * 355\ MPa}{1.05} = 9\ 176\ 049\ Nmm \quad (Eq\ 24)$$

Do a control against the design criteria:

$$9\ 176\ 049\ Nmm \geq M_{Ed} = 53\ 330\ N * (151\ mm - 50\ mm) = 5\ 386\ 330\ Nmm$$

$$\rightarrow OK$$

<u>Equivalent (von Mises) stress</u>

In order to get an idea of the equivalent stress (denoted uniform stress in calculation), tensile and shear stress along cross-section A-A from Figure 7.26-27 is obtained. The reason for choosing the given cross-section, is that we find the greatest bending stress in this particular section, with an expected subsequent high uniform stress. The calculation is performed by first calculating the bending stress to obtain normal tensile- and shear stress. Furthermore parallel shear stress is calculated, before finally obtaining the uniform stress. The uniform stress is compared against the yield limit of the material, where safety against yielding is found. In this calculation Eurocode 3 is <u>not</u> applied.

Calculate bending stress at cross section A-A:

$$\sigma_{b,A-A} = \frac{M_{b,A-A}}{W_{b,A-A}} \quad (Eq\ 25)$$

$$M_{b,A-A} = F_{cyl} * (z6 - z5) \quad (Eq\ 26)$$

$$M_{b,A-A} = 53\ 330\ N * (151\ mm - 50\ mm) = 5\ 386\ 330\ Nmm$$

$$W_{b,A-A} = \frac{b * h^2}{6}\ (=\ W_{el,A-A}) \quad (Eq\ 27)$$

$$W_b = 27\ 140\ mm^3$$

$$\sigma_{b,A-A} = \frac{5\,386\,330\ Nmm}{27\,140\ mm^3} = 198.46\ MPa \quad (Eq\ 25)$$

Calculate perpendicular stress and shear:

$$\sigma_{b_\perp} = \tau_{b_\perp} = \frac{\sigma_b}{\sqrt{2}} = \frac{198.46\ MPa}{\sqrt{2}} = 140.33\ MPa \quad (Eq\ 28)$$

Calculate parallel shear stress:

$$\tau_{parallel} = \frac{F_{cyl}}{A_{A-A}} = \frac{53\,330\ N}{(93.7\ mm - 25.25\ mm) * 20\ mm} = 38.96\ MPa \quad (Eq\ 29)$$

Calculate the uniform stress:

$$\sigma_{uniform} = \sqrt{\sigma_{b_\perp}^2 + 3 * \left(\tau_{b_\perp}^2 + \tau_{parallel}^2\right)} \quad (Eq\ 30)$$

$$\sigma_{uniform} = \sqrt{140.33^2 + 3 * (140.33^2 + 38.96^2)} = 288.66\ MPa \quad (Eq\ 30)$$

Control the uniform stress against yield stress:

$$n = \frac{355\ MPa}{288.66\ MPa} = 1.23$$

$$\rightarrow Heat\ treatment\ required$$

With regards to safety factor by 1.5 the design fails the requirement. However by heat treatment the yield limit should be possible to increase by a corresponding safety factor well above 1.5.

FEM (Finite Element Method)-analysis

The FEM-analysis is carried out using the Ansys Workbench software.

Preparing the model

In the preparation of the model some approximations are made on how to contemplate the physical reality and the way external forces from cylinder and tie rod are acting on the plate. In reality hole 3 from Figure 7.19 is fixed in all three translational degrees of freedom in addition to two of its rotational degrees of freedom. That leaves only one rotational degree of freedom along its center axis, as shown by cylinder support in Figure 7.26. Hole 1 and 2 (illustrated in Figure 7.19) are free in two out of three translational degrees of freedom, where the whole plate is fixed in the third translational degree of freedom as shown in figure 7.27.

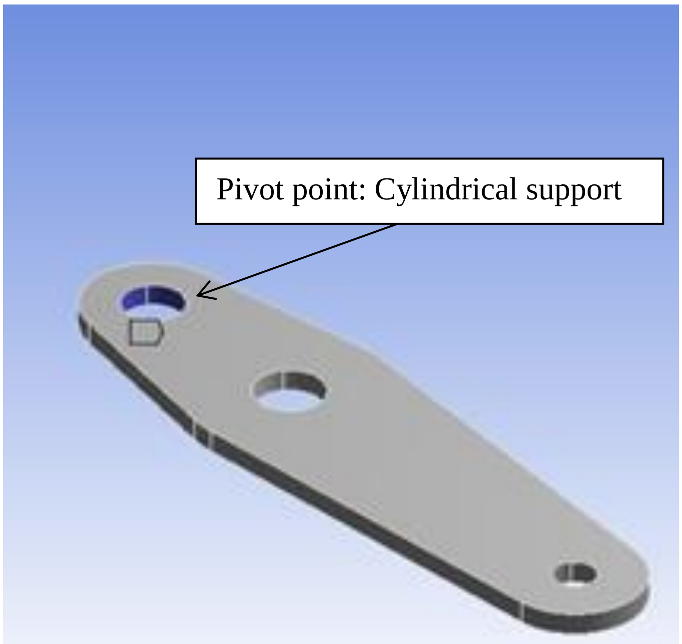


Figure 7.26 (left): *The pivot point is fixed in two out of three translational degrees of freedom and two out of three rotational degrees of freedom by the cylindrical support.*

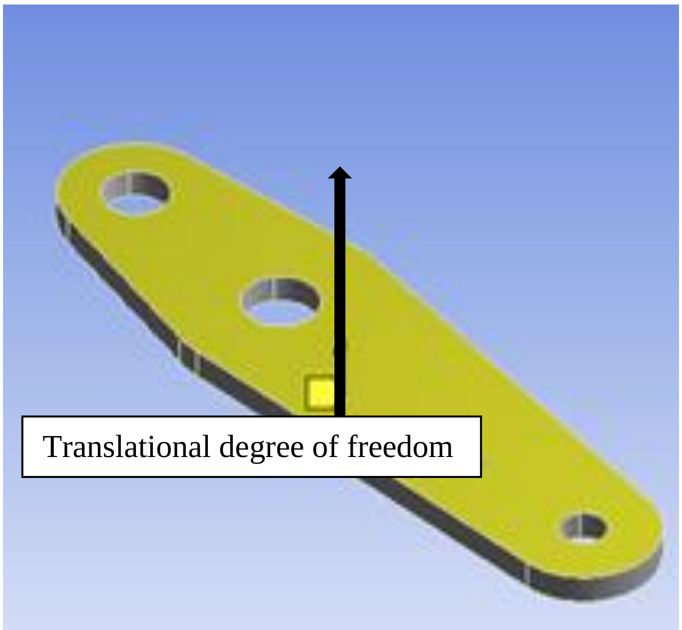
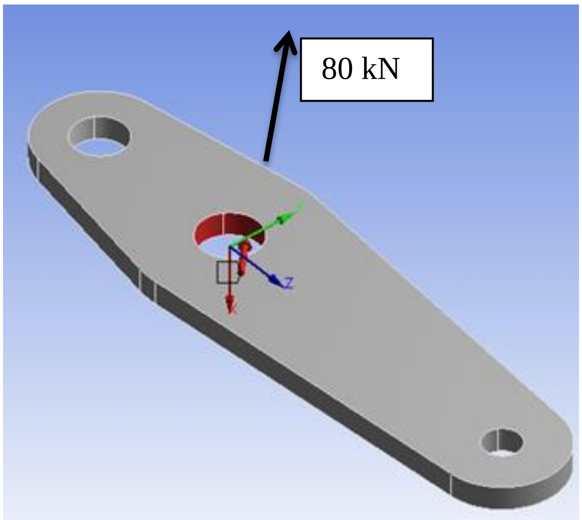


Figure 7.27 (right): *The whole plate is fixed in the third translational degree of freedom along the direction of the black arrow.*

The forces acting along the tie rod- and cylinder center axis and onto the plate are applied in the model equal to the illustrations in Figure 7.28 and 7.29. Since the FEM analysis is carried out with only one plate instead of both the similar plates, the applied forces are naturally applied by one half of the total force acting on the two plates together.



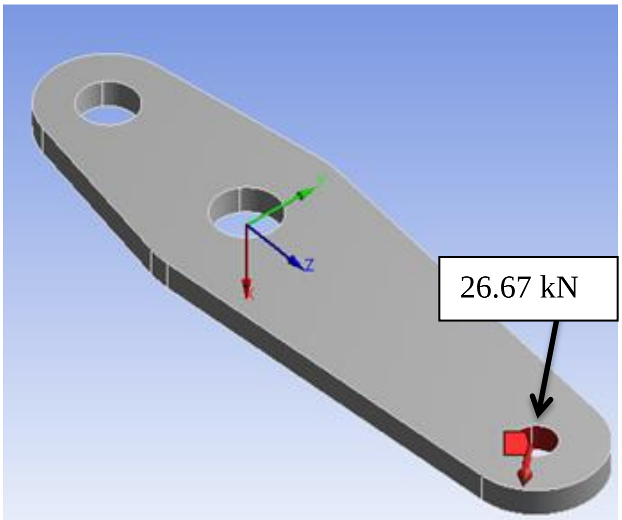


Figure 7.28 (left): *The applied force on tie rod fix is 80 kN, considering the total force of 160 kN distributed on two plates.*

Figure 7.29 (right): *The applied force on cylinder fix is 26.67 kN, considering the total force of 53.33 kN distributed onto the two plates.*

Exploiting symmetry

When looking at model geometry and displacement boundary constraints, it might look like we are dealing with a symmetrical case. However, when looking at the applied forces from Figure 7.30-31, we see that the load case is not symmetrical and the model has to be treated that way, and it can be concluded that symmetry cannot be exploited.

Mesh

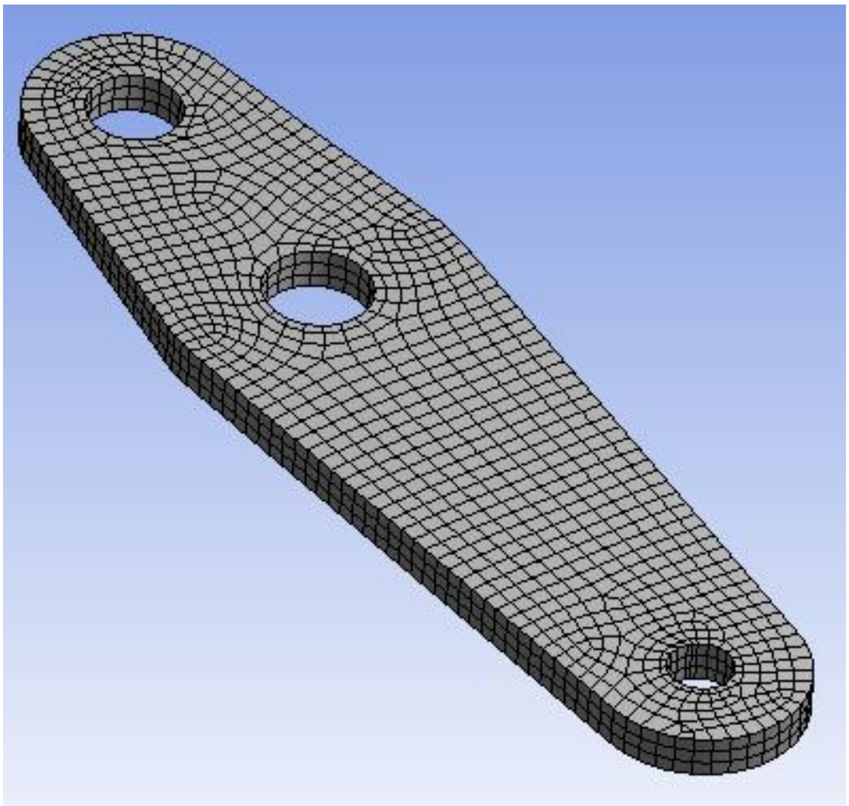


Figure 7.30: *The mesh is attributed with hexahedron elements, where number of elements is 2613 with a minimum element edge length of 8.87 mm, obtained from the Ansys software.*

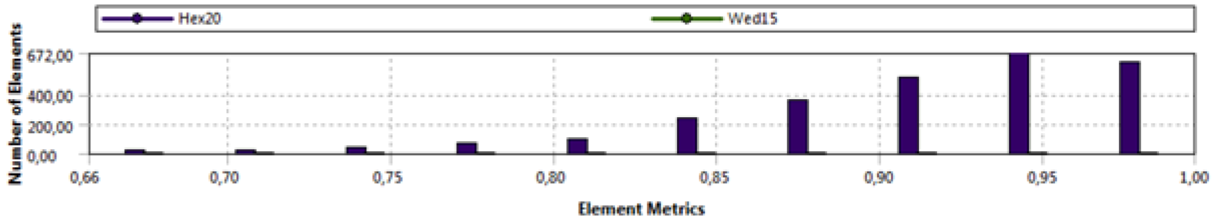


Figure 7.31: *The element quality is good with the chosen refinement, according to the Ansys software. As shown, well over half the elements are in the upper 10% in element quality. The only elements with a bit lower quality are situated at the edges around the holes. So the stress result around the holes should be regarded with precaution.*

Results

When reading out the results, attention should be paid to potential physical singularities, with special regards to sharp corners. There are no corners in the model; the only possible location for such a singularity would be at the ends of the outside radiuses in the model, caused by possible non-tangency between edges. For that reason the model has been examined and there is a consistent tangency at the edges of all the outer radiuses on the model which no sharp corners at radiuses should be present in the model, causing physical singularities.

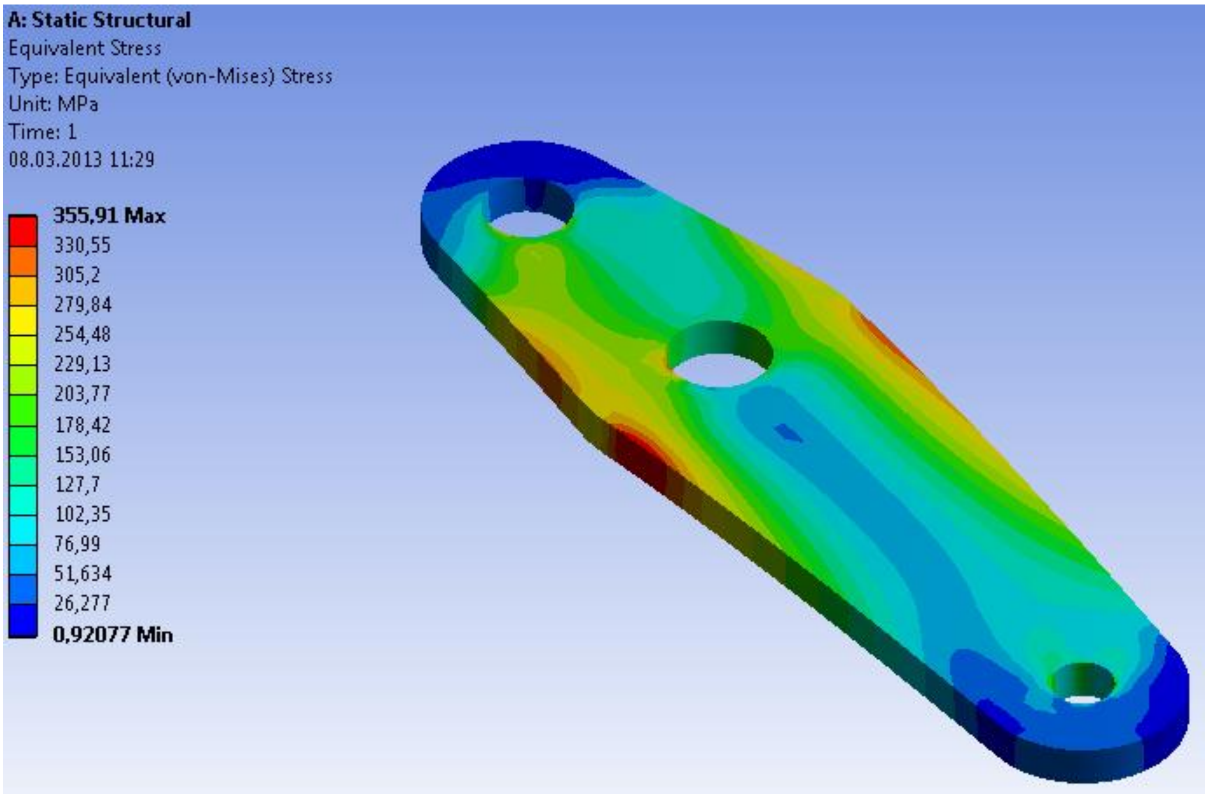


Figure 7.32: *The uniform stress (von Mises stress) distribution in the plate shows that for most of the material, the stress is below the yield limit of the chosen S355 steel. However, there is a small area in the outer volume around the tie rod fix where the analysis index compressional stress just above the yield limit of the untreated material.*

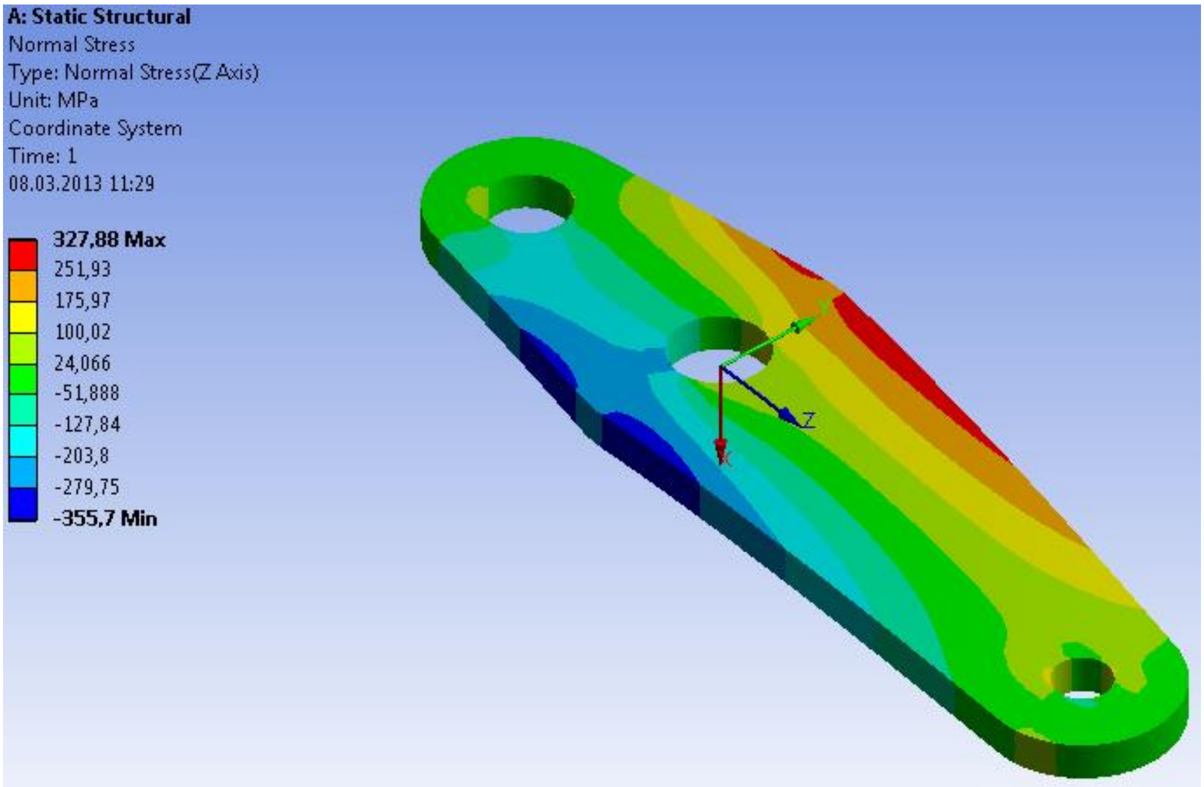


Figure 7.33: *From this illustration of the normal stress along the symmetry axis (z- axis of displayed coordinate system) we obtain the distribution of tensile and compressional stress in the model. Negative prefix denotes compressional stress and positive prefix denotes tensile stress. As expected tensile stress is located at the "front" of the plate towards plough beam accordingly to applied forces once more illustrated in Figure 7.28 and 7.29.*

<u>Recommendations</u>

From both the hand calculation of the uniform stress along the chosen cross-section in Figure 7.24 and the FEM stress analysis with regards to equivalent (or uniform) stress, it is recommended that the tie plate is heat treated to increase the yield limit of the material. The hand calculation shows that for the chosen cross section there is no yielding, although there is a disturbingly low safety factor against yielding. In a very small part of the outer volume of the part, material is actually plastically deformed, according to the FEM analysis. Another question is whether it would be possible to change the geometry of the part. However, it is not desirable to increase the width or the thickness of the plate due to issues regarding free space towards neighbouring parts, once more leaving heat treatment as the obvious choice for dealing with the material stress present.

7.4 Function test model

The function test model consists of the mechanical-, hydro-pneumatic- and electronic system. The full test model will be more closely described in the *"test preparations"* section in chapter 8. But as a summarization for the product design part of the project, the final design will be outlined in the following section along with some rendered images.

7.4.1 Hydro-pneumatic system

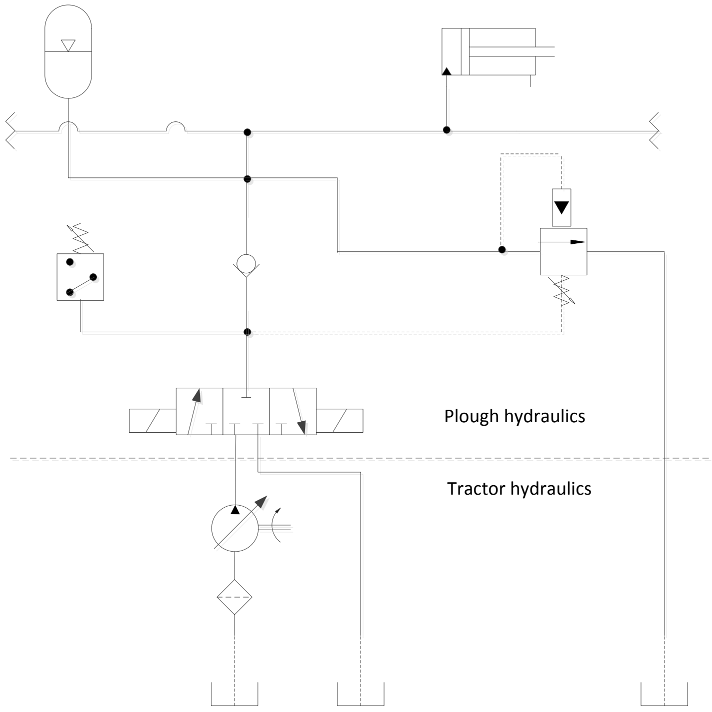


Figure 7.34: *Schematics of the hydro-pneumatic system of the prototype with cartridge sequence valve CP241-21 represented with its proper hydraulic symbol.*

7.4.2 Mechanical system

Final design revision

As a last part of the work on the mechanical design of the function model, there has been taken a decision to make a final alteration on the connection between cylinder and tie rod. This measure is carried out in order to increase the lifting height of the plough body closer to 400 mm. Shown below is the revised geometry.

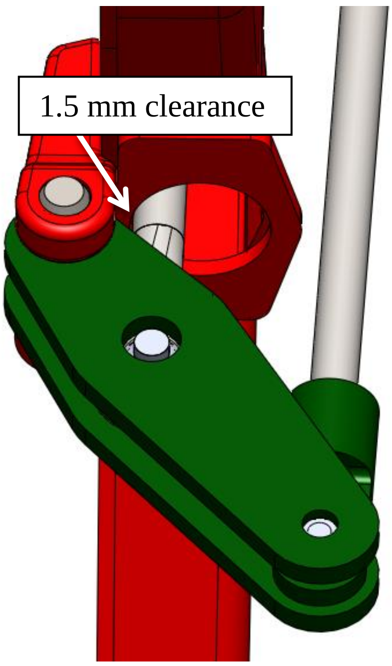


Figure 7.35: *By locating the tie rod as close the inner wall of the beam tube as possible (1.5 mm), there is a maximum of space for the tie rod to travel sideward during the release sequence. New tie plate design and cylinder interconnection on basis of plough beam- and tie rod design obtained from (Kverneland Group 2012).*

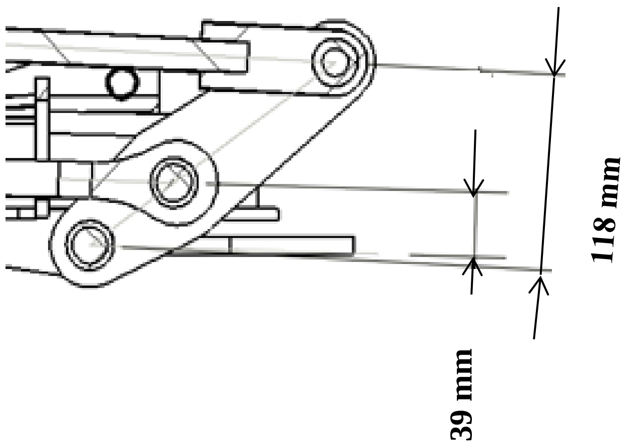


Figure 7.36: *Moment arm z5 is reduced from 52 mm to 39 mm, while z6 is reduced to 118 mm to maintain the same initial moment arm relation 2 as before.*

As a subsequent consequence, the change in geometry induces new release characteristics by calculations shown in appendix A1, where the specific calculations on characteristics regarding the final prototype test can be found as digital copy in Excel-file termed "Calculated release characteristics function model.xlsx" in compact disc accompanying this report. However, the initial relation between moment arms z5 and z6 has been maintained, with the new geometry. The same goes for the relationship between angle 5 and 6, which basically should imply similar release characteristics as before as well.

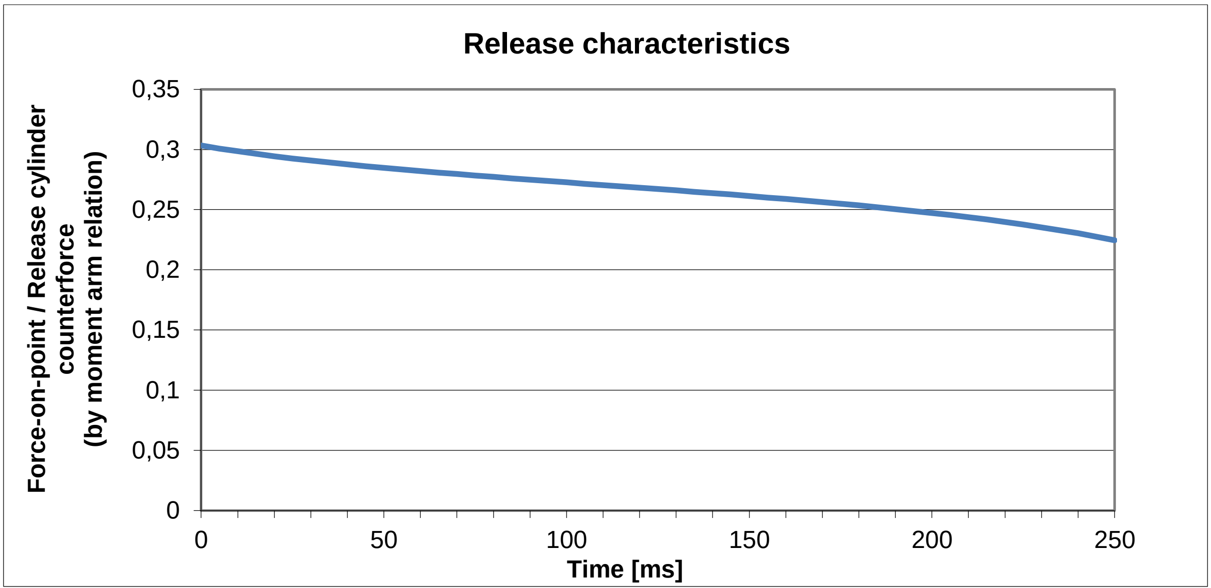


Figure 7.37: *Although the new design implies new release characteristics, the new characteristics are more or less similar with the old one. The release resistance decreases a bit more in the start of the sequence, but the final decrease in force-on-point resistance is similar to the previous design, with a decrease by 25 %. The initial moment arm relation is still at 0.3.*

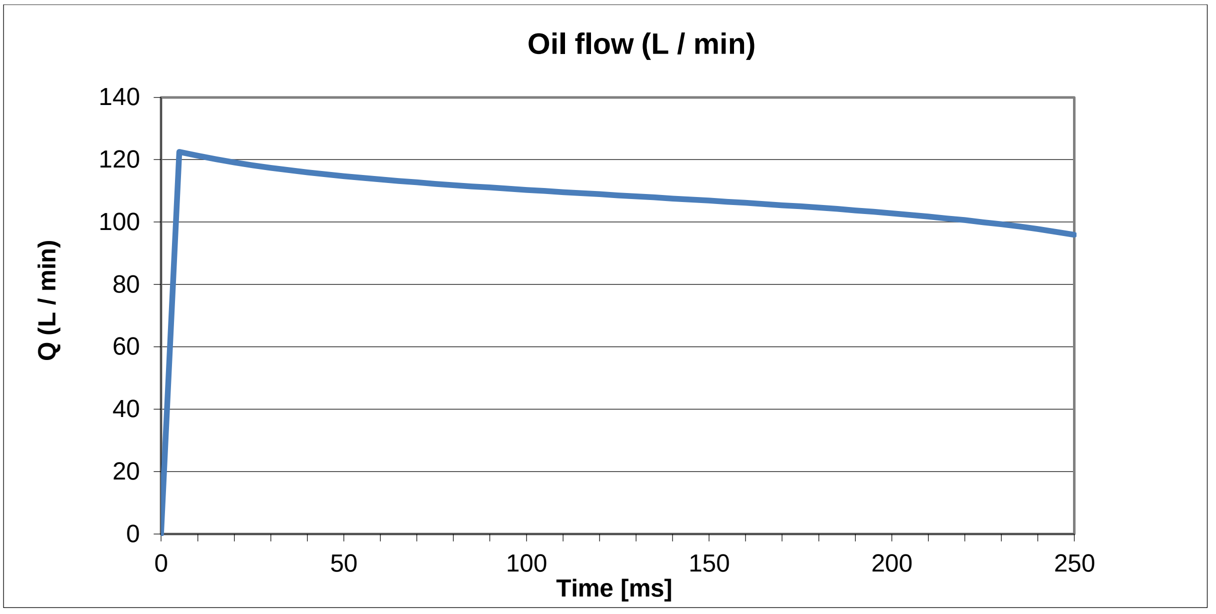


Figure 7.38: *The average oil flow is once more reduced, which is a positive side effect of the new design. However the oil flow at the start of the sequence remains more or less unchanged.*

By using the excel sheet developed as shown in appendix A1, we obtain the following new flow values:

$$Q_{max} = 122.5\ L/min$$

$$Q_{avg} = 108.6\ L/min$$

As already mentioned the average oil flow is a bit lower. In terms of comparing the test results to the fluid dynamic calculations, the new flow characteristics is however still within close enough range to be compared to the previous calculations, with regards to this minor change in design.

Tie plate

As a natural consequence of changing the mechanical geometry between tie rod and cylinder, tie rod has been changed, in the way that it has been made shorter.

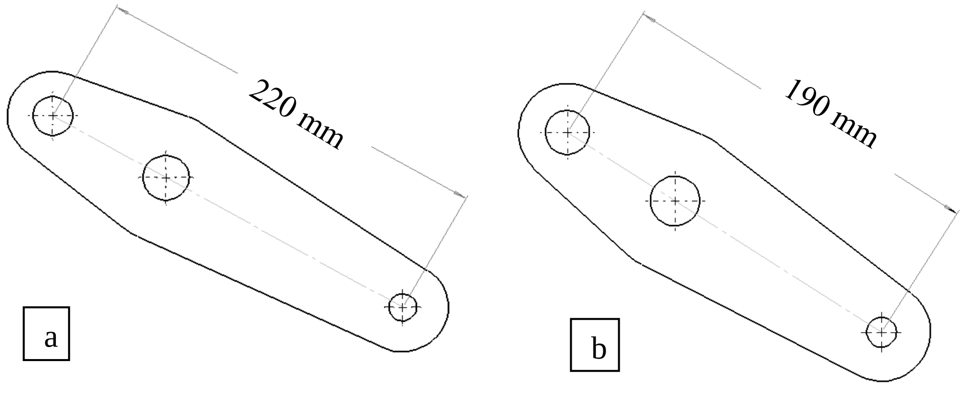


Figure 7.39 a, b: *The old tie plate with 220 mm length, the new plate with 190 mm length.*

The major change in the tie plate relates to its length as illustrated. No recalculations are performed on the tie plate design with regards to this design change, isolated, since the new plate design is on the safe side compared to the previous hand calculations and FEM analysis, with regards to material stress. This assumption is on the basis of the shorter moment arm that the cylinder gets to work on towards the tie rod fixation point on the plate. However, to increase the lifting height of the plough body further, the tie plate should be made narrower, to avoid that it collide too early towards the back of the plough beam. Whether we can do further alterations on the tie plate by making it narrower, depends on whether a high enough yield limit can be achieved by heat treatment, since previous calculation on old design already revealed that we are close to the yield limit. When comparing to the spring regulator used on the Auto Reset system, there has been achieved a ductility within the range of 36-45 HRC (Hardness Rockwell C) upon heat treating the material, obtained from the internal Kverneland material database (Kverneland Group 2013-2). By using a conversion factor, we can obtain an approximate value of the yield strength from the ductility. According to (Erlend Sølvberg 2013), material scientist at Kverneland Group, we can expect to be able to achieve an approximate yield limit of 1080 MPa by heat treating the tie plate. This should make it possible to remove the necessary material on the tie plate to obtain the desired lifting height.

Further FEM-analysis on the redesigned tie plate is not performed, but a recalculation on the equivalent von Mises stress for the cross section at tie rod, is however performed.

Recalculate bending stress at cross section A-A from Figure 7.24, by equation once more obtained from (Terjesen 2012):

$$\sigma_{b,A-A} = \frac{M_{b,A-A}}{W_{b,A-A}} \quad (Eq\ 25)$$

$$M_{b,A-A} = F_{cyl} * (z6 - z5) \quad (Eq\ 26)$$

$$M_{b,A-A} = 53\ 330\ N * (118\ mm - 39\ mm) = 4\ 213\ 070\ Nmm \quad (Eq\ 26)$$

(for updated values for z5 and z6, see Figure 7.36)

$$W_{b,A-A} = \frac{b * h^2}{6} = \frac{(2 * 10\ mm) * ((60\ mm)^2 - (25.25\ mm)^2)}{6} \quad (Eq\ 27)$$

Note that material length (or height as denoted by use of equation 25) has been reduced from 93.7 mm to 60 mm, as seen from previous calculation.

$$W_b = 9\ 875\ mm^3$$

$$\sigma_{b,A-A} = \frac{4\ 213\ 070\ Nmm}{9\ 875\ mm^3} = 426.65\ MPa \quad (Eq\ 25)$$

Calculate perpendicular stress and shear:

$$\sigma_{b_\perp} = \tau_{b_\perp} = \frac{\sigma_b}{\sqrt{2}} = \frac{426.65\ MPa}{\sqrt{2}} = 301.69\ \ MPa \quad (Eq\ 28)$$

Calculate parallel shear stress:

$$\tau_{parallel} = \frac{F_{cyl}}{A_{A-A}} = \frac{53\,330\,N}{(60\,mm - 25.25\,mm) * 20\,mm} = 76.73\,MPa \quad (Eq\ 29)$$

Calculate the uniform stress:

$$\sigma_{uniform} = \sqrt{\sigma_{b\perp}^2 + 3 * \left(\tau_{b\perp}^2 + \tau_{parallel}^2\right)} \quad (Eq\ 30)$$

$$\sigma_{uniform} = \sqrt{301.69^2 + 3 * (301.69^2 + 76.73^2)} = 618\,MPa$$

Control the uniform stress against yield stress:

$$n = \frac{1080\,MPa}{618\,MPa} = 1.7$$

$$\rightarrow OK$$

From the previous calculations and comparison against the FEM analysis, we should expect some higher induced material stress along the edge of the plate, subject to the highest bending moment. However, with the overall safety factor for the entire cross-section, further analysis is not regarded necessary at this point. In order to fit the tie plates in along with the cylinder end fix, the tie plates have been bended, as shown in the final rendered images of the mechanical design in figures below. The final technical drawing on the tie plate can be found in appendix A3.

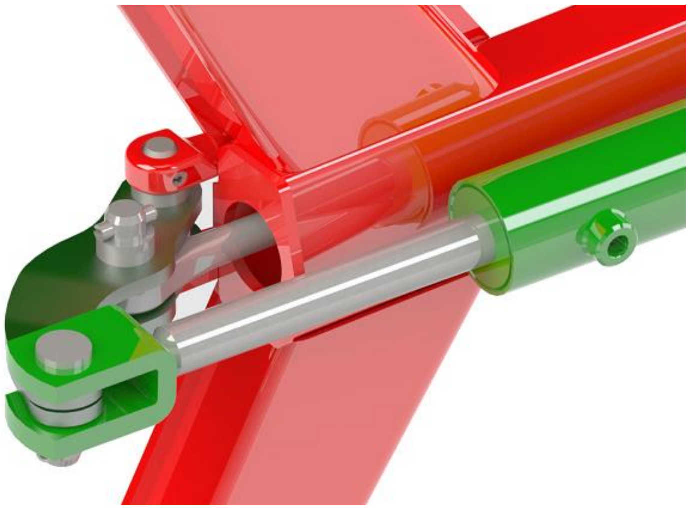


Figure 7.40 (left): *Rendered image of the mechanical design of the system connected onto the plough beam.*

Figure 7.41 (right): *Detail view of the interconnection between tie-rod and cylinder by double tie plates.*

Illustrations represents most recent mechanical design on hydraulic stone release system performed in this project, based on design obtained from (Kverneland Group 2012).

7.4.3 Electronic system

The electronic control of the system is supplied from Kverneland Group Mechatronics. It consists of software, user interface and printed circuit board for input from pressure sensor and output to directional valve.

The electronic control system used in this test consists of software that operates the solenoid operated valve manually, by an activation button on the user interface. On activation of the valve, it is open for a specified duration of time. The opening time of the valve is adjustable in the software. The software will be implemented into the IsoMatch Tellus touch-screen terminal from Kverneland Mechatronics.

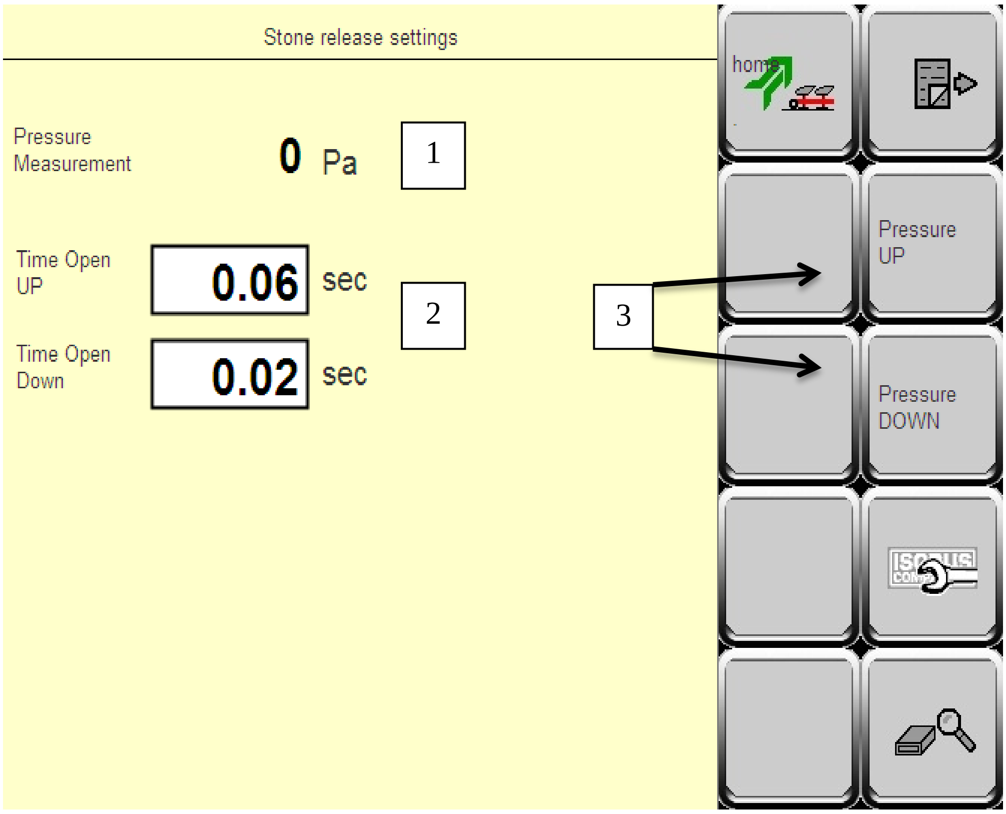


Figure 7.42: *This is the software program with pressure input (1) from sensor and the user input (2) with regards to valve opening time. Additionally the push buttons for activating the valve can be spotted (3).*

Illustration shows software program obtained from (Kverneland Group Mechatronics 2013), developed by Kverneland Group Mechatronics on behalf of this project.

8. FUNCTION TEST

8.1 Test mission

The main mission of the testing is to verify that the system functionality is adequate to the requirements stated in the metric limit specifications in chapter 4 - "Product specification". With regards to the most essential requirements, the test shall reveal whether the system has the required capacity and functionality in order to:

- Keep pressure within +/- 10 % of preset pressure during release (hydraulic system characteristics)
- Keep system resistance during release within 25% of initial system resistance, relative to force-on-point (mechanical system release characteristics)

Secondly the testing shall be used to monitor how the electronic valve control and pressure adjustment interacts with the hydraulic system. The test shall also verify calculations done in chapter 7, as far as it is practically feasible.

<u>The test will consist of three parts:</u>

1. Test of system pressure adjustment and interaction between electronic control of directional valve and hydraulic system as a whole.
2. Verification of mechanical release characteristics and comparison towards the calculated characteristics
3. System function test measuring pressure and flow characteristics, with regards to system performance and valve capacity primarily.

8.2 Test preparations

In order to be as prepared as possible for carrying out the test in practice, some preparations with regards to the system layout and how to carry out the test is performed in the following section. However, alterations to the intended test setup may occur as the test is carried out.

8.2.1 Test system

To enforce the plough body to release, a cylinder will be connected to the plough point with an equivalent force of 16 kN to test the system at the maximum pressure setting. As far as practically feasible, the oil flow into the plough point pull-cylinder should correspond to a velocity of 2.22 m/s, to simulate a driving velocity of 8 km/h. The cylinder shall be allowed to move vertically, and this can for instance be achieved by connecting it to a type of guidance, consisting of a rail to allow for the vertical displacement.

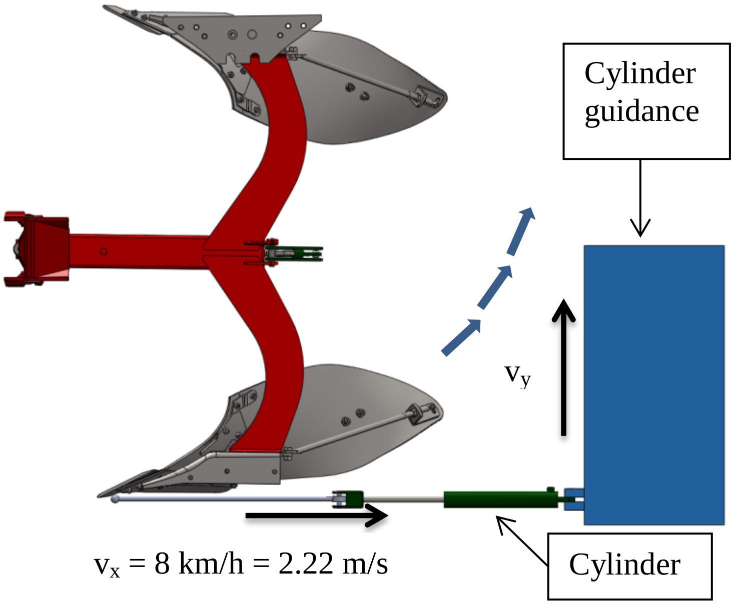


Figure 8.1: *Enforcement of release by use of a cylinder connected to a type of guidance. Part of illustration showing plough beam obtained from 3-dimensional computer model from (Kverneland Group 2012).*

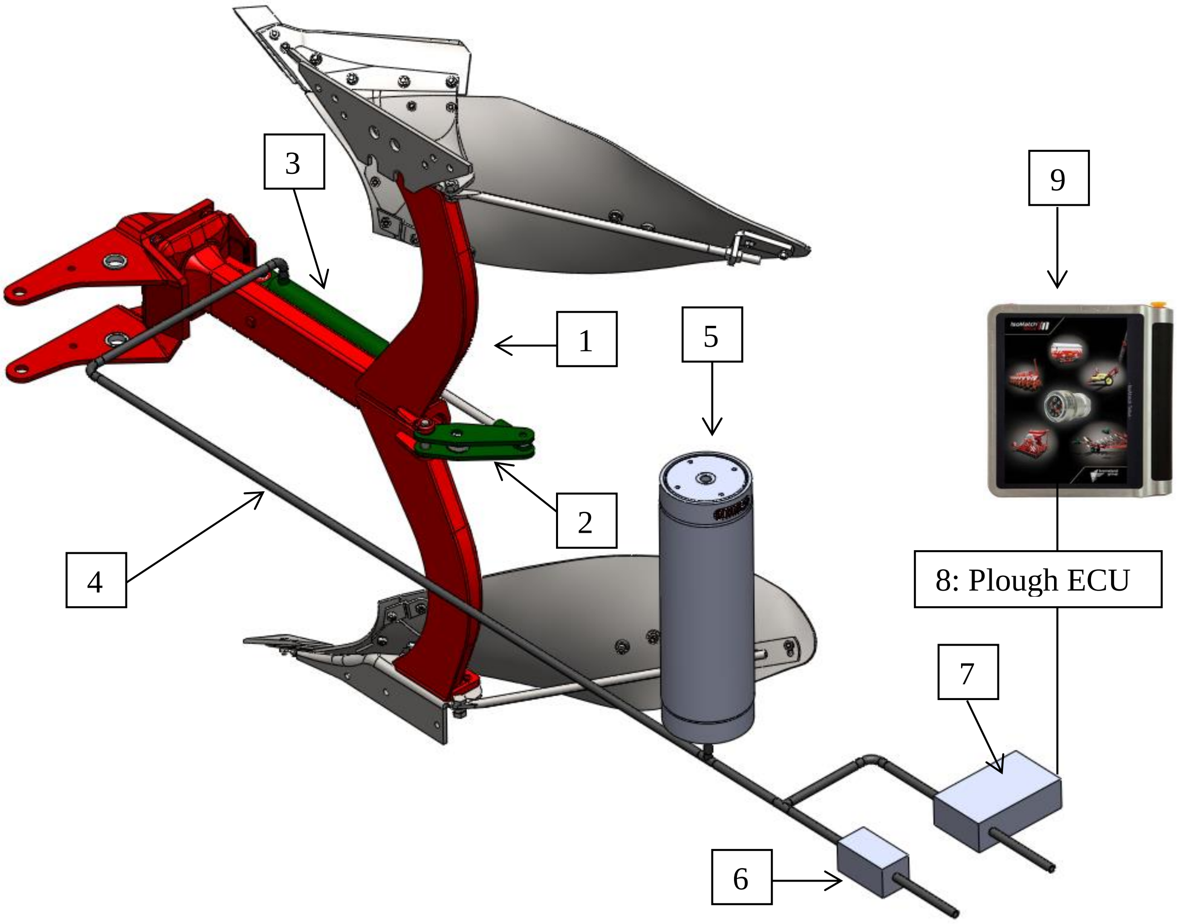


Figure 8.2: *The test system will be built up by the main components illustrated, with plough beam with parts (1), tie plate (2), cylinder (3), pipeline (4), accumulator (5), pressure regulating valve (6), directional valve (7), plough ECU and user interface (8). Part of illustration showing plough beam (1) is obtained from 3-dimensional computer model from (Kverneland Group 2012). Illustration of user interface (9) obtained from (Kverneland Group).*

Table 8.1: *Components used in the test with main specifications defined. The component numbers refers to Figure 8.2.*

Component number	Component description	Specification
1	Plough beam and tie rod	Under-beam clearance 880 beam
2	Tie plate (upper and lower)	(see appendix A3)
3	Hydraulic cylinder	D63-d32 x 400
4	Pipeline	d_i = ½" (approx. 13 mm)
5	Accumulator	Volume: 0.5 L
6a	Pressure sequence valve	Comatrol CP241-21
6b	Pressure relief valve	Comatrol CP211-2
7	Solenoid operated directional valve	Sauer-Danfoss PVG32-PVEO
8	Plough Electronic Control Unit	Printed circuit board with connected pressure sensors
9	User interface	Kverneland IsoMatch Tellus

The plough beam and the tie rod are existing parts. The plough beam shall be of latest design with under-beam clearance of 880 mm as specified. Alternatively the shorter type of plough beam can be used, equipped with an extension piece to maintain the same moment arm towards plough point. There are two options for the pressure valve, where the sequence valve is the primary option. The solenoid operated directional valve is operated by the user interface, through the plough ECU. The plough ECU will have input from two pressure sensors, measuring pressure on both "tractor side" and "plough side" of directional valve to keep track of system pressure in both plough hydraulic circuit and the pressure delivered from tractor.

8.2.2 Measurements and data logging

In addition to the pressure sensors connected to the printed circuit board in the plough ECU, both pressure and flow will be measured using a measurement system from Parker. The measured data input will be handled by their Senso Control software. The measurement device has four inputs, where using 1 flow sensor, 1 temperature sensor and 2 pressure sensors.

Figure 8.3: *The measuring device and user interface for test-data input will be the Parker Service Master, which takes pressure-, flow- and temperature input. Illustration obtained from (Parker Hannifin Corporation).*

Measurement- points and locations

The sensors will be distributed along the pipeline at locations giving a best possible indication of system characteristics for the three types of tests outlined.

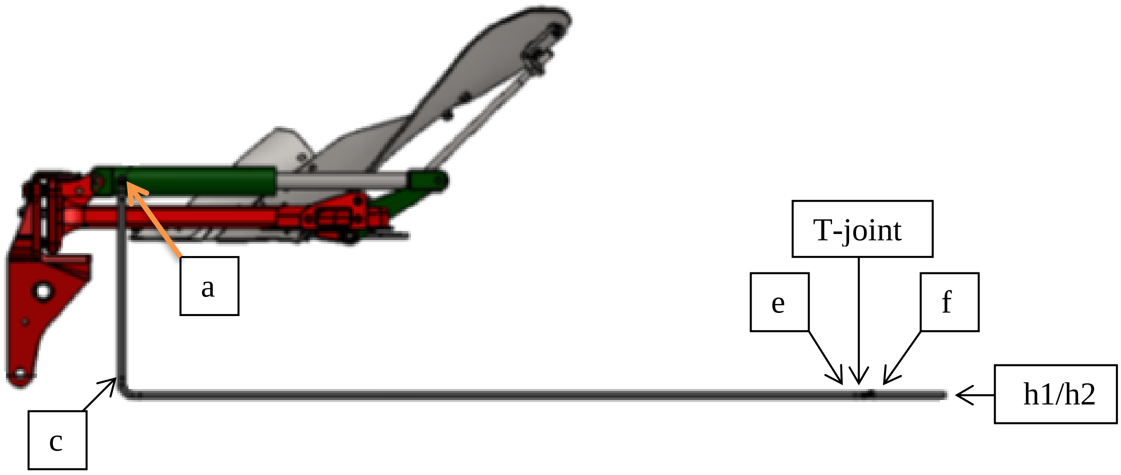


Figure 8.4: *Test measuring locations at nodes a, c, e, f and h specified from Figure 7.9 and Table 7.8. Node h has been split into h1 and h2 for this test, where h1 refers to pressure valve inlet and h2 refers to directional valve inlet/outlet (depending on flow direction). The specific distribution of sensors at nodes is specified in Table 8.2. Part of illustration showing plough beam is obtained from 3-dimensional computer model from (Kverneland Group 2012).*

Table 8.2: *Measurement location and type of output.*

Measurement location	Type of measurement
Node a	Flow
Node a	Pressure
Node a	Temperature
Node e	Pressure

As seen from Table 8.2, all measurements will be achieved by locating pressure-, flow- and temperature measurement at the cylinder. Additionally system pressure will be logged at node e, upon location of accumulator. This is to keep track of possible pressure differences within the system.

8.2.3 Test 1: System pressure adjustment

The first part of the testing consists of testing the system pressure adjustment and the interaction between the hydraulic system and the electronic control system.

Test objectives

1. Logging oil flow and pressure decrease for one opening sequence of the valve when opening valve to tank
2. Logging oil flow and pressure increase for one opening sequence of the valve when opening valve for system pressure increase.

Hydraulic schematics

In the testing of the pressure adjustment the Comatrol CP241-21 valve (6a in Table 8.1) will be used, due to its adjustment functionality that is required for the system in the longer run. The valve description can be found in Table 7.2. The system itself is connected to the pressure port of the cartridge valve sharing the same inlet-flow as the pilot circuit connected to the spring chamber of the cartridge valve when oil is added through the directional valve. Note that the system illustrated below is the same as in Figure 7.4, but with the cartridge valve symbol updated with the proper symbol for the specific valve.

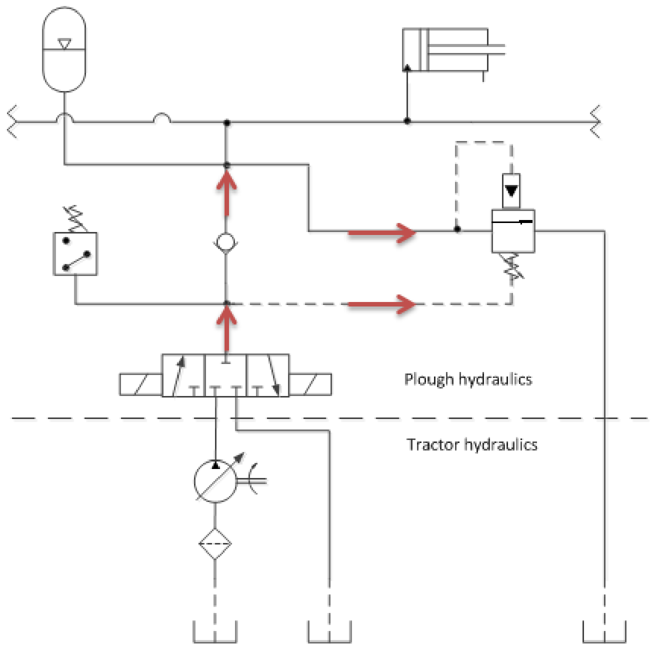


Figure 8.5 a: *Oil is added through directional valve by pushing "pressure up" button on the Tellus terminal illustrated in Figure 7.42. The added oil goes into pilot port and main system, simultaneously increasing system pressure while cartridge valve remains balanced due to the spring force biasing the pilot side of the valve.*

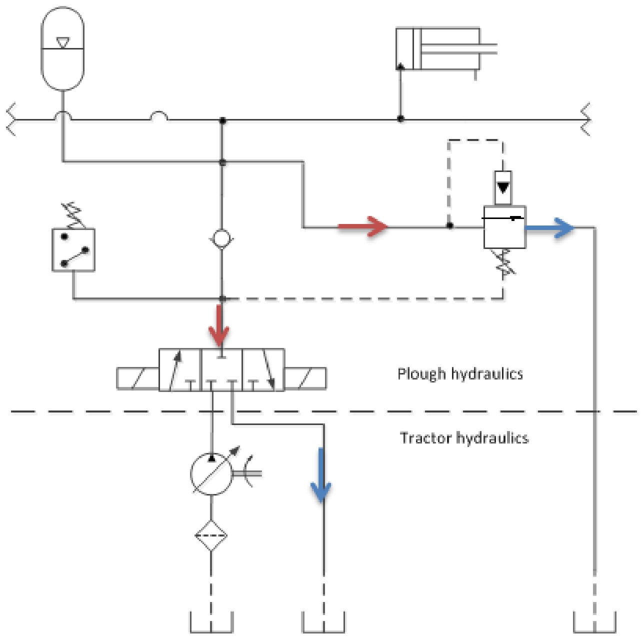


Figure 8.5 b: *When decreasing system pressure, oil is evacuated from pilot circuit through directional valve. This is done by pushing the "pressure down" in the terminal, also illustrated in Figure 7.42. At the same time, the check valve between main- and pilot circuit prevent the oil in the system from being evacuated. This causes a pressure difference between pilot and system, and oil is drained from main system through cartridge valve until pilot-/main system pressure is equalized at the new pressure.*

8.2.4 Test 2: Mechanical release characteristics

In test 2 the aim is to obtain the mechanical release characteristics by measurement, where we get all answers of the influence of mechanical friction and weight of plough beam. An important part of the test is also to be able to compare against the calculated characteristics before taking the test the next step further.

8.2.5 Test 3: Hydraulic system characteristics

After the test of pressure adjustment feature and obtaining the mechanical release characteristics, testing of the hydraulic system characteristics comprises the next step of the testing.

Test objectives
The testing shall include pressure-, flow- and temperature measurement at different locations along pipeline between cylinder and valve as outlined in Figure 8.4 and Table 8.2.

The measured data shall be utilized by:

1. Comparing measured pressure and flow against previously calculated characteristics as far as it is practically feasible.
2. Verifying the system functionality and whether it operates within the system requirements.

Temperature measurements are carried out in order to keep track the overall system conditions. Pressure characteristics are measured and related to the measured flow characteristics.

Hydraulic schematics

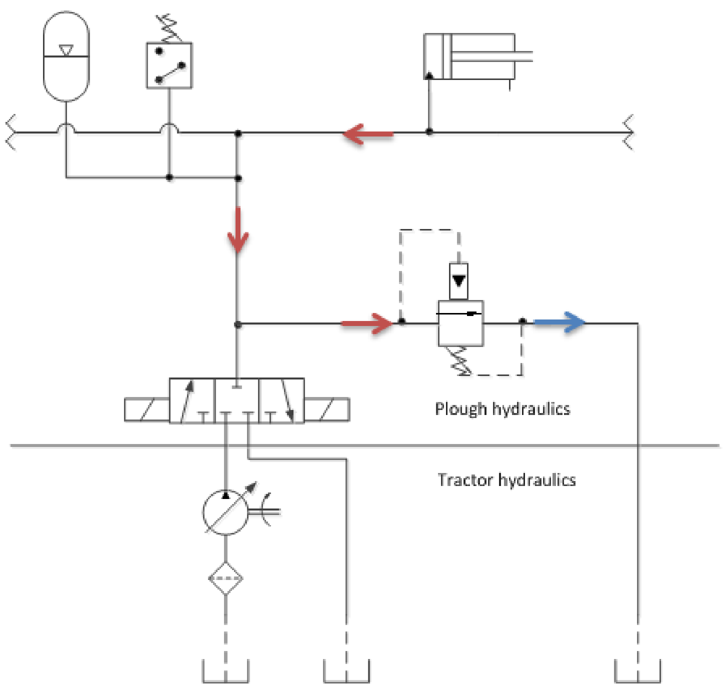


Figure 8.6: *This is the hydraulic schematics with the proper symbol for the mechanical adjustable valve CP211-2. Arrows show oil flow during release, which is also applicable for valve CP241-21.*

In both test 2a and 2b, the testing shall be carried out by testing both Comatrol valves in terms of the CP241-21 sequence valve, and the CP211-2 relief valve, where the idea is to make a final comparison between those two.

8.3 Test setup

Before reaching the point of doing the test measurements and retrieving data on the system behavior and characteristics, the test need to be put up, where assuring its basic functionality also is an important point. Some issues to the basic system functionality arose as putting up the system, especially since the system is consisting of both an electronic and hydraulic system that need to function together. Some smaller alterations on the hydraulic system were also done to improve the system functionality as described later on.

Due to late delivery of the sequence valve the test were only performed by using the mechanical adjustable relief valve, CP211-2. That implies that both the system pressure adjustment test and test of hydraulic system characteristics during release need to be carried out using the mechanical adjustable pressure relief valve. It does not affect much of the testing, but the pressure adjustment test must be carried out by tensioning the mechanical spring of the relief valve above maximum system pressure and do the pressure adjustment without the pilot pressure control as seen in schematics from Figure 8.5a -and b.

Final hydraulic system test setup

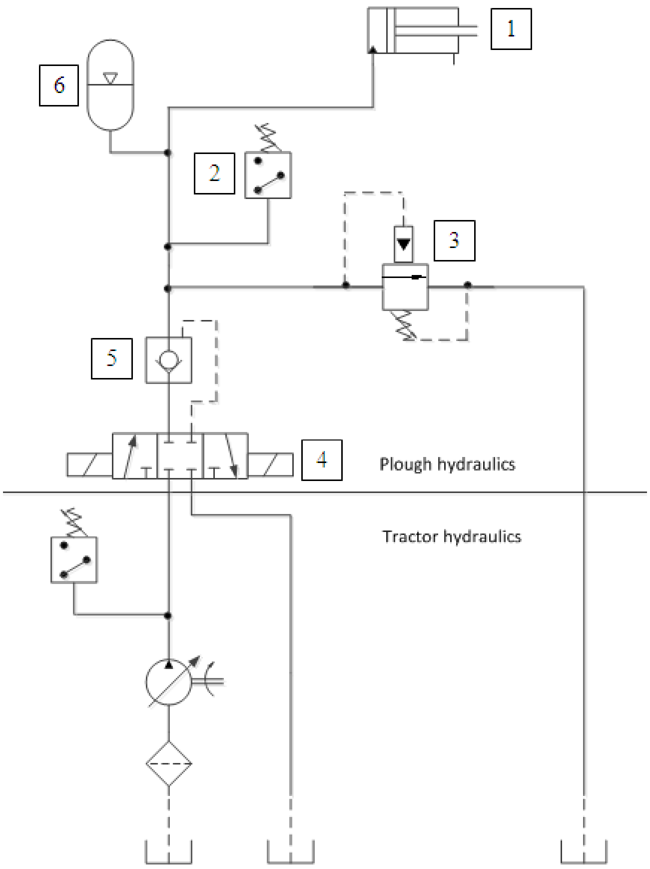


Figure 8.7: *The final schematics of the hydraulic system test setup with the CP211-2 Comatrol valve.*

A pilot-operated check valve (5) is implemented, since the directional valve (4) will have some internal leakage due to its spool valve design. The check valve, which is a seat-valve, should be able to keep the internal system leakage down in an acceptable manner.

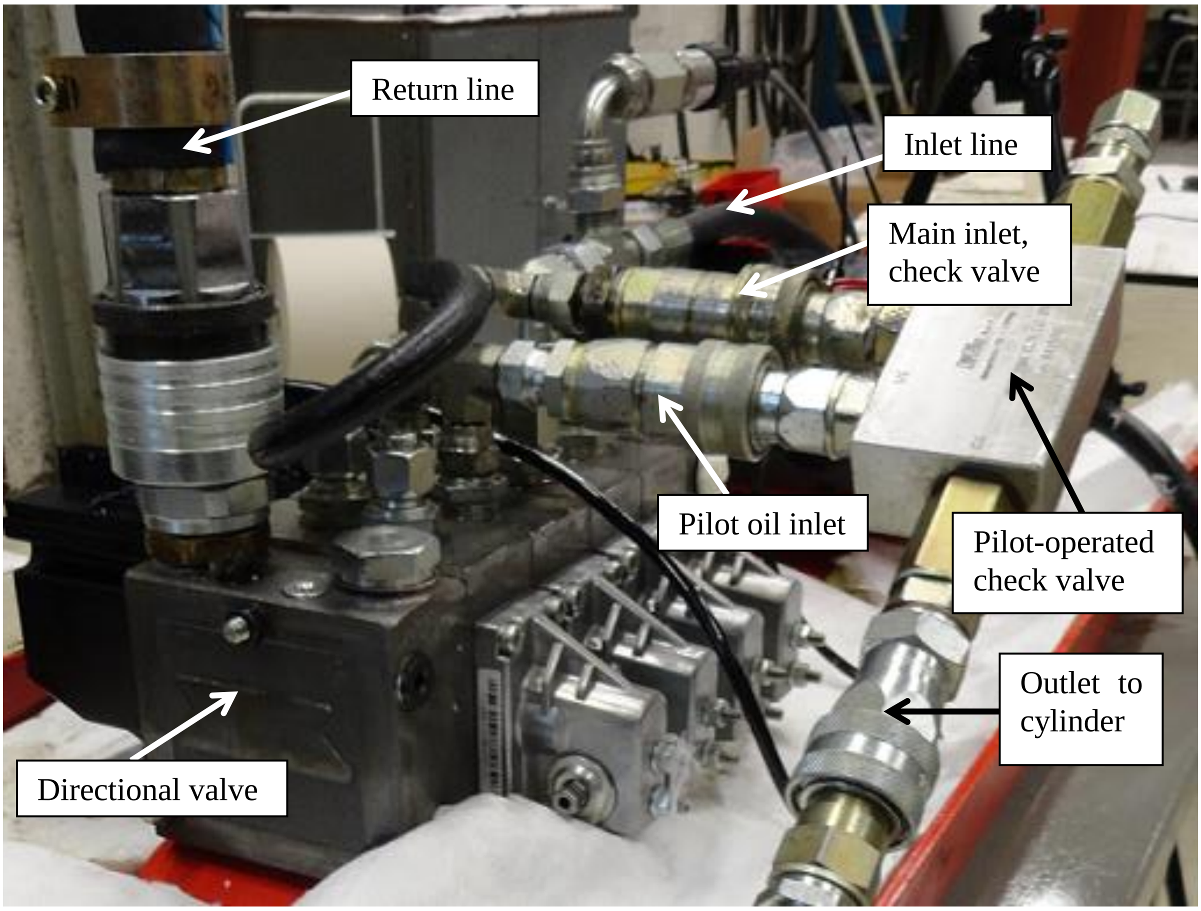
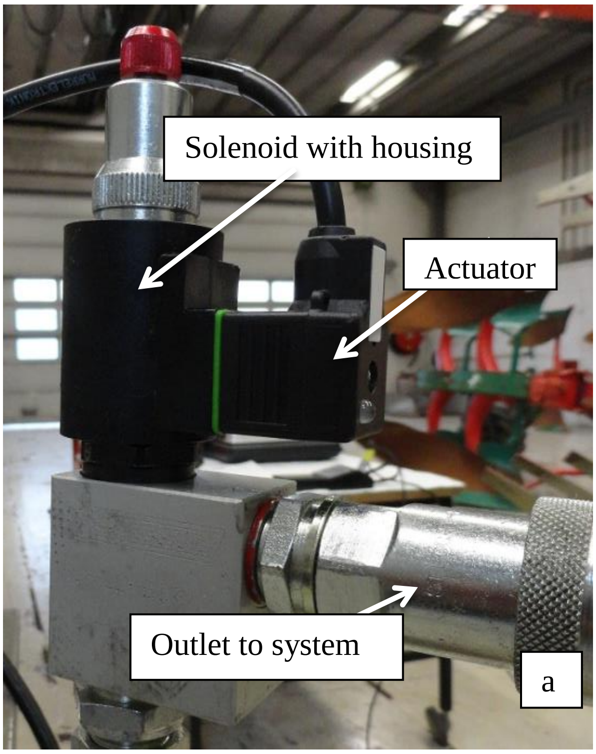


Figure 8.8: *A central part of the system is the directional valve. In this illustration all connections are coupled according to the schematics in Figure 8.7. Illustration by own photo.*



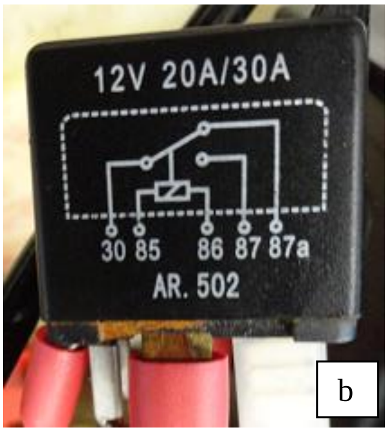
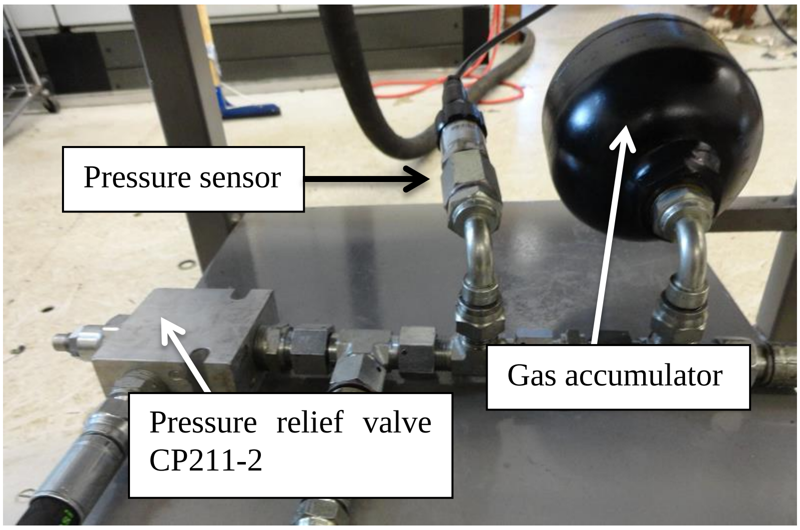


Figure 8.9 a, b: *As an alternative to the pilot-operated check valve an electronic-operated check valve can be implemented. It was wired together with the valve control of the directional valve by using a changeover relay (b) and tested along with the system. Illustrations by own photos.*

Figure 8.10: *The relief valve/accumulator setup in connection with the system pressure sensor giving user output of the system pressure, and hence, its resistance. Illustration by own photo.*



To keep track of as much as possible of the restrictions in the system, pipeline diameters and lengths are listed in Table 8.3 below. Note that the hose lengths in the test setup is increased compared to what listed in the 'preparations' section for the testing. This is due to that the arrangement of the system in relevance to tractor outlets and tank became a bit different, which required longer hose lengths.

Table 8.3: *Final pipeline dimensions.*

Pipeline description and location	Hose length	Hose inner diameter
Inlet line	6.6 meters	½ inch
Cylinder – relief valve	4 meters	½ inch
Relief valve – directional valve	4 meters	3/8 inch
Outlet line	15 meters	1 inch

Electronic system and software

The electronic system and software has been delivered by Kverneland Group Mechatronics, and was implemented as control system of the Sauer Danfoss PVG32 directional valve.

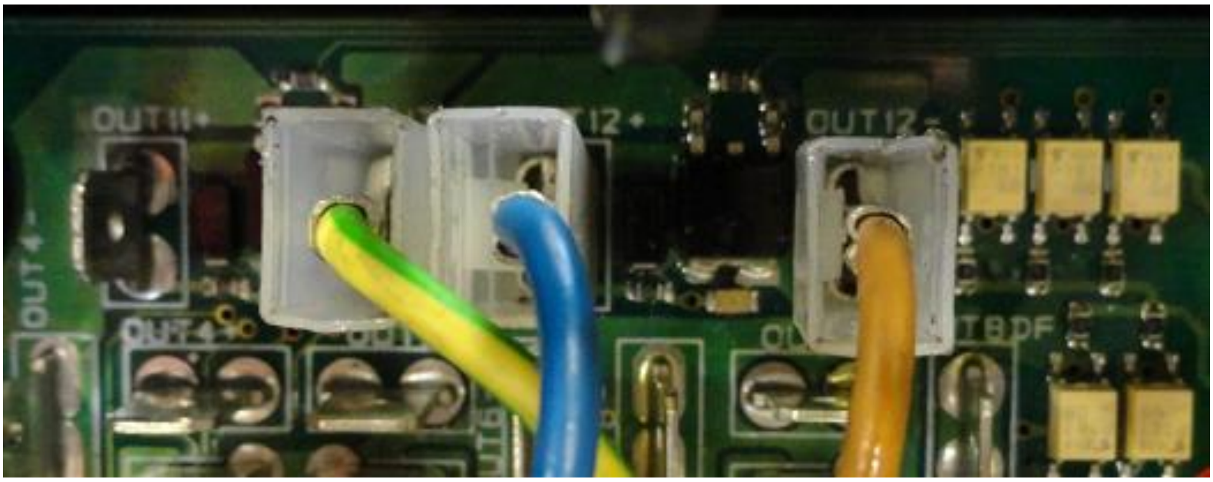


Figure 8.11: *The signal output cables for the directional valve control were connected onto the printed circuit board according to the instructions following the software delivered by Kverneland Group Mechatronics. Illustration by own photo.*

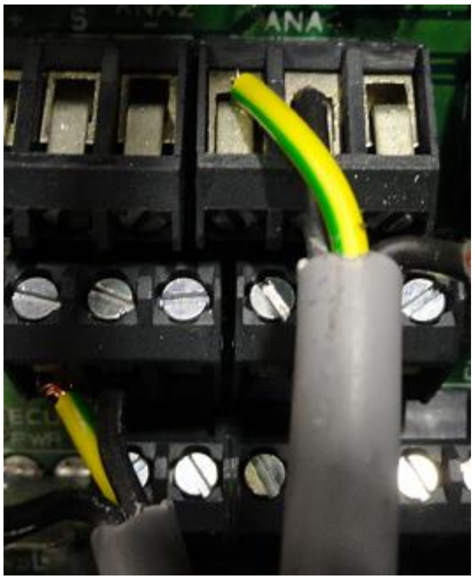


Figure 8.12 (left): *Signal cables and ground connected to the valve actuator with solenoid for to-way actuation of the spool on the directional valve.*

Figure 8.13 (center): *Connection between pressure sensor and cable from printed circuit board.*

Figure 8.14 (right): *Pressure sensors cables connected to circuit board.*

Illustrations by own photos.

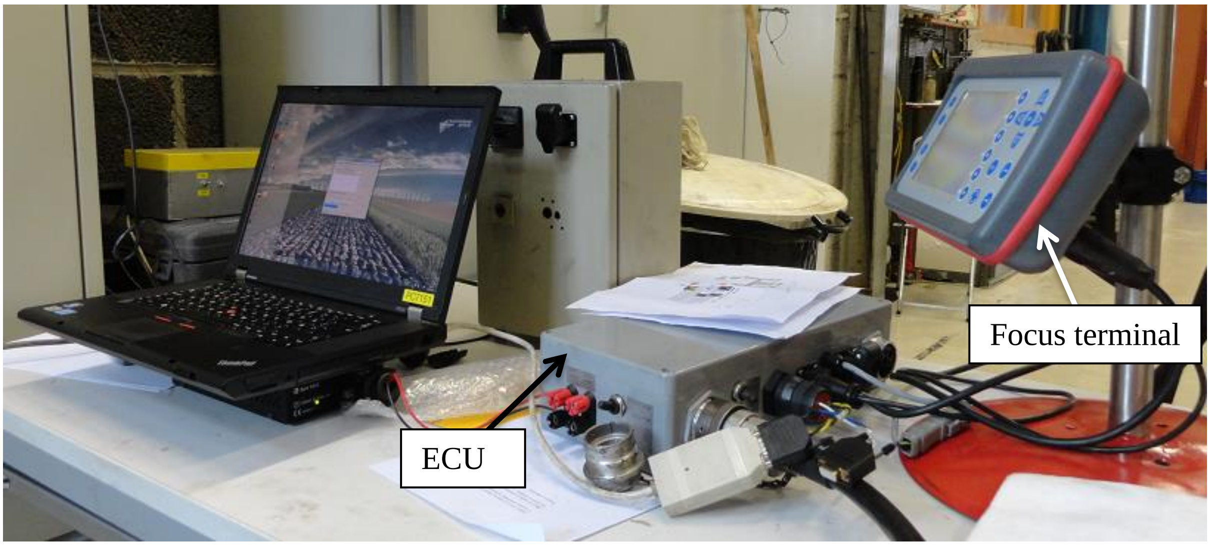


Figure 8.15: *Flashing the software from personal computer through USB outlet into the electronic control system in terms of the printed circuit board and the user interface. For the test the Kverneland Focus terminal were used as user interface, not the Tellus. Anyway, the software is compatible with both. The software is flashed from the personal computer through external ECU (Electronic Control Unit) illustrated and into the Focus terminal and plough ECU with printed circuit board. Illustration by own photo.*

Mechanical system
Tie plate were produced according to technical drawing in appendix A3, by Kverneland R&D workshop. Found below is the design how it turned out on the plough beam.


Figure 8.16 (left): *This is how the mechanical system with tie plates and cylinder turned out on the plough beam.*

Figure 8.17 (right): *When pressurized at only around 50 bar, the cylinder rod has good clearance to the beam, as well as the tie rod.*

Illustrations by own photos.

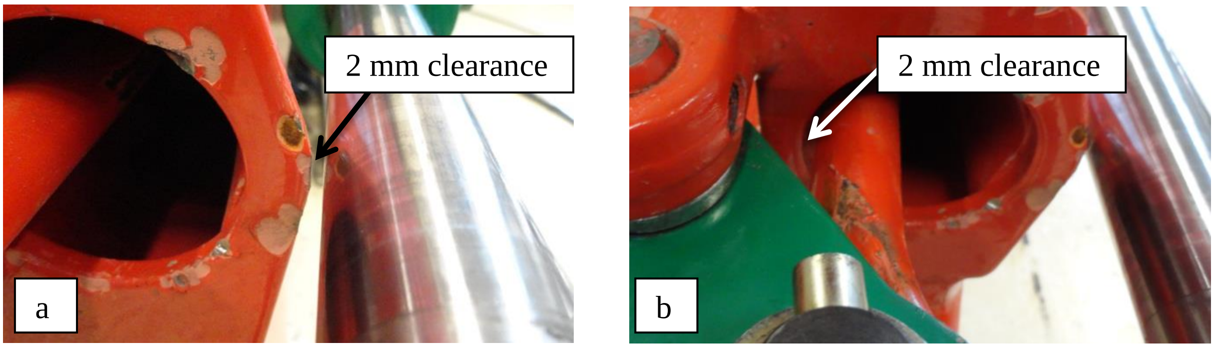


Figure 8.18 a, b: *When the system is pressurized towards the maximum pressure setting, where in this case around 150 bar, the cylinder rod (see arrow figure a) and the tie rod (see arrow figure b) has a clearance of around 2 mm to the beam at both locations which is also indicated by the design performed in computer aided design. Illustrations by own photos.*

The tie rod should be located as close as possible to the edge of the rear opening of the plough beam. This measure should be taken order to reduce moment arm z5 to a minimum, which gives maximum sideward travel of tie rod, resulting in increased maximum lifting height in terms of mechanical stop. Secondly the cylinder piston rod is located as close as possible to the outer plough beam wall to be in angle with the tie rod to direct the maximum of its applied force to the system. However the cylinder rod should be considered located a bit further away from the beam in case of shocks to the system where the tie rod can easily brush into the beam with such a small clearance already in the initial static load case.

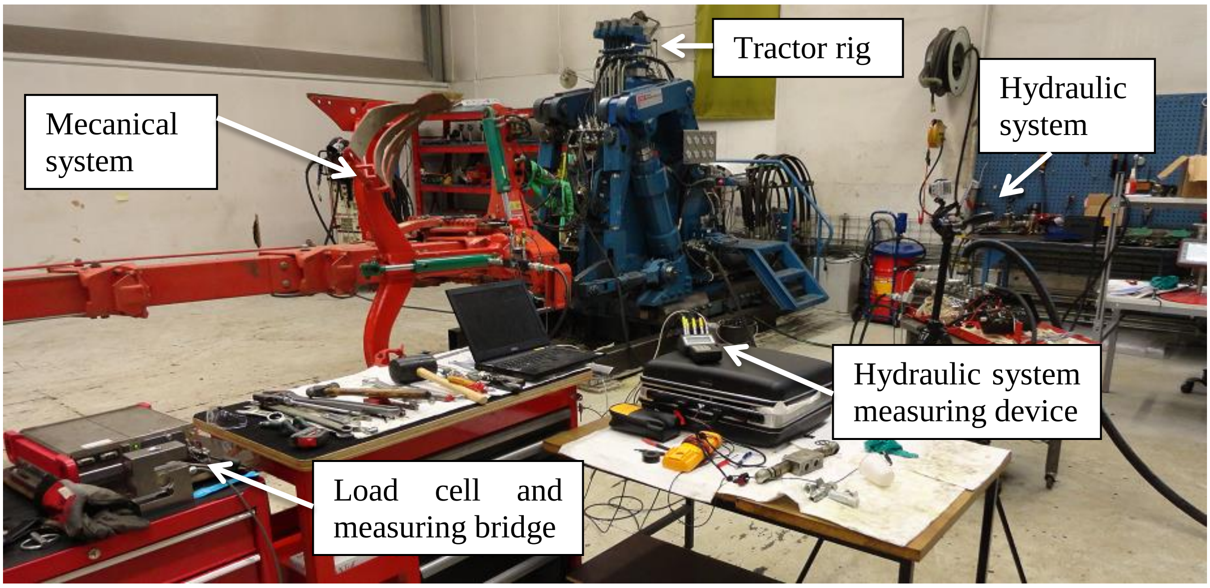


Figure 8.19: *The final test setup with mechanical system connected to plough. Hydraulic system with electronic directional valve control was connected to the tractor test rig at the test facility of Kverneland Group at Klepp. Equipment to measure system pressure, flow and temperature of oil, in addition to load cell and measuring bridge for measuring plough point load during mechanical release characteristics measurement were implemented. Illustration by own photo.*

8.3.1 Technical issues

Some technical issues became clear when putting up the system. In the following section the most essential ones are briefly discussed.

Return flow and pressure

One major problem was that a pressure build-up in the return line arose from the directional valve. Oil was diverted to external outlets on tractor rig, while oil was added from opposite external outlet, where the return line had a hose diameter of ½” (inch). This caused a pressure of 50 bar in the return line, which is above the allowable 25 bar limit for the PVG valve, obtained from (Sauer-Danfoss 2012). The solution became a 1” hose return line, which decreased the return pressure below 25 bar. Some of the reason that the return line needed such a big diameter is that the return line had to be 15 meters in order to reach the test rig oil reservoir. So if connected to a free return onto a tractor, with shorter distance from valve to reservoir, a smaller hose diameter should be sufficient in order to drain oil and still keep the return pressure down below 25 bar.

System internal leakage

There were some problems with internal leakage over the directional spool valve, which was solved by implementing a pilot-operated check valve as previously described. But still after that implementation there were a minor internal leakage in the system, since no external leakage were observed. If that internal leakage is caused by leakage over cylinder piston, pressure relief valve or somewhere else is difficult to say without any further investigation. But for further prototyping of this system this possible problem should be kept in mind, since pressure loss due to even smaller oil leakage can be critical to the system functionality in the longer run.

Temperature

The test prototype utilizes constant oil flow from tractor in order to have instant access to oil flow when valve is directed to increase system pressure. However this will cause a temperature rise in the oil. How much that temperature rise will be naturally depends on the oil reservoir volume from oil source (tractor) and type of oil cooling system implemented. What should be kept in mind for this system is to keep oil flow restrictions down to a minimum to avoid pressure loss and subsequent oil temperature increase. The oil temperature was measured for this test, but due to large oil reservoir volume it was not measured any temperature increase in system worth worrying about. However, with a smaller oil reservoir and lack in capacity of oil cooling system, issues related to high oil temperatures might occur in this type of system and should be beard in mind in future development of the product.

8.4 Test results and results interpretation

In the following section the test results are presented for the three types of tests conducted on the system. The Parker Hannafin hardware along with the Senso Control software has been used for the measurements as described in the test preparations. Comparing the test results towards the calculations carried out on the hydraulic system became a bit more challenging than expected, since the combination of oil flow and pressure out on the pull cylinders on the plough point was not sufficient to simulate a release sequence at 8 km/h with a 170 bar maximum system pressure setting for this test. However, where applicable, the rest results have been compared to the calculated system characteristics. The set-up of the system is not completely identical relative to the intended pipeline system which was outlined in the preparations, so measurement locations are specified once more along with illustrations for each of the tests carried out.

Discussion of test results

Along with the presented test results, the results are interpreted continuously throughout the rest of chapter 8. Furthermore the test results are discussed in terms of system characteristics, measurements and how the system meets the metric limit specifications. As an answer to the issues raised by the discussion of the test results, next chapter comprise product design revision. In that chapter, alterations on design are performed, by calculation and from reasoning related to interpretations and discussion of test results. Consequently, much of the content in both this test chapter and the design revision could belong as a part of the discussion. But for the structure and chronology of the report, it is decided to do the discussion of the test results along with the presentation of the test results in its entirety.

8.4.1 Test 1: System pressure adjustment

In order to see how well the system is loaded and unloaded in terms of system increase and decrease, system pressure is increased from 0 bar to around 200 bar and decreased back to around zero bar.

Inlet flow from tractor

Figure 8.20: *To keep track of the inlet flow into the directional valve, oil flow delivery from the tractor outlet was measured. Illustration by own photo.*

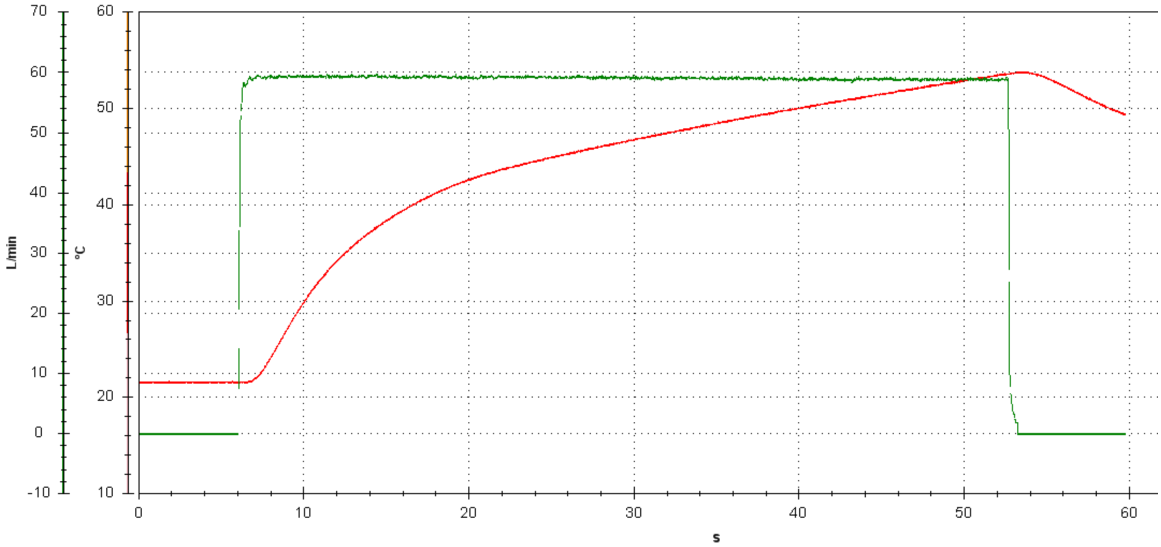


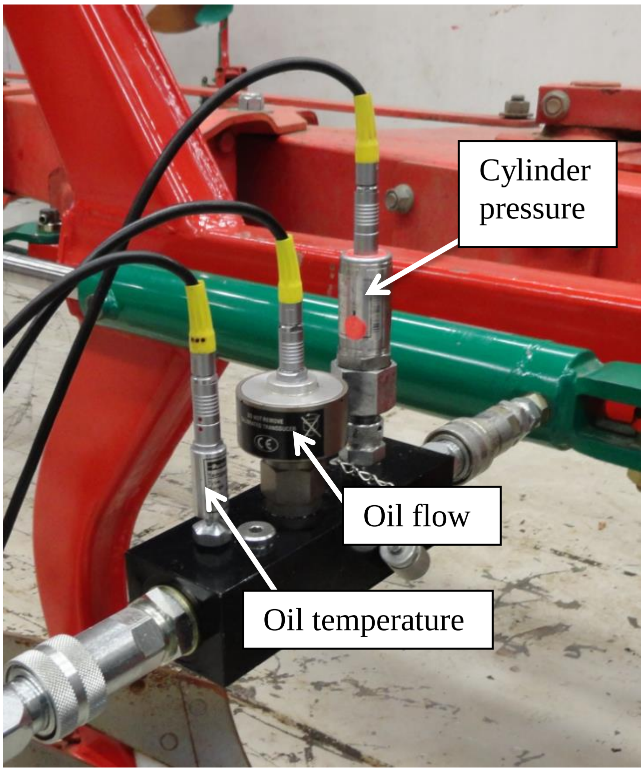
Figure 8.21: *The measured oil flow from tractor outlet is at 60 L/min and forms the basis for the pressure adjustment measurements. Make notice to how the temperature rises from about 20 to 54 degrees C for the sequence from 6 to 53 seconds where oil is flowing from tractor pump. However the temperature will be stabilized at some point given sufficient oil reservoir and cooling system. Graphical output obtained from own test measurement.*

Test and measurements

Figure 8.22: *To avoid that the pressure relief valve burst and interfere on the pressure adjustment measurements, the mechanical spring is tensioned to maximum burst pressure, which is rated to 400 bar in product leaflet, according to (Comatrol-2 2010). Illustration by own photo.*

Table 8.4: Graphical measurement output overview.

Colour code on graph	Type of output	Measurement unit
Blue	Cylinder pressure	bar
Green	Oil flow at cylinder outlet	L/min
Red	Oil temperature at cylinder outlet	°C
Purple	Accumulator pressure	bar



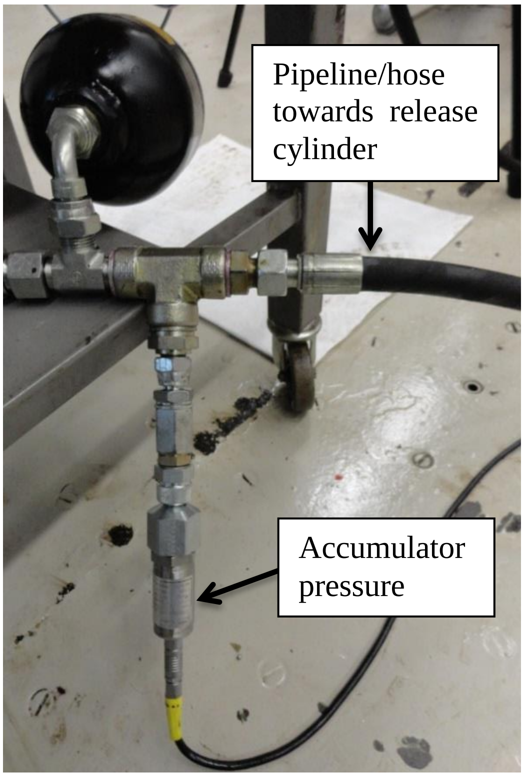


Figure 8.23 (left): *Measurement point location for cylinder pressure, flow and temperature.*

Figure 8.24 (right): *Measurement point for the accumulator pressure. Oil line towards cylinder is indicated as shown.*

Illustrations by own photos.

The accumulator used for this measurement has a pre-charge pressure by 140 bar. This is actually too high to be implemented into the system in the longer run, since the pressure range has been specified between 100-160 bar. But for this test it was found ok, since the intention was to test the system at maximum pressure up to 160 bar or even 170 bar.

However after doing the first measurements it became obvious that the pull cylinders on the plough point were not able to overcome system resistance at 160-170 bar and definitely not with a velocity of 8 km/h. That meant that for most measurements the accumulator only caused disturbances in the measurements or was not able to influence on the system at all. Therefore it was focused on doing measurements without the accumulator. Since the aim for this test is primarily to detect the relief valve function and capacity, leaving the accumulator out of the measurements are generally a good idea for that cause. In future testing of this product however, there should be implemented an accumulator with the correct pressure pre-charge and with the sufficient accumulator volume displacement to cover up for potential lack in capacity at relief valve and for minor shock absorption, as previously mentioned.

For this test 1 regarding the pressure adjustment, there`s been chosen to include measurement both with and without accumulator to illustrate the differences in loading and unloading the system with and without an accumulator when loaded up to about 210 bar which is the maximum pressure delivered from tractor.

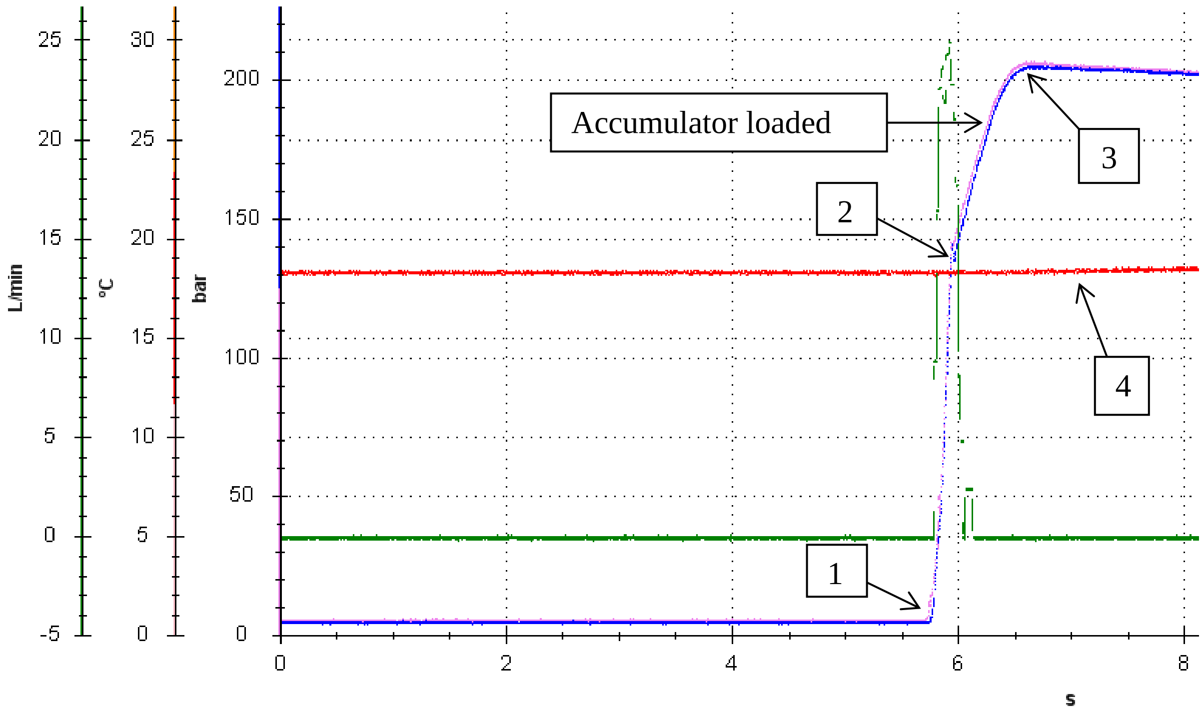


Figure 8.25: *Graphical output on how the system is loaded with accumulator with 140 bar pre-charge pressure. The system is loaded from 0.5 bar (1) to about 140 bar (2) in about 0.2 seconds, indicated by the blue graph (cylinder pressure) and purple graph (accumulator pressure). The accumulator pressure is just slightly above the cylinder pressure during the pressurization, since it is closer to the energy source in terms of the tractor oil supply. For further pressurization from 140 to 210 bar (3) it takes another 0.6 seconds. So in total pressurizing the system from 0 to 210 bar takes about 0.8 seconds in total. The flow (green graph) peaks at 25 L/min before reaching zero at the last part of the pressurization from 150 to 210 bar where the accumulator is loaded, hence no flow at cylinder. The temperature is at 18 degrees C for this measurement. The temperature is increasing (4) with about 0.3 °C or K. The pressure measured at the cylinder is lagging the measured pressure at the accumulator with only some milliseconds. Graphical output obtained from own test measurement.*

Out of the graphical output it is very clear how the accumulator slows down the rate at which the system pressure increases from point (2) at around its pre-charge pressure at 140 bar, caused by that extra system volume that needs to be loaded. The small temperature rise at 0.3 K is most likely to come from the heated circulating oil, which enters the system when directing oil from tractor.

For the first part of the pressure decrease, from 200 to 120 bar illustrated in Figure 8.26, oil doesn't flow from cylinder, since accumulator is being unloaded. When accumulator is unloaded oil starts leaving the cylinder at a flow rate by approximately 50 L/min and decreasing.

As a comparison, the system has also been loaded and unloaded without the use of an accumulator, and by measuring the flow closer to the directional valve. When looking at Figures 8.27 and 8.28, the system is obviously more effectively loaded and unloaded by leaving out the accumulator. The system is reacting quite fast in terms of the pressure increase following the oil flow. However, it is important to note that the reaction time of the directional valve, which is rated at 0.18 seconds according to (Sauer-Danfoss 2012), should be added on to the time it takes to increase and decrease the pressure.

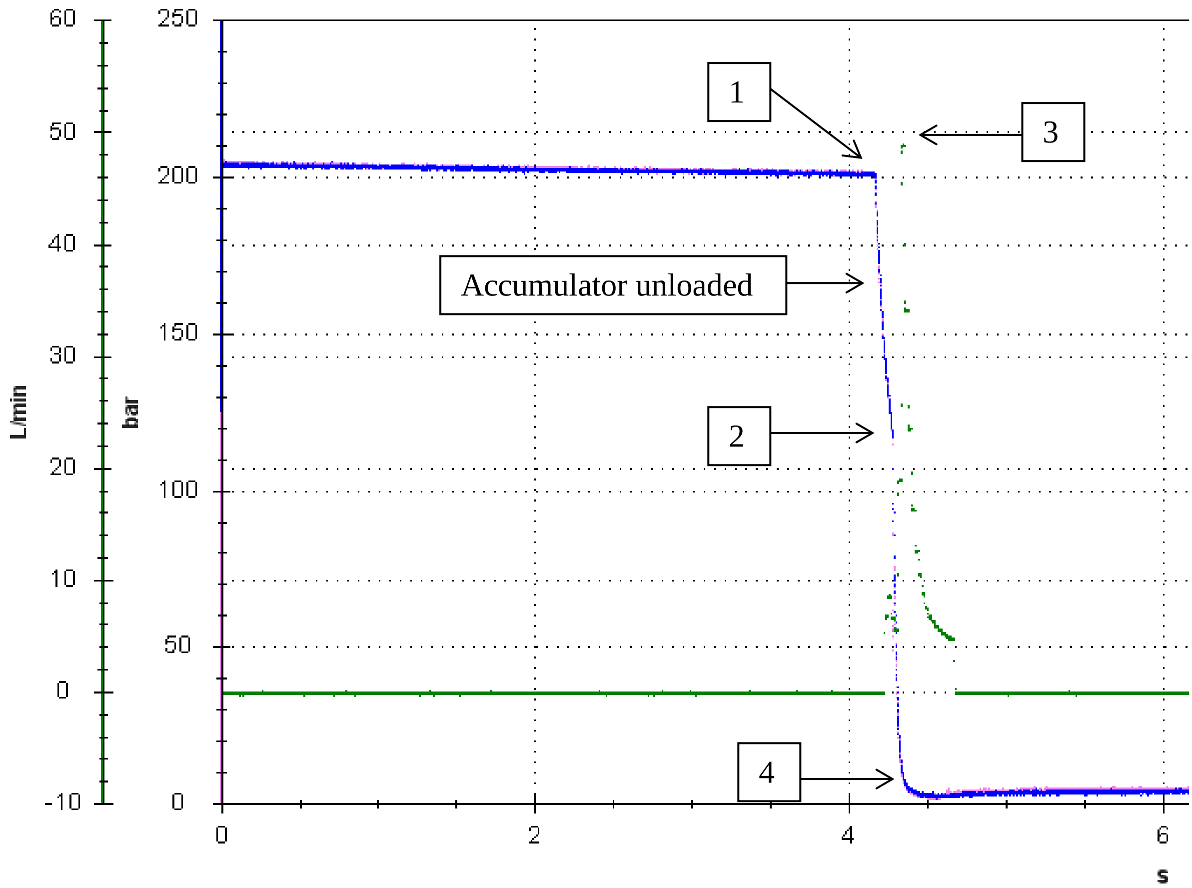


Figure 8.26: *Unloading the system with accumulator. At first, between point 1 and 2 the accumulator is unloaded, while oil leaves the cylinder with flow peak at around 50 L/min at point 3. At point 4 the system is almost fully unloaded and the pressure evens out. Graphical output obtained from own test measurement.*

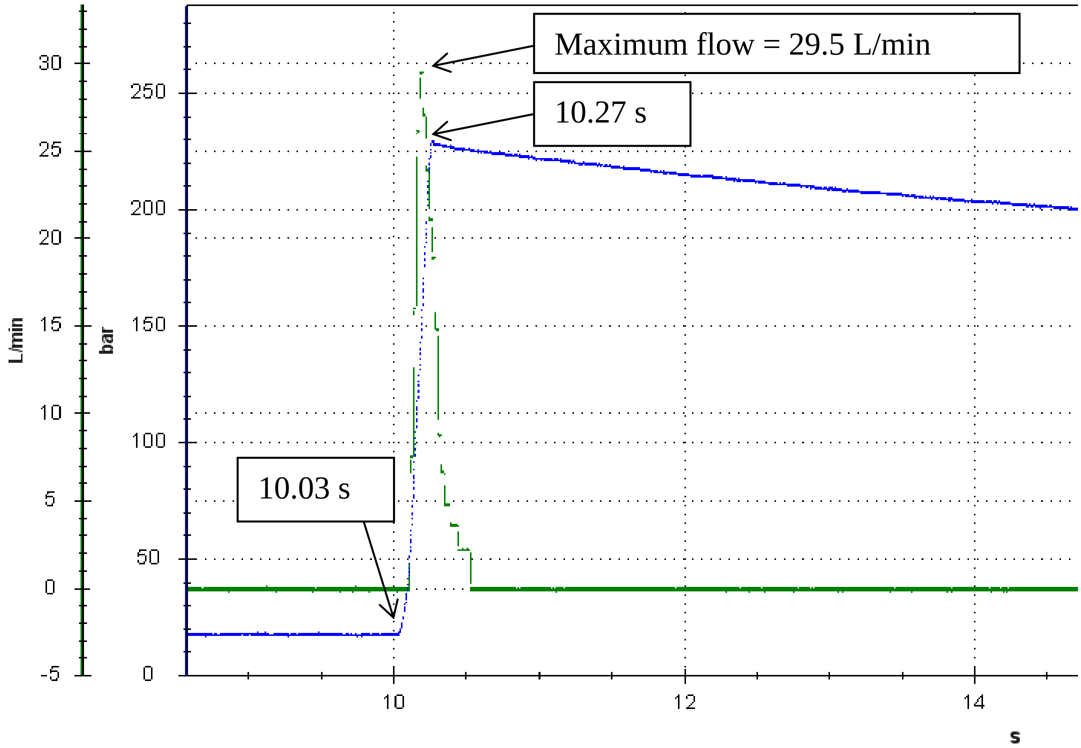


Figure 8.27: *The system is loaded up to 125 bar within 0.24 seconds, at maximum flow by 29.5 L/min. Graphical output obtained from own test measurement.*

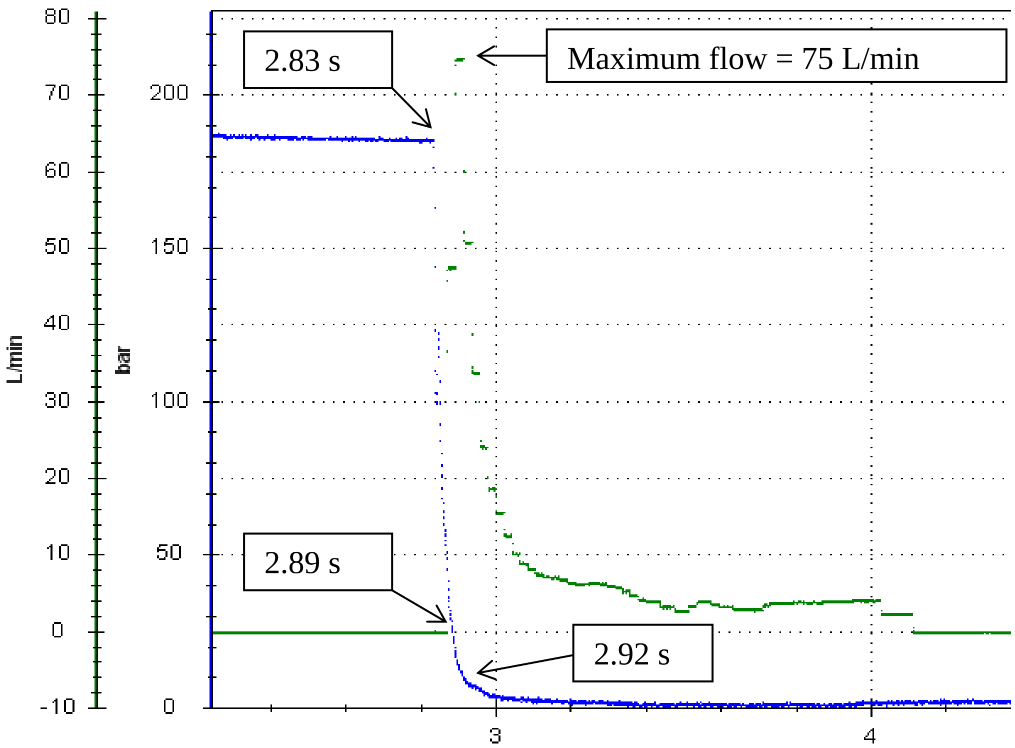


Figure 8.28: *The system is unloaded from 185 bar to 25 bar in just 0.06 seconds while it takes another 0.04 seconds to get down to 10 bar. Maximum flow at 75 L/min. Graphical output obtained from own test measurement.*

8.4.2 Test 2: Mechanical release characteristics

To obtain the mechanical release characteristics measured in practice, the plough point were pulled in horizontal direction by a cylinder. Between the cylinder and the plough point a load cell was connected which continuously measured the force-on-point throughout the release sequence. At the release cylinder the hydraulic pressure was measured to keep track of the system counterforce towards the applied force-on-point. The measurements were taken within a 44 second interval while slowly lifting the plough body by adding force to the plough point. The measured data from this test along with data on the comparison between the measured and calculated mechanical release characteristics can be obtained from Excel document “Verification of release characteristics”, referred to in appendix A1.1.

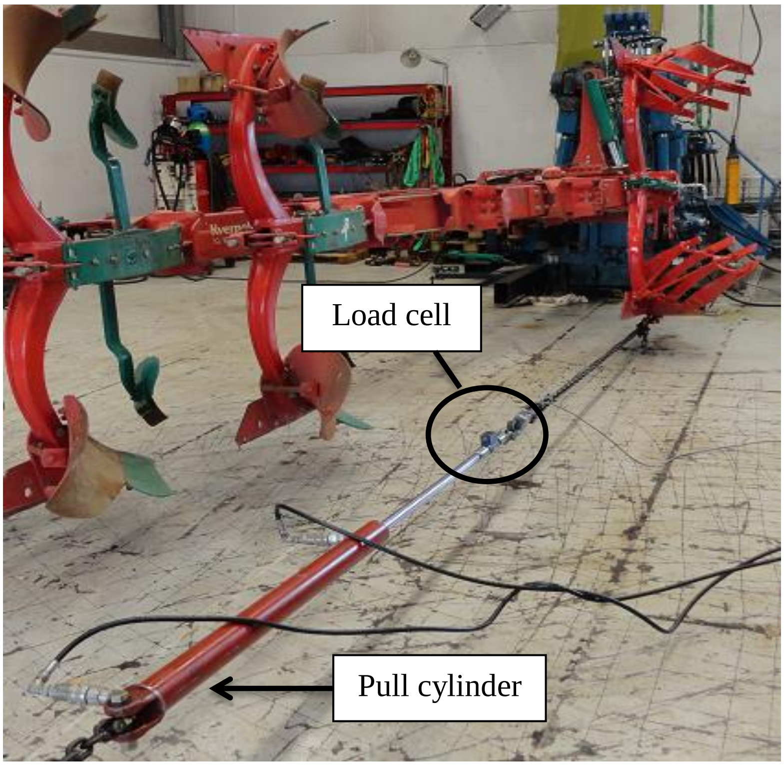


Figure 8.29: *The setup for measuring the force on point connecting a cylinder towards plough point, with load cell connected in between. The cylinder tube was connected towards a column fixed to the floor. Illustration by own photo.*

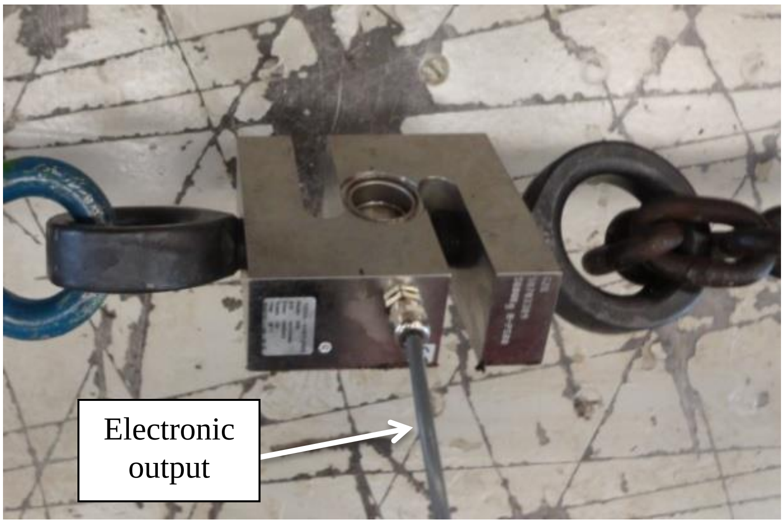


Figure 8.30: *Load cell illustrated in more detail. The measurements from the load cell were displayed in kilograms by the software handling the electronic output. Illustration by own photo.*

Figure 8.31 (left): *The plough body fully released, during test of mechanical release characteristics.*

Figure 8.32 (right): *The pressure measurement was accomplished placing the sensor on a T-pipe connection just outside the cylinder outlet.*

Illustrations by own photos.

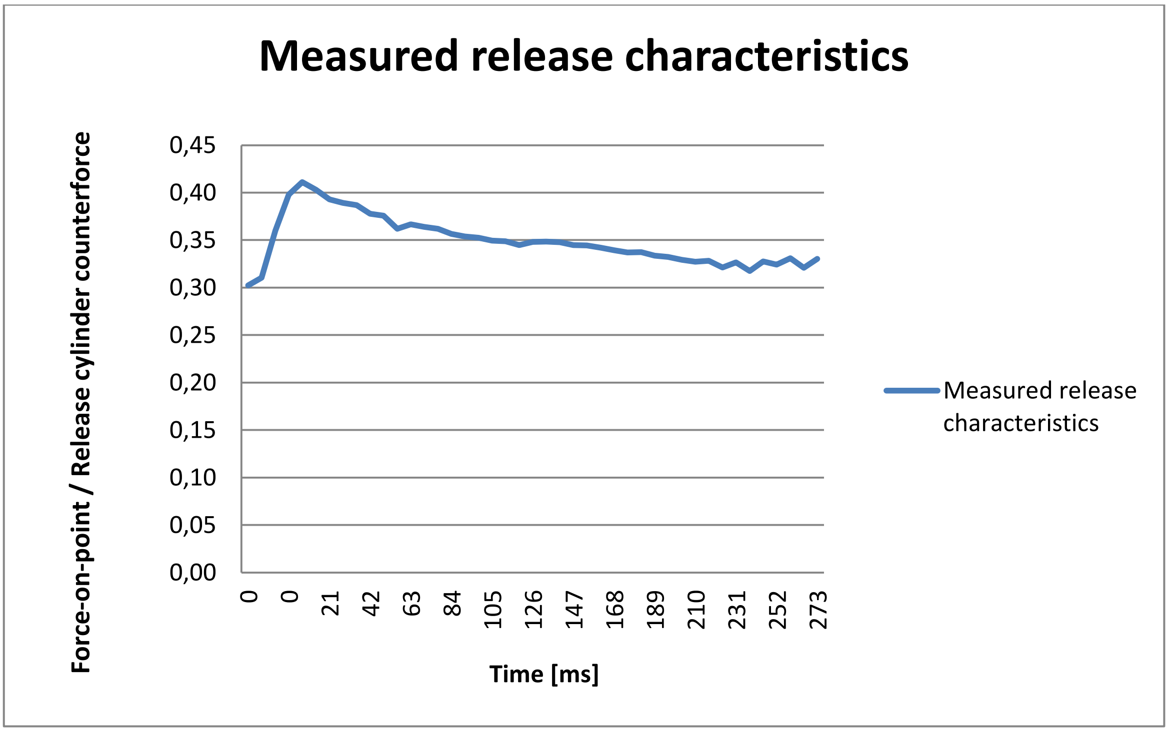


Figure 8.33: *The measured release characteristics are starting off with a relation on force-on-point over release cylinder force by 0.3 before increasing to 0.4 rather, rather rapidly. For the rest of the sequence it is gradually decreasing as well as the system resistance.*

The measured release characteristics are decreasing in terms of system resistance from about 30 ms and throughout the sequence, which is what we want it to do. On the other hand the increase in the beginning of the sequence from 0.3 to 0.4 is a bit more worrying when just looking at this graphics. A smaller increase can be accepted, but an increase by 30 % of the initial resistance is a bit too much. However, the data in the start of the sequence should be handled with caution, since there might be uncertainties to the first measured values until the force-on-point is ready to overcome the system resistance and the plough body has actually started to lift. To assure the quality of the measured characteristics and to obtain a more precise reasoning behind it a further look into the direct measurements has been performed.

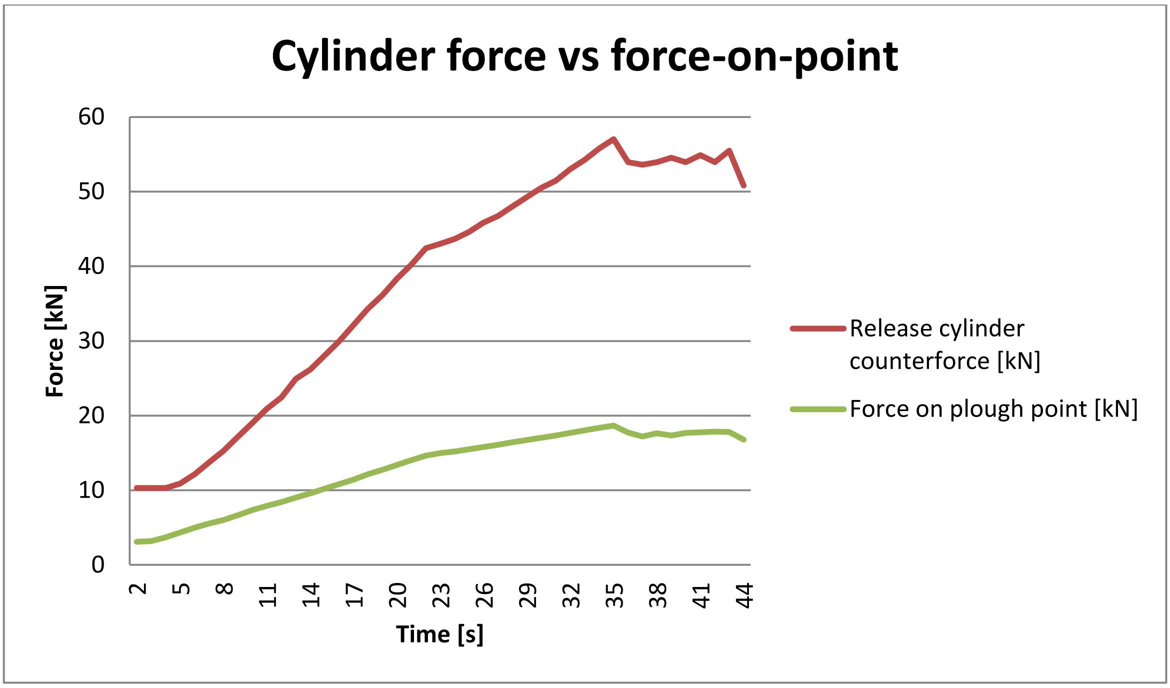


Figure 8.34: *As mentioned the relation between force-on-point and the release cylinder counterforce obtained in this figure define the final characteristics.*

Figures 8.33 and 8.34 are based on own test measurements data, further processed in own Excel document: "Verification of release characteristics.xlsx", which can be found in compact disc accompanying this report.

As seen from Figure 8.34, the increase in cylinder counterforce is lagging behind the increase in force on point for the first 5 seconds of the measurement, which indicates that the cylinder has not yet started to travel.

There might be several reasons for this, where the three most obvious ones are that:

1. The force-on-point has not yet reached the necessary force to overcome the system counterforce when starting the measurement, which means that the system is actually unbalanced in profit of the cylinder counterforce.
2. High initial system pressure and resistance is allowing mechanical parts to elastically deform and therefore absorb energy instead of the hydraulic system at very beginning of the sequence.

3. Low initial system pressure might cause that possible cylinder displacement is not able to compress the oil enough to cause any significant pressure increase in the beginning of the sequence.
4. There are loosely fit connections between mechanical parts which delay the force distribution through the mechanical system from plough point to release cylinder.
5. Tightly fitted connection causing high static friction between mechanical parts in the system, before reaching actual release force.
6. The plough point tracks horizontally or even a bit downwards at the start of the sequence, creating severe change to moment arm relation in start of sequence.

The possibility that the force-on-point has not reached the force of resistance from the system can very possibly be the case, since the system is loaded before force is added onto the plough point. However it is not the only possible reason and the other possible reasons need to be discussed as well.

The second reason suggesting high initial system resistance can most likely be ruled out, since the force on point is rather low or at least moderate with 3 kN at the beginning of the measurement. That force should not be enough for the plough beam to absorb much energy until the also rather low system pressure of 33 bar and cylinder counterforce give after. On the other hand the low pressure setting might make it possible for the cylinder to travel some millimeters without much pressure increase of the oil in the release cylinder. However, increase in plough point force should be proportional to system pressure increase in order to obey the law of conservation of energy, as long as possible energy losses are neglected. Therefore point 2 and 3 can most likely be ruled out.

When it comes to the fitting of the connections, the mechanical system is not yet fine-tuned at this stage for the first prototype model and uncertainties regarding the amount of static friction in the joints between tie rod, tie plate, cylinder fixes and beam and their interference with the measurements in the start of the sequence is difficult to predict without any further testing. If the fittings are to loosely fit this could cause a kind of delay in measurements, but since the system is pre-tensioned to a certain extent, loaded very gradually and the applied load is practically one-directional for the whole measured sequence, no sudden change in measured system resistance should occur because of possible loose fittings.

When it comes to the possible horizontal or even downward travel of the plough point at the start of the sequence this should not cause a resistance increase by as much as 30% and can be ruled out.

After looking at possible reasons for the sudden increase in characteristics at the beginning of the measurement, the most obvious reason is that it is mainly a disturbance in the measurement caused by the fact that the measured force-on-point is not the actual force equivalent to the system resistance when starting the measurement. The only other possible explanation would be if the system pressure was higher and the plough body was allowed to absorb a great deal of energy. Static friction could play a role, as mentioned, but neither that or internal material friction should in this case cause a sudden 30% increase in resistance.

By suggesting measurement error in the first part of the measurement, that part of measurement is chosen neglected when comparing to the calculated characteristics.

Furthermore, when comparing the calculated and measured release characteristics in Figure 8.35, the measured characteristics imply about 30% higher resistance than the calculation. However the decrease in the characteristics made out of the moment arm relations is pretty much the same and seems reasonable. The most obvious reason why the measured characteristics imply a higher resistance is the fact that the calculation was performed in terms of neglecting both external and internal friction in and between mechanical parts. Additionally the weight of the plough body has been neglected, as previously mentioned. So an increase in release resistance should be expected and a 30% increase caused by the weight of the plough body and friction forces is not unreasonable.

The newly measured characteristics imply a change in either moment arm relation geometry or a decrease in system pressure or cylinder diameter. In that manner the actual characteristics can be lowered down to the calculated characteristics in order to maintain the desired variable release resistance in the system between 10 to 16 kN. The most advantageous solution in order to maintain initial system resistance is to decrease the cylinder size in terms of piston diameter. This way, we also get to decrease the amount of oil that needs to be evacuated and once more added to the system for each release sequence. The recalculation of required cylinder piston diameter in accordance with the measured release characteristics are included into the chapter regarding product design revision, which is following the results and findings in this functional test. From the relevant metric limit specification, the reduction in system release resistance should not be more than 30 % for a complete release resistance. The measured characteristics only imply a reduction by 21 % in release resistance, which is fully acceptable.

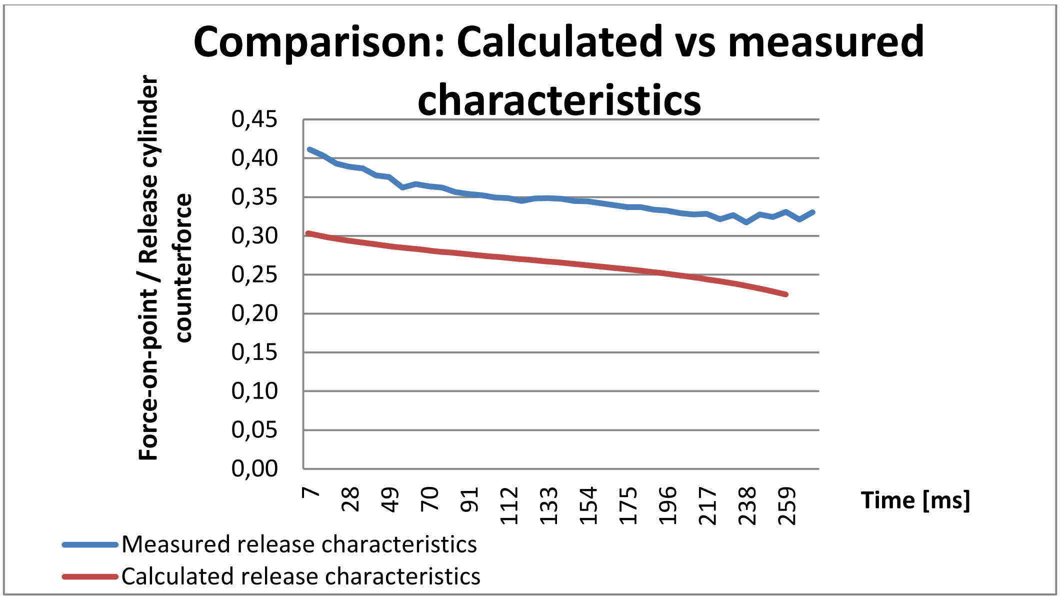


Figure 8.35: *Comparison between calculated and measured release characteristics. Based on own test measurements data, further processed in own Excel document: "Verification of release characteristics.xlsx".*

8.4.3 Test 3: Hydraulic system characteristics

The main objective of this test was to find out whether the hydraulic system has the required capacities in order to keep up with the system requirements. The system was tested with regards to the ability to keep constant pressure during the first phase of the release sequence (release upwards). At the same time flow and temperature were measured in order to keep track of the system conditions.

Plough point pull

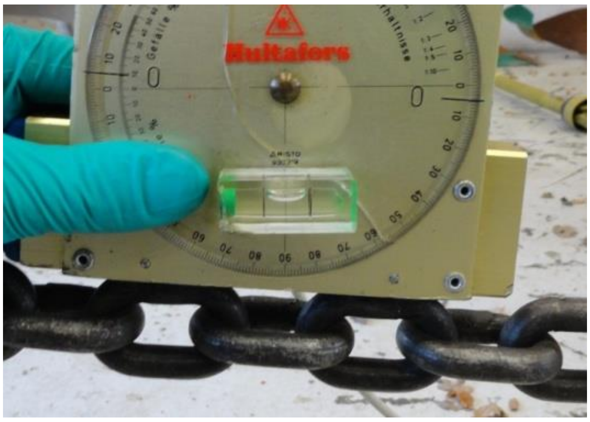


Figure 8.36 (left): *Cylinder pressure during test was 180 bar, equivalent to 17 kN of force-on-point during the release sequence.*

Figure 8.37 (right): *The pull chain was fixed at the end with no possibility of vertical displacement, so there had to be a certain angle relative to the horizontal plane. As shown in this illustration the plough point was pulled in 5 degree angle at the start of the sequence where it decreased towards zero when travelling towards full lifting height. The chain was made as long as possible just in order to make the pull angle as small as possible. Since the angle in pull was not larger than 5 degrees it was found negligible with regards to the measurements.*

Illustrations by own photos.

Figure 8.38 *To pull the plough point, two hydraulic cylinders with dimensions D40-d20 was connected with chains to it. The cylinders were chosen with a cross-sectional area as small as possible in order to get as much velocity on the plough point as possible out of the oil delivery from tractor rig. In order to get sufficient stroke length two cylinders were connected in series with a total stroke length by 650 mm.*

Illustration by own photo.

Figure 8.39: To keep track of the flow delivered out to the pull cylinder, a flow measurement was performed in order to predict the plough point horizontal velocity.

Illustration by own photo.

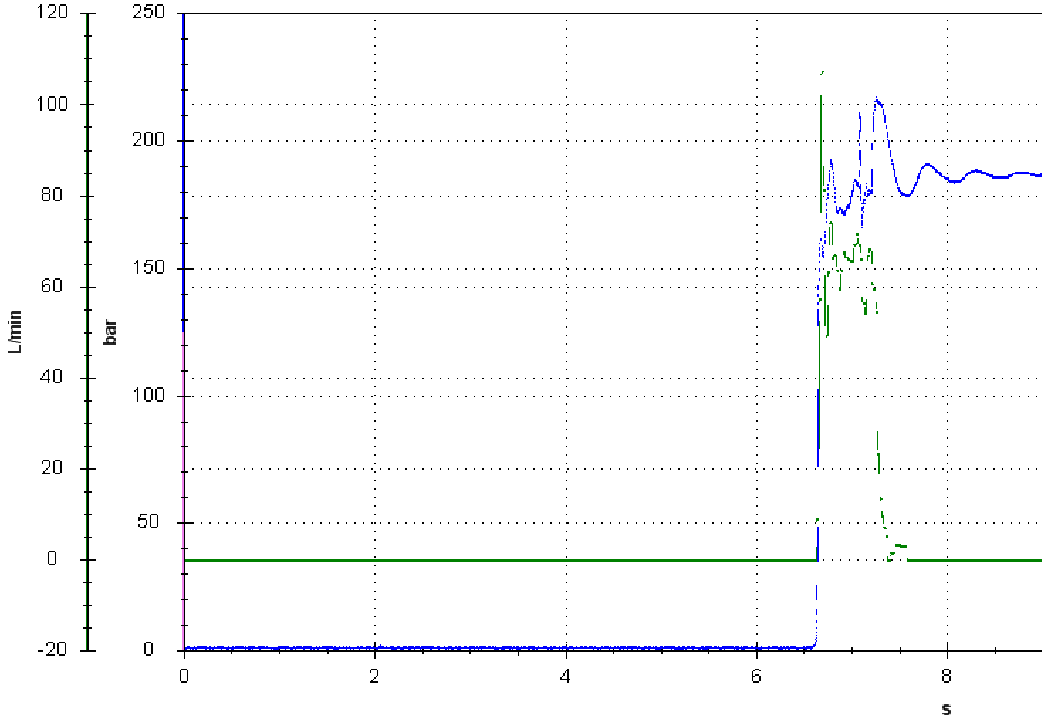


***Figure 8.40:** The flow peaks at 107 L/min before it evens out at around 65 L/min for the rest of the release sequence. The pressure at the tractor rig outlet was at around 180 bar, just like the pressure measured out on the cylinder. The counter pressure from the hydraulic stone release system was at 50 bar for this measurement, so for higher system pressures the flow and plough point velocity should be expected to decrease due to the higher resistance when pulling the plough point. Graphical output obtained from own test measurement.*

Calculating the maximum and average plough point velocity from flow measurement in Figure 8.40:

$$v_{point} = \frac{Q_{tractor}}{A_{cylinder}} \quad (Eq\ 9)$$

$$Q_{max} = 107\frac{L}{min} = 0.00178\ \frac{m^3}{s}$$

$$Q_{avg} = 65\frac{L}{min} = 0.00108\ \frac{m^3}{s}$$

$$A = \frac{\pi * (40^2 - 20^2)}{4} = 943\ mm^2 = 0.000943\ m^2 \quad (Reworked\ from\ Eq\ 3)$$

$$v_{point,max} = \frac{0.00178\ {m^3}/{s}}{0.000943\ m^2} = 1.89\ \frac{m}{s} = 6.8\ \frac{km}{h} \quad (Eq\ 9)$$

$$v_{point,avg} = \frac{0.00108\ {m^3}/{s}}{0.000943\ m^2} = 1.15\ \frac{m}{s} = 4.1\ \frac{km}{h} \quad (Eq\ 9)$$

With the capacities measured we will not be able to fully simulate a driving velocity by 8 km/h for this test, but a maximum velocity by 6.8 km/h is at least relatively close to that target. However, this measurement for the plough point pull is carried out with a counter pressure in the release system by only 50 bar. When increasing the system pressure we would expect the plough point travel velocity to decrease, as previously mentioned. This signifies that the total energy that the system has to absorb is pretty much lower than for the calculations where assuming even 170 bar by 8 km/h of plough point travel velocity. This fact has to be taken into consideration when handling and evaluating the test data.

However, the fact that the plough body is releasing with a plough point horizontal velocity of 8 km/h is a rather strict assumption in itself. Taken into consideration that most stones in the soil are more or less round in shape, the horizontal plough point velocity will tend to drop of following the round contour of the stone. If dropping off at 45 degrees, hitting a stone in ploughing velocity of 8 km/h imply a plough point velocity by 4 km/h in horizontal direction. However, it is right to assume oil flows in the hydraulic system equivalent to one plough body releasing with horizontal velocity of 8 km/h since we need to take consideration of the possibility that more than one-, or at least 2-3 plough bodies release at the same time, inducing the equivalent high oil flows in the system, at pressures up to or even above 160 bar.

Regardless of the fact that the applied plough point velocity is a bit lower than assumed for the calculations, it should be absolutely feasible to record measurement data that will tell much about how well the system respond in accordance with the desired system functionality and capacities. Additionally, as a one plough body test as this is, the plough point velocity ranging in between 4-7 km/h is fully within what we could expect in a real scenario in field for a single plough body isolated, with reference to the short discussion just performed in previous section. To fully get an answer how the system works it has to be tested with several plough bodies and out in field. But that lies further ahead in the overall project where further functionality needs to be implemented into the system before doing a test on a final prototype.

When it comes to the applied load on the plough point versus the load on the release cylinder, we will not be able to release the plough body with a system counter pressure up to 160-170 bar when applying 17 kN on the plough point due to the recently obtained release characteristics. The maximum system pressure recalculated for this test is:

$$F_{cyl,max} = \frac{17\ kN}{0.41} = 41.5\ kN \quad (Eq\ 1)$$

$$A_{cyl} = \frac{\pi * 63^2}{4} = 3117\ mm^2 \quad (Reworked\ from\ Eq\ 3)$$

$$p_{cyl,max} = \frac{41\ 500\ N}{3117\ mm^2} = 13.3\ MPa = 133\ bar \quad (Reworked\ from\ Eq\ 2)$$

Since the maximum system pressure now is below the pre-charge pressure of the accumulator, it is left out of the system, where now only the pressure relief valve alone should absorb energy from the system by evacuating oil. As also mentioned for test 1 it is more ideal to leave out the accumulator, in order to obtain the actual capacity and behavior of the relief valve without having the accumulator disturbing the measurements.

Points of measurement and release sequence

The measurements for this test is split into two parts, with one set of measurements at the release cylinder outlet and one set of measurements closer to the relief valve.

Figure 8.41 (left): *The plough body fully released at the end of the first lift phase of the release sequence subject to this test*

Figure 8.42 (right): *Pressure, oil flow and –temperature were measured on the cylinder outlet to keep track of the flow out of the cylinder in addition to the actual pressure acted upon the system from the cylinder during the sequence. Temperature is measured to keep track of the overall temperature conditions in the system when doing the measurements.*

Illustrations by own photos.

Figure 8.43: *Relocation of pressure-, flow and oil temperature measurement to obtain conditions by pressure relief valve for the second part of the measurements regarding phase 1 (plough body lift) of the release sequence. Note that the accumulator is removed.*

Illustration by own photo.

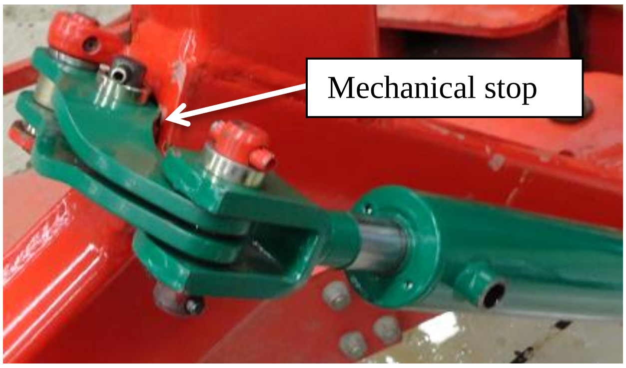


Figure 8.44: *The plough body is fully released when the tie plates hit the rear part of the beam, by mechanical stop, as illustrated. The lift height is measured to 396 mm, which is well above the minimum requirement by 350 mm stated in the metric limit specification.*

Illustration by own photo.

The measurements have been carried out adjusting the mechanical adjustable relief valve at certain burst pressures governing the preset pressure from 40-90 bar for the test. When loading the system up to 130 bar, the load on the plough point is not able to enforce the plough point to move, so to get measureable data with a certain amount of plough point- and release cylinder travel velocity, it was decided to do the upper pressure measurement at around 90 bar and the lower pressure measurement at around 40 bar. Despite the fact that the 40 bar pressure measurement is way below the minimum pressure that the system should keep, it is valuable in the way that we get to compare differences in system behavior over a wider range of pressures.

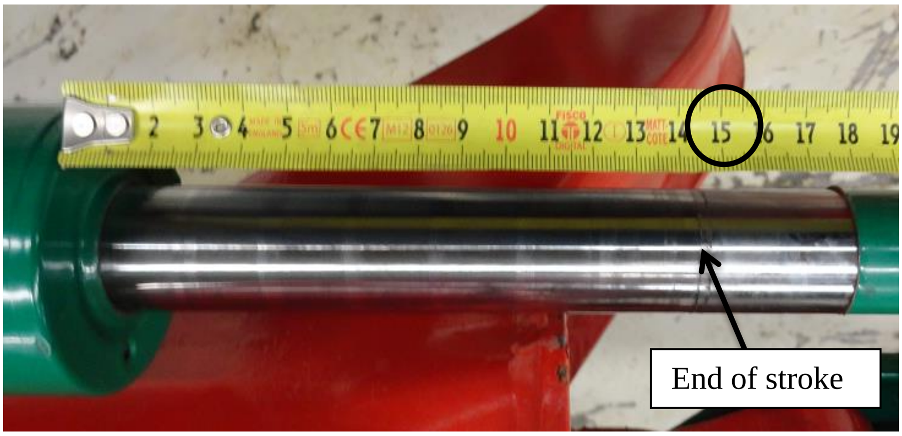


Figure 8.45: *The stroke length of the cylinder for the release sequence by 396 mm lifting height was measured to 150 mm (see marks of end of stroke on cylinder rod from illustration). Illustration by own photo.*

Measurements by cylinder

Table 8.5: *Measuring outputs in colour code according to graphical outputs. This table refers to type of output, measurement location and measurement unit displayed.*

Colour code on graph	Type of output and location	Measurement unit
Blue	Cylinder pressure	bar
Green	Oil flow, cylinder outlet	L/min

For better readability the measured temperature is left out of the graphical output for these measurements. But for all measurements oil temperature was consistently at around 20° C (293 K).

<u>88 bar measurement</u>

From the product specification the system pressure is allowed to deviate by 10% in each direction of the preset pressure. With the 88 bar pressure setting, we get the following allowable pressure range:

$$p_{range} = \frac{88\ bar}{10} \approx \pm 9\ bar$$
$$p_{max,allowed} = 88 + 9 = 97\ bar$$
$$p_{min,allowed} = 88 - 9 = 79\ bar$$

Table 8.6: *Table output of critical measurement data from 88 bar measurement at release cylinder location from Figure 8.46.*

Time of measurement (s)	Type of event	Measurement
3.23	Pressure starts increasing – start of sequence	88 bar
3.30	Pressure peak	135 bar
3.32	Oil flow from cylinder starts	0 L/min
3.42	Oil flow peak	42 L/min
4.2	Oil flow from cylinder stop	0 L/min

Total time of release sequence:

$$t_{sequence} = 4.2\ s - 3.23\ s = 0.97\ s$$

Obtain average cylinder velocity from sequence time and measured cylinder displacement:

$$v_{avg} = \frac{0.15\ m}{0.97\ s} = 0.15\frac{m}{s} \quad (Reworked\ from\ Eq\ 7b)$$

Calculate average oil flow throughout sequence:

$$Q_{avg} = v_{avg} * A_{cyl} = 150\frac{mm}{s} * 3117\ mm^2 \quad (Reworked\ from\ Eq\ 9)$$

$$Q_{avg} = 467\ 550\ \frac{mm^3}{s} = 0.47\ \frac{L}{s} = 28.2\ \frac{L}{min}$$

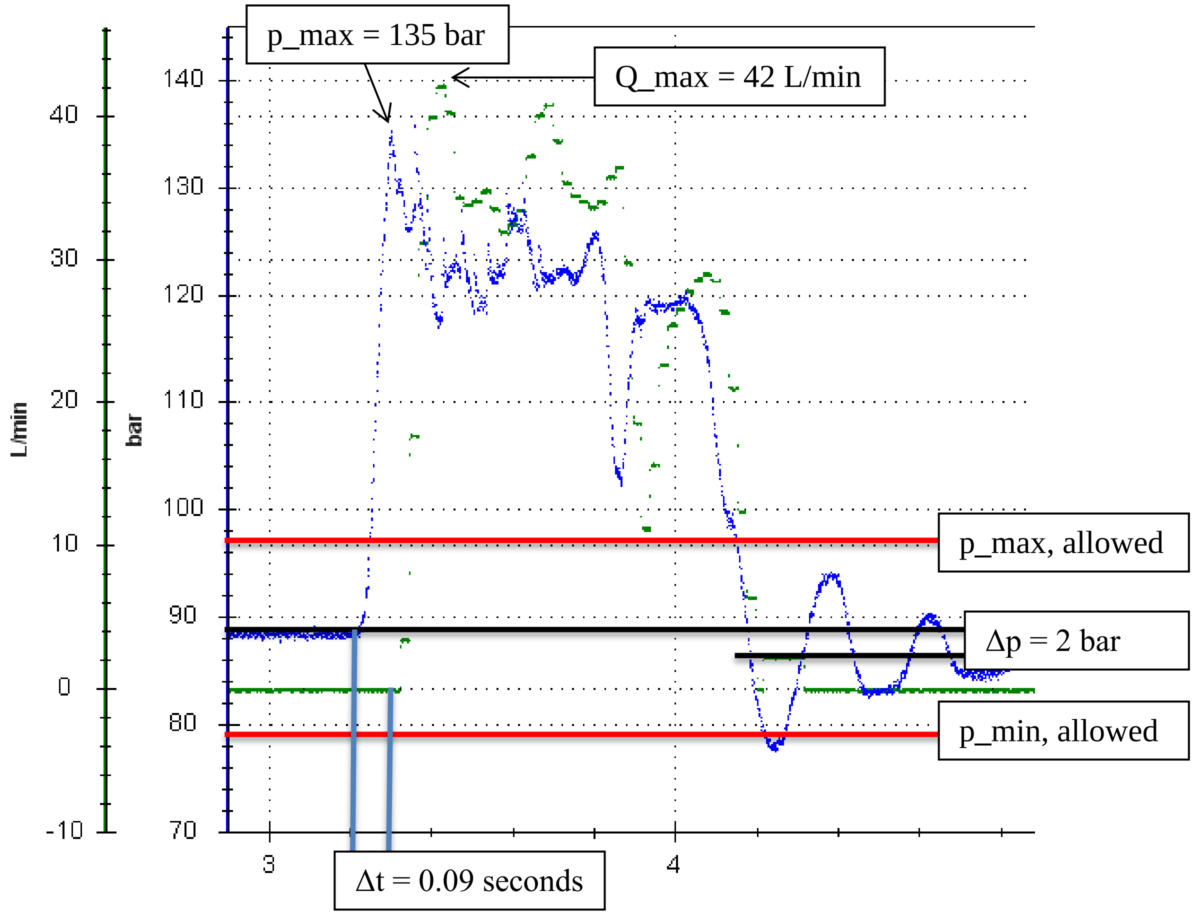


***Figure 8.46:** Hydraulic system release characteristics with system pressure pre-setting by 88 bar. From this illustration we read out a maximum pressure at 135 bar, a maximum flow by 42 L/min. The flow is delaying the pressure rise by 0.09 seconds. After the release sequence the system pressure stabilizes at 2 bar lower than the initial setting. The red lines indicate the allowable maximum and minimum pressure previously calculated. Graphical output obtained from own test measurement.*

The illustrated hydraulic release characteristics reveal that the pressure relief valve is seriously underperforming in terms of keeping pressure within 10 % of pre-set pressure. Most of the reason behind this seems to be due to delay in opening of the valve upon release. The system pressure increases from 88 bar and peaks at 135 bar even before the oil starts to leave the cylinder, as also obtained from data in Table 8.6. Along most of the sequence the oil flow seems to follow the pressure characteristics, however with a delay, which is initially 0.09 seconds, as illustrated. The oil flow doesn't catch up on the pressure until the sequence is over with. Since there seems to be a delay on the valve upon closing as well, the system pressure is 2 bar lower at the end of the sequence than before the release. The system pressure is also oscillating a bit after valve close, but within the allowable pressure range. These oscillations are most likely to come from minor shocks due to the sudden close of the valve and deceleration of the oil.

<u>46 bar measurement</u>

Just as performed for the previous measurement, we once more calculate the allowable pressure range:

$$p_{range} = \frac{46\ bar}{10} \approx \pm 5\ bar$$

$$p_{max,allowed} = 46 + 5 = 51\ bar$$

$$p_{min,allowed} = 46 - 5 = 41\ bar$$

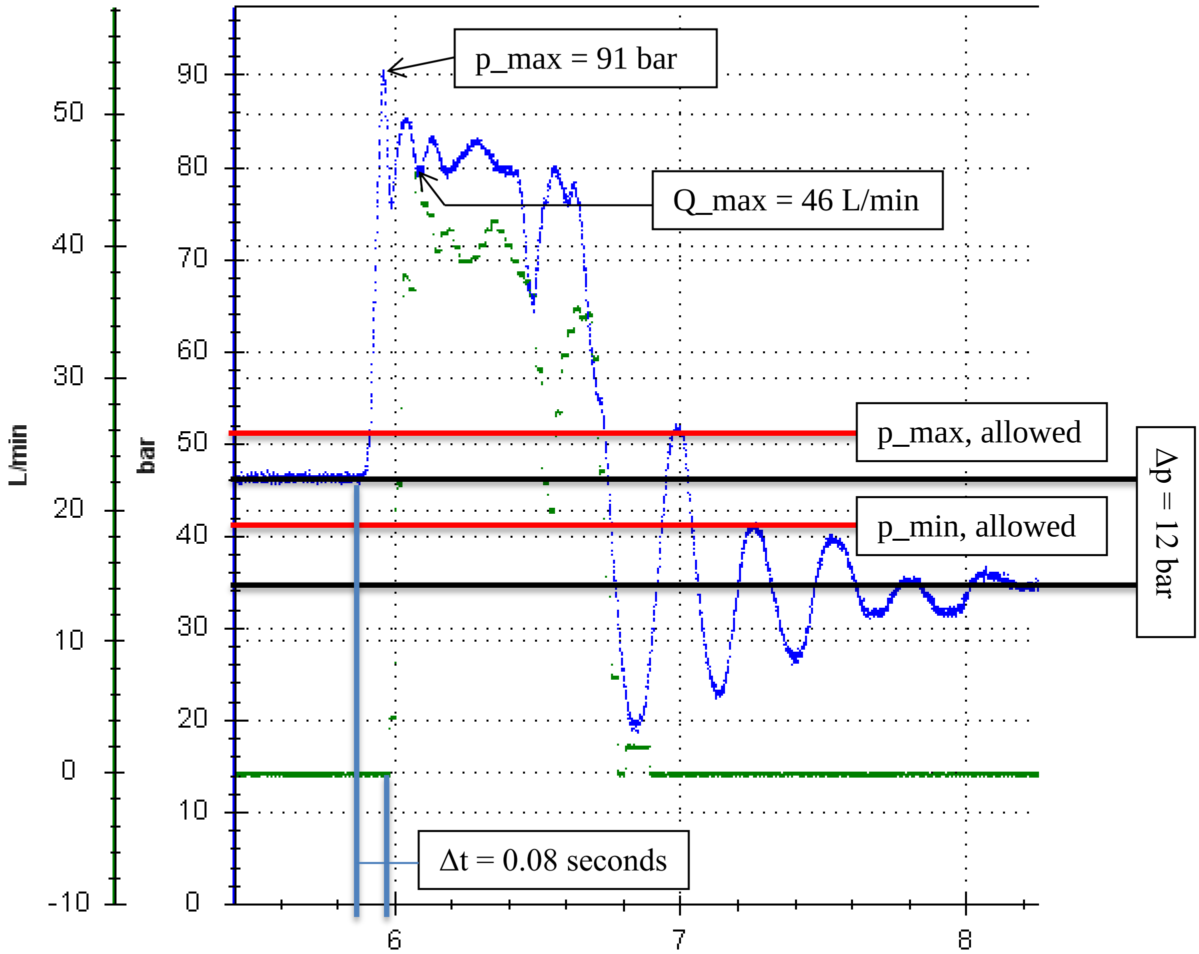


Figure 8.47: *Hydraulic system release characteristics with system pressure pre-setting by 46 bar. From this illustration we read out a maximum pressure at 91 bar, a maximum flow by 46 L/min. The flow is delaying the pressure increase by 0.08 seconds. After the release sequence the system pressure stabilizes at as much as 12 bar lower than the initial setting. The red lines indicate the allowable maximum and minimum pressure previously calculated. Graphical output obtained from own test measurement.*

Total time of release sequence:

$$t_{sequence} = 6.78\ s - 5.90\ s = 0.88\ s$$

Obtain average cylinder velocity from sequence time and measured cylinder displacement:

$$v_{avg} = \frac{0.15\ m}{0.88\ s} = 0.17\ \frac{m}{s} \quad (Reworked\ from\ Eq\ 7b)$$

Calculate average oil flow throughout sequence:

$$Q_{avg} = v_{avg} * A_{cyl} = 170 \frac{mm}{s} * 3117\ mm^2 \quad (Reworked\ from\ Eq\ 9)$$

$$Q_{avg} = 529\ 890\ \frac{mm^3}{s} = 0.53\ \frac{L}{s} = 31.8\ \frac{L}{min}$$

Table 8.7: *Table output of critical measurement data from 46 bar measurement in Figure 8.47 at release cylinder location.*

Time of measurement (s)	Type of event	Measurement
5.90	Pressure starts increasing – start of sequence	46 bar
5.96	Pressure peak	91 bar
5.98	Oil flow from cylinder starts	0 L/min
6.09	Oil flow peak	46 L/min
6.78	Oil flow from cylinder stop	0 L/min

The valve is still not close to being able to keep the system pressure constant within 10% allowable deviation in each direction. The drop in system pressure after the release sequence has come up on 12 bar compared to the initial set value and reaches below the minimum allowable pressure. However, that might be covered by adding oil back into the system through the directional valve at the end of the plough body lift, once implementing the automatic plough body retraction in the complete release sequence.

The pressure build up and the lack in response of the relief valve may mainly come out of three reasons or a combination of those; inertia in the hydraulic oil between cylinder and relief valve, restrictions along the pipeline causing pressure loss or simply that the delay is directly related to response time of the pressure relief valve. In the further measurement review we shall try to rule out possible causes to reveal whether it is the pressure relief valve that is actually causing the delay and subsequent pressure increase in the start of the sequence. In the next measurement the pressure is measured at two locations along the pipeline, to reveal potential system inertia and pressure loss between cylinder and valve.

Simultaneous measurement by cylinder and valve

Table 8.8: *Overview of system states related to behavior of components and events pinpointed along measurement output in Figure 8.48.*

Type of event	Pressure- and flow characteristics	System state
Start of measurement	Constant pressure, zero flow	Static
Cylinder travel starts	Pressure increase, zero flow	Quasi static
Valve opens	Pressure relief, flow increase	Dynamic
Mechanical stop – cylinder standstill	Pressure relief, flow decrease	Dynamic
Valve closes	Pressure oscillation, zero flow	Quasi static

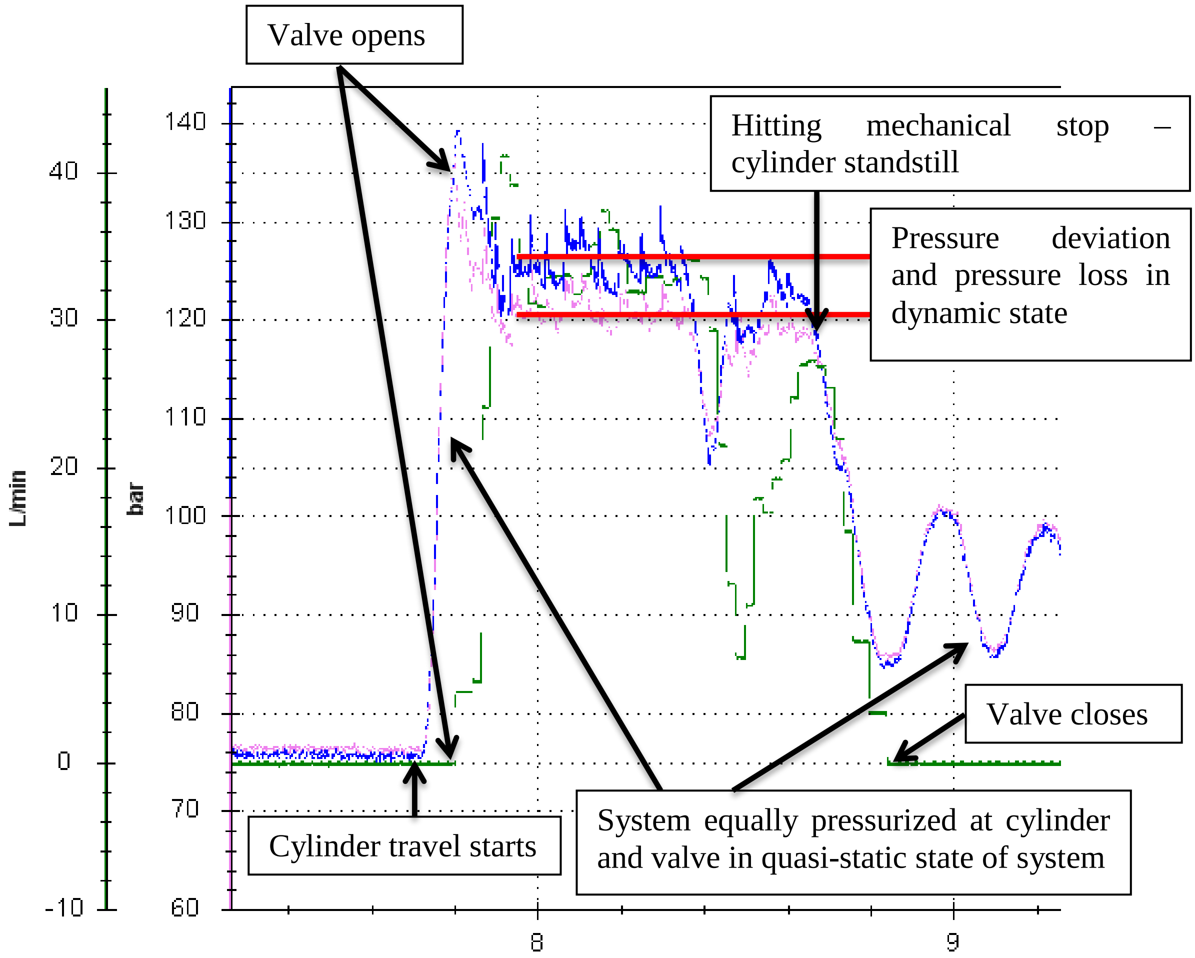


Figure 8.48: *This graphical output at 76 bar pre-set pressure has the pressure measured at both cylinder and closer to the valve. Pressure measurement by cylinder is still in blue, flow measurement by cylinder in green, while pressure closer to valve is indicated by the purple coloured graph. As seen from this illustration, the pressure is somehow a bit lower closer to the valve during the release sequence itself, while the system is in the dynamic state. Graphical output obtained from own test measurement.*

The graphical output in Figure 8.48 reveals a difference in pressure measured at cylinder and by valve in the dynamic state of the hydraulic system, when oil is being evacuated. This pressure difference between the two locations might be caused by a pressure loss along the pipeline, in terms of an energy transition from mechanical- to thermal energy. This was also predicted in calculations performed in chapter 7 under "7.2.4 – Fluid dynamics", both with hand calculation and by computational fluid dynamic analysis. The pressure drop in this case can be approximated to about 6 bar, but at an average flow rate at no more than 30 l/min for most of the measurement. By the fluid dynamic calculations in chapter 7, the pressure drop along the similar pipeline was calculated to 10 bar by hand calculation and 8 bar by CFD analysis, but at a flow rate at 112 L/min.

Calculate theoretic pressure loss in this case with flow at 30 L/min with formulas once more obtained from (Brautaset 1983):

$$\Delta p_{loss} = \lambda * \frac{l}{d} * \rho(rho) * \frac{v_m^2}{2} \quad (Reworked\ from\ Eq\ 11)$$

Calculate fluid friction coefficient in terms of recalculated Reynolds number for present flow:

$$\lambda = \frac{0.316}{Re^{0.25}} \quad (Eq\ 13)$$

(Calculate Reynolds number with a flow rate at Q=30 L/min, by equation obtained from (Kjølle 1995)). First, we convert the flow:

$$Q = 30\frac{L}{min} = 0{,}5 * 10^{6}\left(\frac{mm^{3}}{s}\right)$$

Calculate Reynolds number:

$$Re = \frac{Q * d}{\vartheta * A} = \frac{0.5 * 10^{6}(mm^{3}/s) * 13\ mm}{46\frac{mm^{2}}{s} * (\frac{\pi * (13\ mm)^{2}}{4})} = 1065 \quad (Eq\ 8)$$

$$\lambda = \frac{0{,}316}{1065^{0.25}} = 0.055 \quad (Eq\ 13)$$

Calculate the fluid velocity from measured flow:

$$v_{m} = \frac{Q}{A} = \frac{0.5 * 10^{-3}\left(\frac{m^{3}}{s}\right)}{\left(\frac{\pi * (0.013\ m)^{2}}{4}\right)} = 3.77\frac{m}{s} \quad (Eq\ 9)$$

$$\Delta p_{loss} = 0.055 * \frac{4\ m}{0.013\ m} * 865\frac{kg}{m^{3}} * \frac{\left(3.77\frac{m}{s}\right)^{2}}{2} = 104\ 028\ Pa = 1\ bar$$

In terms of the calculations just performed, all of the measured pressure drop along pipeline from measurement in Figure 8.48 cannot be explained by mechanical energy loss to thermal energy for the case in this measurement, with a flow rate at around 30 l/min.

So a further explanation of the pressure deviation just illustrated, could be that we are dealing with system inertia in the dynamic state of the system. The oil closer to the valve is accelerated just a bit earlier in the sequence due to its location closer to the system outlet. That causes the oil flow at the cylinder to delay the flow closer to the valve, and as a subsequent result, the pressure relief at the cylinder is delaying the pressure relief at the valve. This theory can moreover be supported by the fact that the pressures at both measured locations are equal in quasi-static state, when relief valve is still closed, while the system gets loaded by the displacement of the cylinder. Once more, the cylinder pressure is catching up on the pressure measured closer to the valve when the system goes back to the quasi-static state upon cylinder standstill and subsequent closure of the pressure relief valve.

However, neither inertia nor pressure loss in the dynamic state explains the delay in valve reaction until its opened, since the system remains in the quasi-static state as long as the valve is closed. System inertia or pressure loss doesn’t tend to be present for quasi-static load cases in hydraulics, simply by the fact that it is no flow worth mentioning in the system, it becomes more and more clear that the valve is causing the delay in system response and the subsequent pressure rise in the beginning of the sequence.

Measurements by valve

In Figure 8.49 below, the fact that there is a delay in the response time of the valve relative to the pressure exerted on it is pinpointed in the graphical measurement outputs. In this measurement both the oil flow and pressure is measured at the same location by the relief valve. By performing the measurement so close to the valve itself, all uncertainties related to how the pipeline system interfere the system behavior can now be left aside. Therefore, it should now be relatively safe to say that the pressure relief valve is causing that the oil flow is delaying the pressure increase.

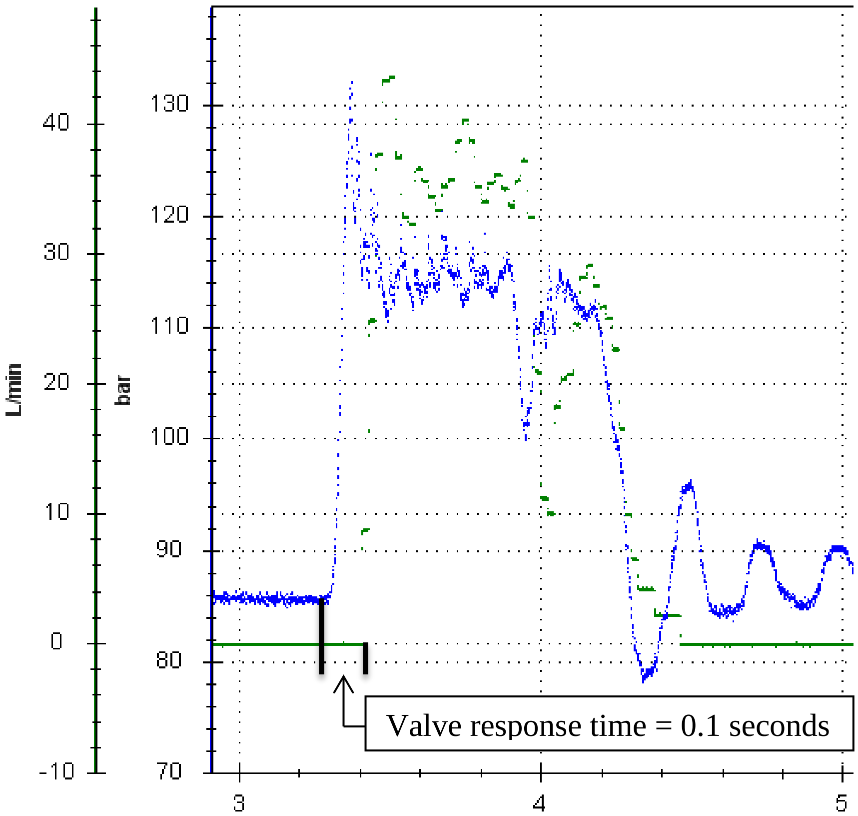


Figure 8.49: *Pressure-/flow measurement by the valve at 86 bar pre-set pressure. Graphical output obtained from own test measurement.*

The interpretation of the test results of this final test on the hydraulic system imply that the response time of the pressure relief valve is too long, creating high system pressure upon release which isn't fully relieved until the end of the plough body lift. On the other hand the ability to drain oil should be more than sufficient in terms of the rated capacity of the valve. So in that terms the valve still seems to do what it is intended to do, as soon as it starts draining oil.

As the system is designed at this stage it does however not fully comply with the system requirement in terms of pressure stability requirement within 10% of system pressure, stated by the relevant metric limit specification. At the most the system pressure increases by the following in the previous case with preset system pressure by 88 bar:

$$Pressure\ deviation = \frac{132\ bar}{\frac{88\ bar}{100}} - 100 = 50\ \%$$

The pressure deviation is calculated to 50 % at the most from the measurement readings. However, this varies a bit depending on the initial pressure setting for each measurement.

In the upcoming chapter, a proposal on system re-design and re-test will be outlined in order to improve system performance and keeping system pressure deviation within 10 %.

8.4.4 Test by complete release sequence

As already specified, the test at this stage only comprises the first part of the release sequence with regards to plough body lift. On basis of that fact, the functionality of the system electronics at this stage is mainly arranged for the first part of the release sequence as well, since there is no automatic retraction of the plough body in order to implement the second stage of the complete release sequence at this point.

However, it has been made possible simulating a complete release sequence, by operating the electronic control system manually. A brief description of how the test is carried out and how the electronics and the plough point pull-cylinder is operated in order to simulate a full release sequence, can be found in the following tables along with the graphical outputs.

Implementing the automatic retraction of the plough body in the second phase of the sequence imply further development of the electronic control system. The control system needs to make the directional valve direct oil back to the system in time before the obstacle causing the release, is passed. The intention at this point is to simulate a complete release sequence in order to predict how the system behaves in the second phase of the release sequence in order to implement the electronics at a later point.

Test 1

In this test, the first phase of the release sequence is carried out as usual, with the pressure relief valve draining oil until reaching point of full release height. The force is kept constant at 17 kN on the plough point for a while, simulating that the obstacle is yet not passed. Then the directional valve is operated in order to add oil into the system for the second phase of the release sequence. This is followed by releasing the force-on-point which results in retraction of the plough body.

Table 8.9: *Critical events throughout simulated release sequence – test 1.*

Event	Measurement time (s)	Type of event
1	17.92	Activation of plough point pull cylinder - start of sequence, phase 1
2	18.80	Point of reaching mechanical stop and full release height – end of sequence, phase 1
3	19.56	Directional valve is operated, adding oil into the system
4	20.16	Oil pressure on plough-point pull-cylinder is relieved – start of sequence, phase 2

Table 8.9 continued.

Q_max	20.36	Reaching maximum oil flow of oil added to cylinder (Q_max = 72 L/min)
5	20.93	Plough body reaches nominal ploughing position – end of sequence, phase 2
6	21.42	Directional valve closes (due to its pre-set opening time by 1.86 seconds)

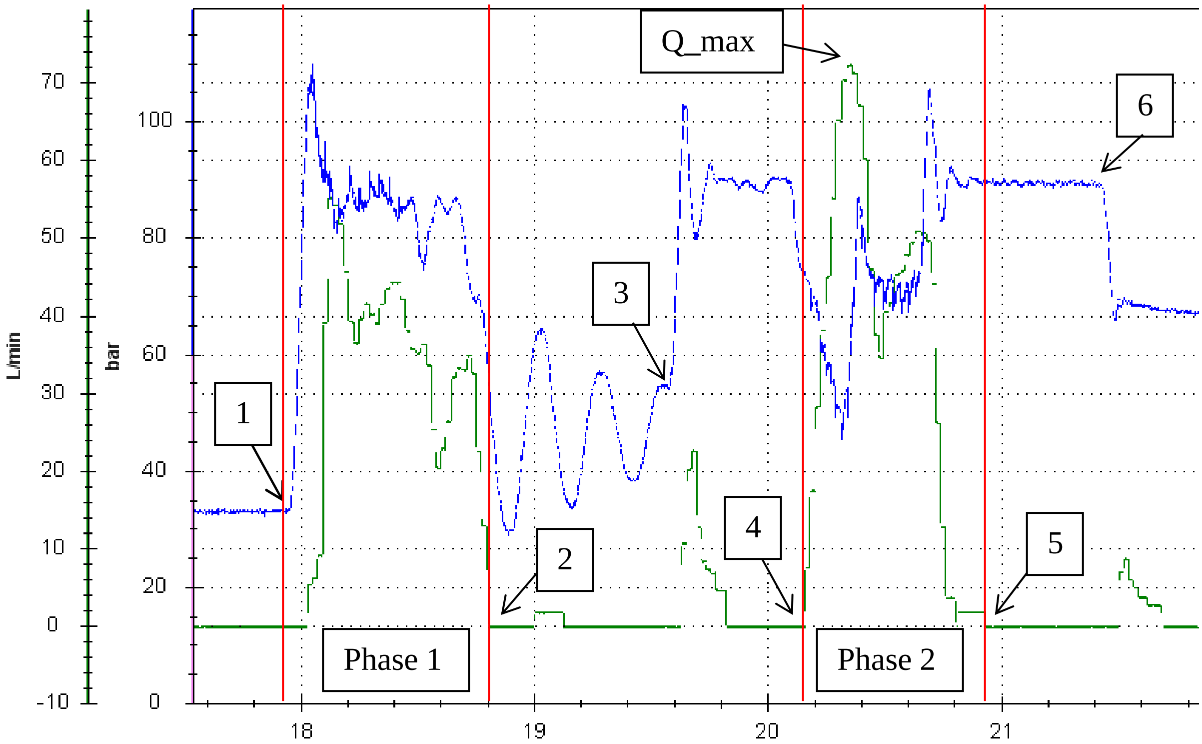


Figure 8.50: *The graphics shows how the system behaves during the complete release sequence for the given scenario. Descriptions along graphics can be obtained in Table 8.9. Graphical output obtained from own test measurement.*

During this test, oil is added into the system after the first phase of the release sequence, but before relieving the load on the plough point. This way we're assured that the system doesn't lose system pressure when the load on the plough point is relieved. In other words, if the load on the plough point is relieved without adding any oil to the system, the weight of the plough beam and –body make it travel towards ground, but as the cylinder is extended without adding any oil back to the system, the system pressure will reach zero in just a matter of milliseconds. That will make the plough beam fall off the plough as the final result. So in the next step of implementing the second phase of the release sequence, where it automatically follows the first phase, there are some challenges in order to be able to add enough oil back to the system - and at the right time. This is extra critical, especially if several plough bodies are released at the same time. That means they also have to be redirected back into the soil at about the same time, with a corresponding high demand for oil flow into the system. Since a release sequence can vary in time, depending on the size of the obstacle, the automatic control should be time independent, but rather depend on either pressure or displacement of the cylinder piston. This will be briefly discussed in the later section “Further system implementations” in chapter 10.

When adding oil back to the system, we can once more see that a pressure increase occurs in the system. This is most likely to come from the same reason as the when the pressure increases at the beginning of release sequence – phase 1; that there is a delay in the pressure relief valve. So once more there is a need of a gas accumulator at this point as well. Later when the plough body starts retracting, we get a problem with a pressure decrease even though oil is added to the system at a maximum flow rate by 72 L/min. An accumulator might also be able to assist the system at this point serving as an additional energy source adding oil back to the release cylinder. The system does not go back to initial preset pressure until the directional valve goes back to neutral position (at point 6 in Figure 8.50).

Test 2

Once more a force by 17 kN is added onto the plough point throughout the first phase of the release sequence, until unloaded. However, in this test oil is added back into the system just upon the moment of release in phase 1 of release sequence

.

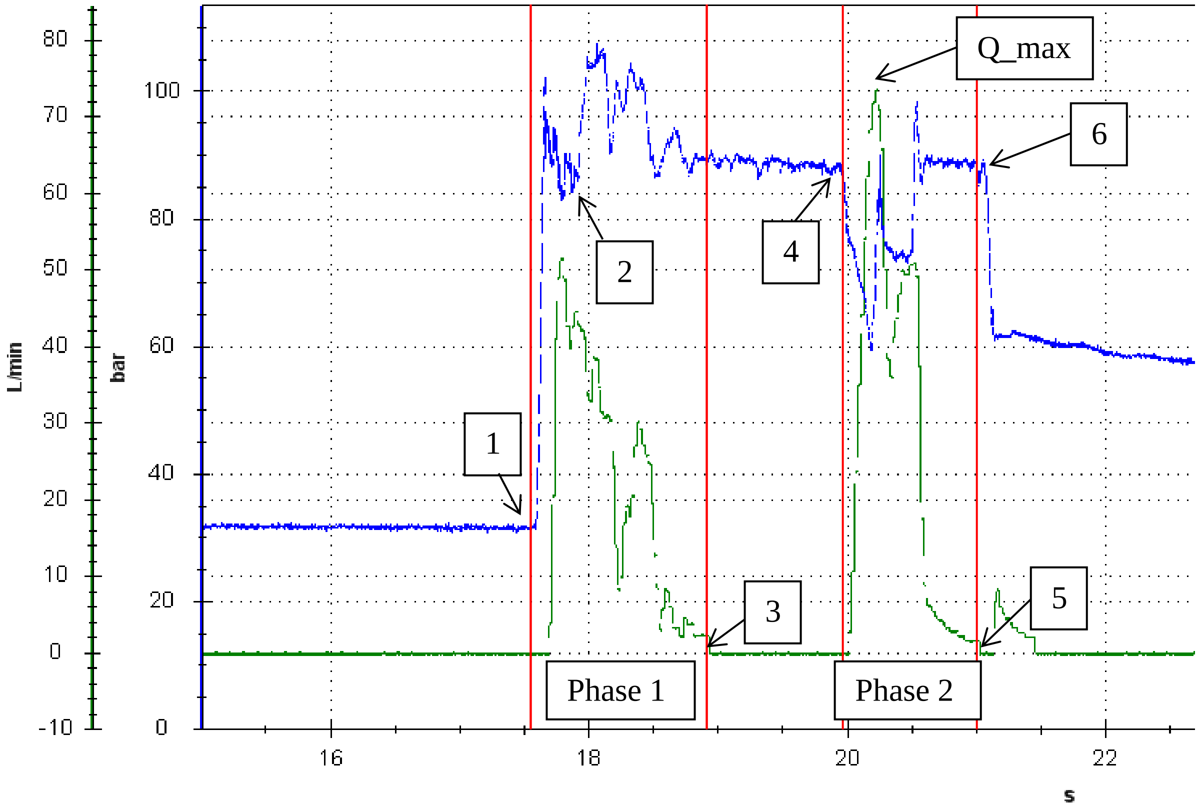


Figure 8.51: *Complete release sequence measurement with graphical output on measured data, when adding oil back to the system almost instantly upon release. Descriptions along graphics can be obtained from Table 8.10. Graphical output obtained from own test measurement.*

Table 8.10: *Critical events throughout simulated release sequence – test 2.*

Event	Measurement time (s)	Type of event

Table 8.10 continued.

1	17.55	Activation of plough point pull cylinder - start of sequence, phase 1
2	17.87	Directional valve is operated, adding oil into the system
3	18.93	Point of reaching mechanical stop and full release height – end of sequence, phase 1
4	19.91	Oil pressure on plough-point pull-cylinder is relieved – start of sequence, phase 2
Q_max	20.22	Reaching maximum oil flow of oil added to cylinder (Q_max = 73.5 L/min)
5	21.02	Plough body reaches nominal ploughing position – end of sequence, phase 2
6	21.04	Directional valve closes (due to its pre-set opening time by 3.17 seconds)

In this second test oil is added into the system just upon the moment of release. The pressure is relatively stable if neglecting the undesired pressure increase in the start of the sequence. Once more we experience a pressure drop when the plough body starts retracting in the start of the second phase of the sequence. Under the assumption that it is possible to get rid of the pressure rise in the start of the sequence oil could be added into the system in a matter of milliseconds after the start of the release sequence just as simulated in this second test. This should be pursued in order to assure that the system is able to react in time for even small obstacles with correspondingly higher demands to the reaction time of the electronic control and the directional valve in order to add oil back to the system in time. As seen from both measurements, the maximum flow rate through the directional valve and into the release cylinder is just above 70 L/min upon second phase of release sequence.

A recommendation on further implementation of automatic electronic control of directional valve based on this last test can be found in the next chapter, dealing with the design revision.

8.4.5 Evaluation of test results

It has been required to accept some uncertainty to certain aspects of the test results, which should be beard in mind. This uncertainty lies within the fact that it has not been possible to accomplish a 100 % standardized time duration for the simulated release sequences, since the achievable horizontal travel velocity of the plough point is dependent on how much resistance there is in the hydraulic stone release system relative to its initial hydraulic system pressure setting. For the measurements with a low pressure pre-setting at around 30-50 bar, there was achieved a higher velocity on the plough point than when the system pressure was pre-set at around 100 bar. One finding from the test is that the pressure stability through the plough body lift is better at higher system pressures when calculating in percent of pre-set pressure. But whether the better pressure stability is due to that the system actually gets more stable with higher system pressures or that it is a result of the lower velocity and subsequent lower oil flow in the system leaves a bit of uncertainty with regards to how the system behaves between the different load scenarios. However all measurements show the same patterns revealing delay in valve reaction upon release with subsequent high pressures in system upon release.

9. PRODUCT DESIGN REVISION

In the following sections product design revision for a second function model is performed on basis of the experiences and measurements obtained from the function test.

9.1 Release cylinder

In order to correct the system resistance in accordance with the measured mechanical release characteristics obtained from the function test, it is decided to alter the release cylinder design in terms of reducing the piston cross-sectional diameter. Recalculating required cylinder piston diameter in accordance with measured release characteristics and initial system release resistance, gives values calculated below.

Table 9.1: *Calculation basis and measurements, with system at maximum resistance.*

Input	Measurement
Force-on-point	16 kN
Maximum system pressure	160 bar (16 MPa)
Initial moment arm relation	0.41

Recalculate required axial force on cylinder upon release:

$$F_{cyl} = \frac{16\ kN}{0.41} = 39\ kN \quad (Eq\ 1)$$

Recalculate the required cylinder piston area:

$$A_{cyl} = \frac{F_{cyl}}{p_{cyl}} = \frac{39\ 000\ N}{16\ MPa} = 2437.5\ mm^2 \quad (Eq\ 2)$$

Recalculate the required cylinder diameter:

$$d_{cyl} = \sqrt{\frac{4 * A_{cyl}}{\pi}} = \sqrt{\frac{4 * 2437.5}{\pi}} = 55.7\ mm \quad (Eq\ 3)$$

$$d_{cyl} = 55\ mm \longrightarrow OK$$

The recalculated cylinder piston diameter by the newly obtained mechanical release characteristics shall be implemented into a prospective re-design of the function model.

9.2 Valve/accumulator system set-up

In order to potentially cover up for the lack in response from the pressure relief valve revealed in the function test, it is decided to implement a gas accumulator for that specific purpose.

When calculating the required accumulator capacity, we make some assumptions for the worst-case scenario. Previously, the release sequence were calculated to last for 250 ms for the worst-case scenario, where the plough body hits solid ground in 8 km/h or alternatively that we allow that two plough bodies hit a more round or circular shaped stone at driving velocity by 8 km/h, which correspond 4 km/h for each plough body. This is under the assumption that the plough point displaces in a 45° angle to the horizontal driving velocity for this last case. No matter what of those two worst-case scenarios we choose, they corresponds to the same amount of energy that needs to be absorbed by the system.

For the function model made for this present test it allows plough body lift up to 396 mm, corresponding to release cylinder travel by 150 mm. By recalculating the duration of the first phase of a release sequence for the worst-case scenario with a plough body lift by 396 mm, we obtain the recalculated time duration of the release sequence:

$$Recalculation\ factor = \frac{plough\ body\ lift}{sequence\ time\ duration} = \frac{350\ mm}{250\ ms} = 1.4$$

$$Recalculated\ sequence\ time = \frac{396\ mm}{1.4\frac{mm}{ms}} = 283\ ms$$

To obtain the release cylinder displacement relative to that time, we get:

$$\frac{Release\ cylinder\ travel}{Sequence\ time\ duration} = \frac{150\ mm}{283\ ms} = 0.53\frac{mm}{ms}$$

The measurement performed in the function test, indicate a delay by 0.1 seconds (100 ms) for the pressure relief valve. Hence, we can predict how far the release cylinder would have travelled for that time:

$$s_{cyl} = 0.53\frac{mm}{ms} * 100\ ms = 53\ mm \quad (Eq\ 7b)$$

If assuming that we exploit the design of the newly proposed release cylinder, with a cylinder piston diameter by 55 mm, we get the volume displacement of the cylinder for that first 100 ms:

$$V_{cyl,displacement} = \left(\frac{\pi * (55\ mm)^2}{4}\right) * 53\ mm = 125\ 991\ mm^3 \quad (Eq\ 31)$$

Furthermore we assume maximum system pressure by 160 bar, corresponding to 16 MPa. Then we are able to predict the energy that the accumulator must absorb:

$$E_{accumulator} = p * V_{cyl,displacement} \quad (Eq\ 6)$$

$$E_{accumulator} = 16\ MPa * 125\ 991\ mm^3 = 2\ 015\ 855\ Nmm \quad (Eq\ 6)$$

$$E_{accumulator} = 2\ 016\ Nm\ (J) = 2{,}0\ kJ$$

According to the assumptions and approximations made, there is 2 kJ of energy that needs to be absorbed due to valve response time. Most likely this energy will need to be absorbed by a gas accumulator, as previously mentioned. In order to predict the accumulator size required, some calculations are performed with regards to the maximum allowable pressure increase in the system by 10 %. When calculating the required accumulator volume, we need to know the system state and thermodynamic conditions that apply for the calculation. As already obtained from Figure 8.48 and Table 8.8, the system is in the quasi-static state for the case we are dealing with, when the accumulator is primarily needed to assist the system. Furthermore, the sequence is proceeding so rapidly, that it is assumed that there is no heat transfer from the gas to its surroundings during the compression, which implies that we're dealing with an adiabatic case. So for the accumulator gas compression it is then assumed a quasi-static adiabatic gas compression, with the following pressure and volume relations applicable by use of equation obtained from (Tipler et al 2008):

$$p_2 * V_2^{\gamma} = p_1 * V_1^{\gamma} \quad (Eq\ 32)$$

Rewritten in terms of initial accumulator volume required:

$$V_1 = V_2 * \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} \quad (Eq\ 32)$$

The γ- (gamma) value describes the relation between the heat capacity of the gas at constant pressure and –volume, respectively, under the assumption that it obeys the ideal gas law. From (Tipler et al 2008), we obtain this relation:

$$\gamma = \frac{C_p}{C_v} = \frac{C_v + nR}{C_v} = 1 + \frac{nR}{C_v} = 1 + \frac{nR}{\frac{5}{2} * nR} = \frac{7}{5} = 1.4 \quad (Eq\ 33)$$

, where:

$$n = number\ of\ mol\ of\ gas$$
$$R = Boltzmann'constant = 8.314\ \frac{J}{mol * K}$$

Calculate maximum final pressure:

$$p_2 = p_1 * 1.1 = 16\ MPa * 1.1 = 17.6\ MPa$$

Express final volume in terms of initial volume:

$$V_2 = V_1 - V_{cyl,displacement} = V_1 - 125\ 991\ \ mm^3$$

Calculate required accumulator initial gas chamber volume:

$$V_1 = (V_1 - 125\ 991) * \left(\frac{17.6}{16}\right)^{\frac{1}{1.4}}$$

$$V_1 = V_1 * 1.07 - 125\ 991 * 1.07 = V_1 * 1.07 - 134\ 810$$

$$V_1 = \frac{134\ 810}{0.07} = 1\ 925\ 862\ mm^3 = 1.9\ L$$

To absorb the added energy by 2 kJ and keeping pressure within 10 % in the first 0.1 seconds of the release sequence upon valve opening, the minimum required accumulator volume is 1.9 L according to the calculations. In order to have an additional buffer in the system it is recommended to choose an accumulator with 2.5 L initial gas chamber volume. The accumulator should have a pre-charge pressure by 90 bar in order to cover the whole system pressure range from 100-160 bar.

9.3 Automatic plough body retraction and system pressure adjustment

In order to assist the plough body in retracting back into the soil after the first phase of the release sequence, the directional valve in the hydraulic system must be operated by the electronics in order to direct oil back into the system. It is critical that this is achieved before the load on the plough point is relieved in terms of passing the obstacle. This in order to avoid loss of system pressure, as previously mentioned.

The question that remains is how to implement this functionality by the electronic control system and at the same time assuring the system functionality in every type of scenario out on the field. As seen from the last part of the testing, the functionality was rather good when adding oil back into the system before unloading the plough point. The only issue is whether it creates a further pressure increase if directing oil into the system at the same time as the pressure relief valve is trying to evacuate oil in the first phase of the release sequence. But as the last measurement shows in Figure 8.51, the pressure increase caused by opening the directional valve is not significant compared to the pressure increase at the very start of the sequence. Taken into account that the accumulator should serve as an additional energy absorbing buffer, it should be possible to add oil into the system already in the first phase of the release sequence, without disturbing the system characteristics too much. We also have to encounter a delay in the directional valve by 180 ms obtained from (Sauer-Danfoss 2012). For that reason it is recommended that the directional valve is activated as soon as a pressure above 5% of preset system pressure is measured from the system pressure sensor, adding oil back into the system in time before plough body retraction. The valve opening time should be adjustable up to 5 seconds in order to cover up for the duration of a complete release sequence.

The system software should have at least one main menu. The menu should cover the manual pre-set pressure adjustment, where desired system pressure can be specified, whereupon the system adjusts the system pressure until reaching pre-set value. The system pressure must be corrected by applying a limited opening time for the valve in order to have maximum precision for this adjustment. This implies several activations of the valve until reaching the preset pressure setting. From the pressure adjustment testing in chapter 8, we obtained that it is just a matter of milliseconds for the system to adjust from 0-200 bar without an accumulator in the system. When using the accumulator with pre-charge pressure by 140 bar we obtained from figure 8.25, that within a time of 0.6 seconds, the system was loaded from 140-200 bar. So for the pressure adjustment to be fine enough, the effective opening time interval for the valve will not be above 100 ms for the pressure adjustment itself, even if accounting for a bit larger accumulator than the one present during the measurement in the pressure adjustment test in this project. However the expected reaction time of the directional valve by 180 ms must be considered, which results in a maximum opening time by 280 ms implemented into the electronic control for the pressure adjustment function.

The software should be implemented with the possibility to adjust the opening time of the valve, for both the pressure adjustment and upon plough body retraction in the second phase of the release sequence.

Table 9.2: *System functionality in relation with electronic control functions.*

Control function	Action	Time	Valve activation input
System pressure adjustment	Pressure up	0-0.28 seconds	New pressure input by user through user interface
	Pressure down		

Table 9.2 continued.

Plough body retraction	Pressure up	0-5 seconds	Pressure increase by 5 %, sensed by pressure sensor

The information on the functionality of the plough body retraction shall be sent to Kverneland Mechatronics at a later point in order for them to supply with updated electronic control for a second function test.

9.4 Function model with design revisions

A last summarization, system capacities as a result of the design revision can be found below, including specifications on components. This is also a check list towards the metric limit specification from the product specification.

Table 9.3: *Function model design revision and final system specifications.*

System specification	Measurement	Comparison towards metric limit specification
Force-on-point resistance	10 - 16 kN	Identical – OK
System pressure range	100 - 160 bar, adjustable from user interface	Identical – OK
Reduction in release resistance throughout plough body lift for 350 mm lifting height (release characteristics)	21% reduction in system release resistance with regards to force-on-point	Within required range below 25 % reduction in force-on-point - OK
Allowed pressure deviation relative to pre-set system pressure	Calculated to 50 % from single measurement	Not obeying 10 % requirement at this stage
Measured release height	396 mm	Within specified range (350-400 mm) – OK.

We can once more conclude that the measurements revealed that the system lacks capacity in order to be able to keep system pressure within 10 % of preset pressure. However, from the product design revision, it pursued to improve the system design so that it obeys this requirement. But the actual capacity for the new design can only be verified by further testing. This will be part of the next step in the process of developing this product, which comprise the next project step, as part of further work with regards to this specific project.

On the other hand, the function test confirmed that the function model obeyed the rest of the so far stated metric limit specifications for the product.

10. DISCUSSION

10.1 Product design

The discussion on the product design in this section include further system implementations intended for the development of this product in the longer term. Discussion on the present product design in terms of outcome of testing and actions performed on basis of the test can be found in chapter 8 and 9 in terms of function test and product design revision.

10.1.1 Future system implementations

In terms of developing a final product, some thoughts on both mechanical design and hydraulic system functionality is outlined in the discussion in the next sections. It is important to note that this discussion deals with proposals to the system function and design in a longer perspective than where the project is standing at this point.

Mechanical system

The mechanical design of the function model present at this point is exploiting the geometry that is present on the Kverneland plough beam used today. If however there is an alternative to change the design of the plough beam, it should be considered to integrate the mechanical design of the stone release system into the plough beam design. Preferably, the cylinder would then be located inside the beam closer to the front linkage towards the beam holder.

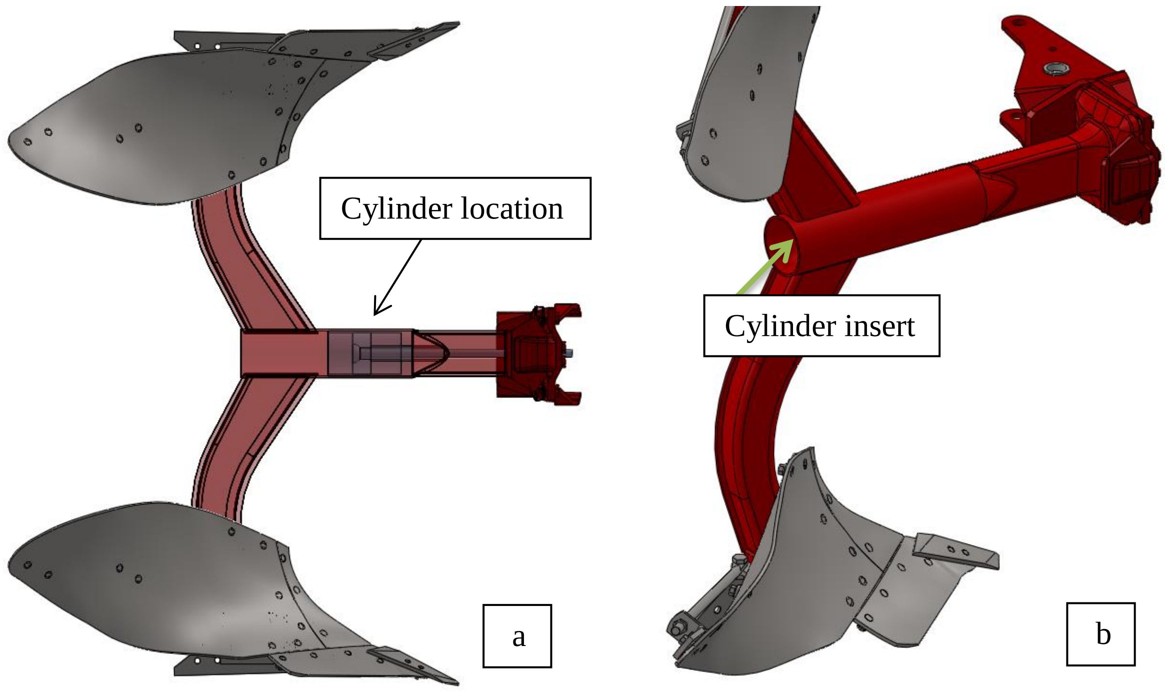


Figure 10.1 a, b: *Overview of how an alternative solution for the mechanical system might look like if alternating the beam design and locating the cylinder inside the beam.*

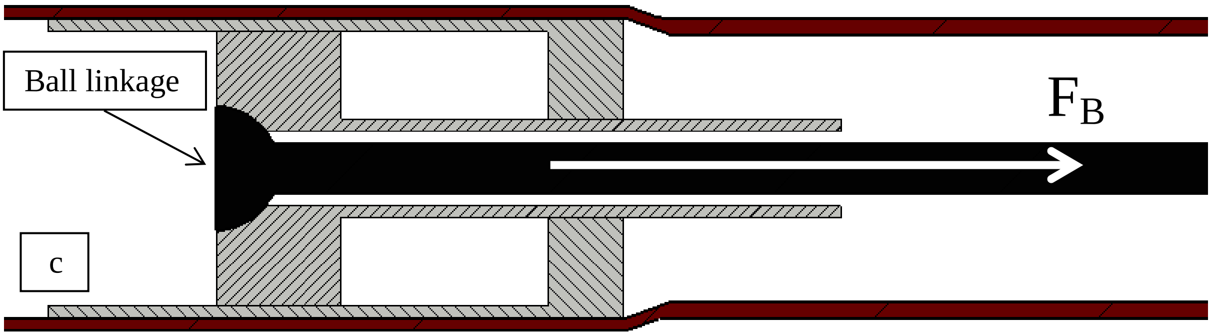


Figure 10.1 c: *More detailed section view of how the intended cylinder mechanism might look like. The force (F_B) on the tie rod is illustrated working directly on the cylinder piston. Note the ball link at the piston in order to allow rotation and hence a trouble-free linear translational movement of the piston.*

Design on illustrations are result of own re-design on existing plough beam design obtained from (Kverneland Group 2012).

The main reason for locating the cylinder inside the beam rather than at the very rear is that we obtain more ideal release characteristics in terms of less decrease in release resistance throughout the release sequence. This is caused by the fact that we have a shorter tie rod, meaning less decrease moment arm y4 (referring to figure A1.9 in appendix A1). Another reason for choosing this solution over possible solutions of locating the cylinder outside the beam is that it appears tidier in terms of aesthetic design.

Automatic hydraulic system adjustment

The hydraulic stone release system and function test that has been worked out so far only implement manual adjustment of the system pressure and subsequent system resistance. However, in the longer run, if the system is proving successful and not failing system requirements in further testing, the aim should be to implement an automatic adjustment to the system. This automatic system adjustment should pursue system pressure according to soil conditions. Since this lies further ahead, it will not be dealt with in detail at this point, but a rough idea of how it could be implemented is presented.

<u>Background</u>

The idea behind the automatic system adjustment of hydraulic stone release systems come from the Comfort plough project carried out by Kverneland in the early 1990’s. This was only a research project and the final product never reached the market. However if this hydraulic stone release system becomes a reality it should be pursued to have the same kind of automatic adjustment for the final product, as the one used for the Kverneland Comfort plough.

It has been difficult to obtain information on the principle behind the stone release system adjustment on the Comfort plough. However, from a phone call with Sigve Klakegg who took part in the Comfort plough project, the following info was obtained on that system:

- A linear resistance cylinder position sensor including a permanent magnet was used to keep track of the position of the cylinder piston.

- If the cylinder piston travelled outside the given position interval, indicating an increase in resistance in the soil, the system pressure was increased in order to force the cylinder and subsequently the plough body back into nominal ploughing position

Once more, info obtained from (Klakegg 2013).

Intended working principle, automatic adjustment position control

Based on the brief information obtained from Sigve Klakegg, a proposal on how to implement such an automatic adjustment to the Hydro Reset system is outlined in the following section.

On the plough beam of the first plough body at the front of the plough, a "position-sensor cylinder" is connected as illustrated below. For the rest of the plough bodies, there are cylinders with no position sensor attached, whereas all cylinders are connected together with the "sensor cylinder" in the same hydraulic system. The idea behind this type of measurement is that a permanent magnet inside the cylinder piston creates a magnetic field along with the external linear transducer, where electric current is applied. The location of this magnetic field makes it possible to locate the cylinder piston and its displacement along the cylinder.

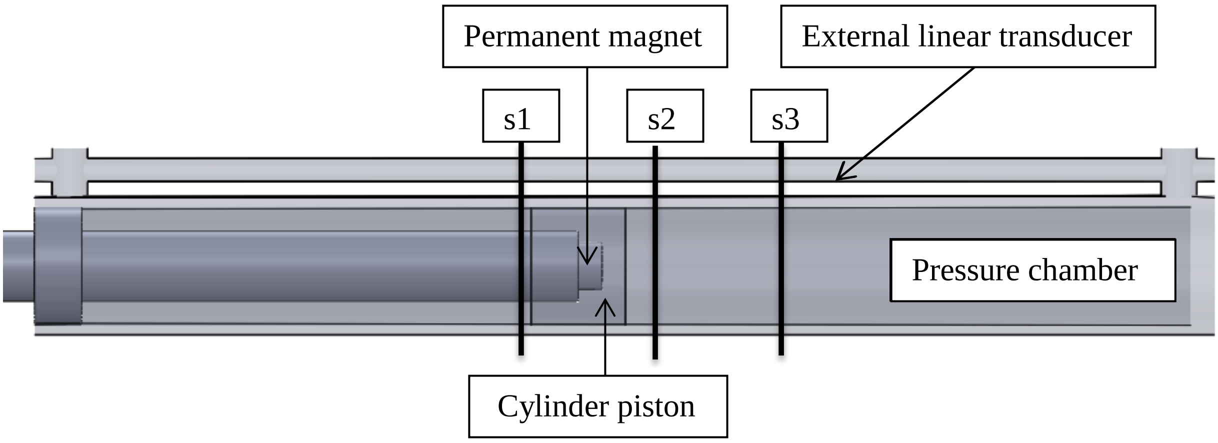


Figure 10.2: *"Sensor cylinder" implemented with external linear transducer along the cylinder tube. A permanent magnet is located inside the piston in order to measure position on cylinder piston. The sketch in this figure is not scaled, and distances between positions s1, s2 and s3 are exaggerated in order to make the illustration more clear. Illustration is result of own work by 3D model worked out in Solid Works. The idea behind the design is however based on previous Comfort plough project by Kverneland Group with information once more obtained from (Klakegg 2013).*

System adjustment due to various soil conditions

During ploughing with "sensor cylinder" plough body in nominal position (in soil) the following applies:

- When "right" end of piston moves out of position (passed s2) → PRESSURE UP - ON
- When "left" end of piston reaches s1 → PRESSURE UP – OFF

As briefly specified, the cylinder piston has a position interval between s1 and s2, where it is regulated by adding oil to the system, moving the piston from s2 to s1.

Furthermore the fact that there is a minor leakage over the spool on the directional valve is exploited. This is to make the system adjust to the present conditions in the case that a pressure reduction is required due to entering lighter soil. Then the system can “bleed” out, and when once more reaching position s2 the system can correct itself by adding oil. However the opening time of the directional valve should be very small to avoid serious and undesired over-loading of the system, since only a minor correction of the pressure conditions is required (e.g. valve opening time by 10 ms until reaching position s1, depending on system volume and number of plough bodies on plough).

<u>System functionality when hitting stone/obstacle</u>

Regardless of if it is the “sensor cylinder” or one of the other cylinders that are being forced to displace, oil will be evacuated through hydraulic-operated pressure relief valve in order to keep pressure down at preset pressure in the best possible manner in the first part of the release sequence, when body is lifted. However there is a need to make the system sense the difference between when only adjusting the system as described in the previous section, and when “refilling” the system when plough bodies retract back into the soil after being released.

When one of the cylinders other than the “sensor cylinder” is releasing, the “sensor cylinder” works as described when it is in nominal ploughing position. When the “sensor cylinder” itself releases, the system detects that the piston is moving passed s2 as before and starts adding oil in short intervals as given in previous exemplification. However, when passing s3 it is an indication that the “sensor cylinder” is releasing, and oil is added until the piston is once more reversing passed s3 and furthermore corrected in smaller sequences back into ploughing position passed s2. When the system starts pumping oil constantly when the piston is passing s3 in the first phase of the release sequence, it requires that the system is able to drain oil from both the cylinder and the oil flowing into the system through the directional valve. The reason for directing oil into the system already in the first phase of the release sequence is, as mentioned earlier, due to the uncertainty of the size of the obstacle – and subsequently the time duration until the plough point is once more unloaded. So, adding oil already in parts of the first phase of the sequence, secures the availability of oil when needed in the second phase.

It is important to note that ideas regarding the automatic system adjustment outlined in this last section are really on the idea stage. So the ideas outlined are only a guideline to how this functionality might be implemented at a later point, and therefore the information given should be treated correspondingly. Its functionality is not at all studied in detail. Much more research is needed on this automatic system adjustability feature before implementing it into the system.

10.2 Product development process

This second section of the discussion deals with the product development process of this project. The project process is evaluated including aspects of learning while carrying out the work.

The basis on which this project is carried out, was the pre-project where a temporarily product concept was chosen. The introduction of this project dealt with the findings of the pre-study, where problems regarding the design were discussed. Furthermore, a mission statement for this project was worked out. Following the mission statement, that also implies that this is a product development project, a suitable project template has been used by encroaching on the product development step-by-step, using the method by Stuart Pugh. The product development starts with a product specification, followed by a conceptualization- and concept selection phase, before starting on the product design and the relevant calculations in order to assure the performance of the chosen product design as far as feasible by theoretic calculations.

However, even if following a relatively strict, step-by-step product development regime, a new aspect that affects the product and needs to be taken into consideration can arise, especially by experience during testing. Fine tuning of the design in the last part of the development process may also alter on assumptions made in previous calculations.

When performing such a product development project, it is important to work proactively. This should be achieved in terms of taking all possible pre-cautions upon defining both product and the project process. All evident considerations with regards to the product design should be taken into account before carrying out the development process. At the same time, we should account for situations that might make it necessary to alter on both the product design and the pathway of the development process.

Due to late delivery of hydraulic valve the testing became a bit delayed, but by leaving a 3-week “buffer” between deadline for functional test and deadline for the results interpretation and design revision, it was achieved to reach the overall deadline of the testing including the results interpretation and design revision in the end.

This project has been relatively comprehensive involving several special fields including function test, where development of electronics and software has been outsourced.

11. CONCLUSION

The function test revealed lack in system capacity in terms of maintaining constant system pressure during the release sequence. On the other hand, the measured mechanical release characteristics showed the same tendencies as the calculation and proved to be acceptable, however with a bit higher resistance due to mechanical friction in system and weight of plough beam. This implies that the system require a smaller cylinder, which is ideal due to the smaller volume of oil that needs to be transported in and out of the system for each release sequence. In order to cover up the lack of capacity in the hydraulic system upon release, an accumulator has been suggested implemented for the further design. Some assumptions on the required timing on the control system for further implementation of the second phase of the release sequence have been performed. Test data from the simulation of the complete release sequence and assumptions in relation to those measurements should be communicated further to software supplier (Kverneland Group Mechatronics) in order to implement the plough body retraction functionality.

The problems stated at the very end of the project introduction chapter in section 1.9, have been worked out as far as possible. The problems with regards to oil flow were handled by implementing the pressure relief valve. However the problem with regards to valve delay was not completely solved by implementing the chosen pressure relief valve. However, as once more specified in the design revision, that problem is intended solved by the implementation of the specified accumulator.

The project mission statement which set the target for this project implied the development of a function model in order to perform a function test on the specified stone release system. That has been achieved, with test results and a proposed system re-design as the final outcome of this project. The mission statement for this project is hereby considered answered.

11.1 Recommendations

The project is recommended taken further for a re-test along with the proposed product re-design, since the problems due to hydraulic system capacity is still not solved. Furthermore the automatic plough body retraction functionality is recommended implemented into the system for the next test.

11.2 Further work

In order to bring this project the necessary step further, the following primary actions are required:

- Re-test once more dealing with system performance in the first phase of the release sequence
- Implementation of second phase of release sequence facilitated by electronic control of directional valve, followed by test in terms of automated complete release sequence
- Further development implementing system pressure adjustment by activating directional valve via pressure sensor input

Further ahead there are some secondary actions that must be taken. This is under the assumption that the mentioned primary actions are taken, and that the related problems to the functionality are solved in order to bring this project further. The secondary actions that should be taken are:

- Multiple body test in test lab with 2-3 plough bodies simulating release sequences with different load scenarios
- Full scale test on plough out in field
- Further design optimization on tie plate with stress analysis by dynamic structural loading and life time assessment, both by calculation and by test
- System adaption towards different types of tractor hydraulic systems

On the non-technical field the following actions must be taken and carried out:

- Market analysis, including analysis of cost/benefits, collection of customer feedback, determination of price level and research on intellectual property rights

12. REFERENCES

Book (5)

Brautaset, K. (1983). *Innføring i oljehydraulikk*. [Oslo]: Universitetsforlaget. ISBN 978-82-00-28325-6. 340 p. (chapter 3, 5)

Fox et al. (2010). *Introduction to Fluid Mechanics*. [Asia]: John Wiley & Sons. 7th Edition. ISBN 978-0-470-23450-1. 687 p. (chapter 2, 4, 6, 8)

Johannessen, J. (2002). *Tekniske tabeller*. [Skien]: Cappelen. 2.utgave. ISBN 82-02-16822-8. 158 p (p. 38)

Kjølle, A. (1995). *Oljehydraulikk : Teknologisk grunnlag*. [Trondheim]: Tapir. 2. utgave. ISBN: 82-519-1197-4. 469 p.

Tipler et al. (2008). *Physics for scientists and engineers*. [New York]: W.H. Freeman and Company. 6th Edition. volume 1. ISBN-10: 0-7167-8964-7. 1412 p. (chapter 17, 18)

Computer Program (3)

Kverneland Group Mechatronics. (2013). *Stone release settings*.
Microsoft Corporation. (2010). *Microsoft Visio*.
Parker Hannifin Corporation. *Senso Control*.

Pamphlet (1)

Terjesen, G. (2012). *Prosjektering av stålkonstruksjoner: Skrueforbindelser*. [Ås]: IMT/UMB. 13 p.

Personal Communication (3)

Erlend Sølvberg. (2013). *Heat treatment and material yield limit*
Forbord, P. (2013). *Phone call and email correspondence*: *Pressure Valves*
Klakegg, S. (2013). *Phone correspondance: Comfort plough stone release system*.

Internal documents, Kverneland Group Klepp R&D (6)

Report (2)

Kverneland Group. (2002). Test rapport 11A-01-33
Kverneland Group. (2005). Test rapport 11A-01-37

Product and production information (2)

Kverneland Group. (2004). *Tie rod: Technical drawing*. Kverneland Group.
Kverneland Group. (2013-2). SAP internal database.

Standard (1)

Kverneland Group. (2006). *Generell testprosedyre åser*.

Computer Aided Design 3D Model (1)

Kverneland Group. (2012). *3D model plough beam*.

Unpublished Work (1)

Bjørås, E. (2012). *Kverneland Hydro Reset System - Hydraulic Stone Release System on Plough*: Norwegian University of Life Sciences. Unpublished manuscript.

Web Page (11)

Hydraulic datasets (5)

Comatrol-1. (2010). *Cartridge Valves Technical Information – Sequence Valves: CP241-21*. http://www.comatrol.com/cartridge-valves-catalog/Section06_Sequence_valves.pdf (accessed 13.03.2013)

Comatrol-2. (2010). *Cartridge Valves Technical Information – Relief Valves: CP211-2*. http://www.comatrol.com/cartridge-valves-catalog/Section04_Relief_valves.pdf (accessed 13.03.2013)

Hydac International. (2009). *Electronic Pressure Transmitter - HDA 4400*. http://www.hydac.com/no-en/products/sensors/pressure-sensors/pressure-transmitters/hda-4400/show/Download/index.html (accessed 08.03.2013)

Sauer-Danfoss. (2012). *PVG32 Proportional Valve Group - Technical Information*. http://www.sauerdanfoss.com/stellent/groups/publications/documents/product_literature/520l0344.pdf (accessed 15.01.2013)

Shell. (2013). *Tellus S3 M 46 - Premium sinkfri hydraulikkolje*. http://www.epc.shell.com/Docs/GPCDOC_Local_TDS_Norway_Shell_Tellus_S3_M_46_%28no%29_TDS_v1.pdf (accessed 19.02.2013)

Computational fluid dynamics (2)

cfdesign.com. *Two equation turbulence models*. http://www.cfdesign.com/OnlineHelp/2013/Theory/governing%20equations/Two_Equation_Turbulence_Models.htm (accessed: 22.03.2013).

CFD-Online. *Turbulence intensity*. http://www.cfd-online.com/Wiki/Turbulence_intensity (accessed: 22.03.2013).

Historical review (1)

Jones, A. *A Brief History of The Plough*. www.ploughmen.co.uk (accessed: 18.02.2013).

Corporate information (3)

Kubota Corporation. (2013). *Corporate Information*. www.kubota-global.net. (accessed:

18.02.2013).

Kverneland Group. *Product Information*. www.kvernelandgroup.com. (accessed: 18.02.2013).

Kverneland Group. (2013). *Corporate Information*. www.kvernelandgroup.com. (accessed: 18.02.2013).

13. APPENDIX

List of appendix

A1: Calculation of release characteristics

The release characteristics are made out of the calculation of the moment arm relations in the mechanical part of the stone release system. It is comprised by the force that is applied to the plough point and the necessary axial counterforce on the release cylinder to withstand that force, and their relation throughout the release sequence. The calculations are exploiting trigonometry exploiting picked distances and angles that are dependent on each other throughout the sequence. The trigonometric relations will be presented both visually and by mathematical equations. The calculations themselves have been executed in Excel and can be found as digital copy in compact disc accompanying this report. The plough beam and "moment arm relation 1" geometry, which is referred to later, is based on the design owned by Kverneland Group obtained from (Kverneland Group 2012). However, all calculations on the release characteristics on this geometry in addition to extended geometry by "moment arm relation 2" are a result of own work. Figures showing plough beam are consequently reworked from (Kverneland Group 2012).

Trigonometric relation 1 and 2

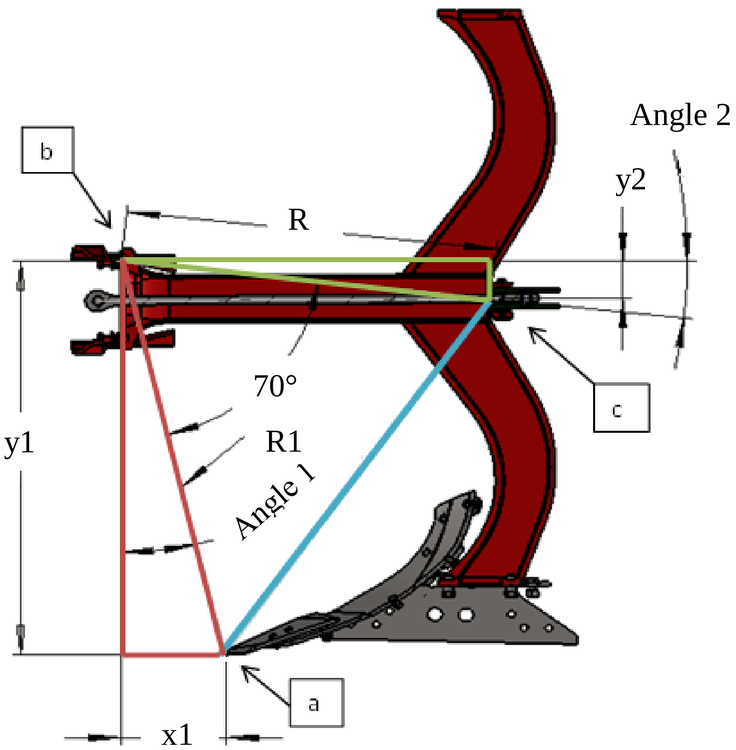


Figure A1.1: *This first trigonometric relation shows the fixed triangle geometry (a, b, c) between the plough point (a) pivot point (b) and the very rear of the beam tube at which the tie rod centerline intersects (c). By exploiting this rigid triangle 2, we get the relation between the "red" triangle 1 and the "green" triangle 3.*

First of all we use Pythagoras and the dependency between y1 and x1. "y1" is calculated for every input value of x1 by Pythagoras since R1 is constant.

$$y1 = \sqrt{R1^2 - x^2}$$

By trigonometry we obtain the value of angle 1:

$$Angle\ 1 = \left(cos^{-1}\left(\frac{y1}{R1}\right)\right) * \left(\frac{180}{\pi}\right)$$

Obtain angle 2:

$$Angle\ 2 = 90° - Angle\ 1 - 70°$$

Calculate y2:

$$y2 = \sin\left(\frac{Angle\ 2 * \pi}{180}\right) * R2$$

Trigonometric relation 3

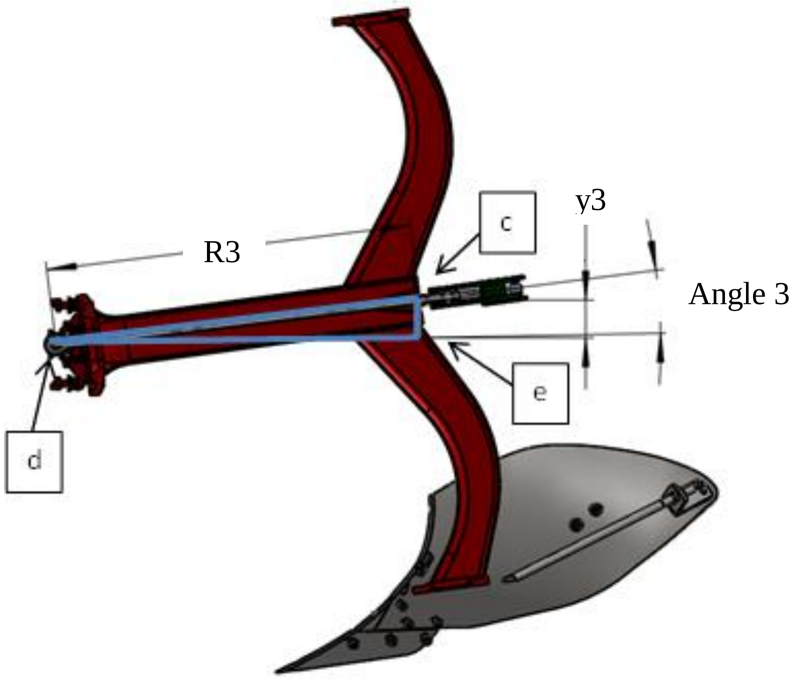


Figure A1.2: *Illustration of the triangle geometry (c, d, e), where (c) is the same point as on figure A1, (d) is the front fix of the tie rod and (e) is the coordinate where the variable x-coordinate of (c)intersects with the constant y-coordinate of (d). R3 is constant, whereas y3 and angle 3 are variables. Note: This illustration show the plough beam at which it is lifted. In nominal position the initial value of y3 = 0.*

First y3 is obtained from the differentiation between the initial value and present value of y2:

$$y3 = y3_{initial\ value} + (y2_{initial\ value} - y2)$$

Furthermore by trigonometry:

$$Angle\ 3 = \left(sin^{-1}\left(\frac{y3}{R3}\right)\right) * \left(\frac{180}{\pi}\right)$$

Trigonometric relation 4

Obtain angle 4 from angle 3:

$$Angle\ 4 = Angle\ 4_{initial\ value} - (Angle\ 3 - Angle\ 3_{initial\ value})$$

Calculate y4:

$$y4 = \sin\left(\frac{Angle\ 4 * \pi}{180}\right) * R4$$

By Pythagoras we obtain x4:

$$x4 = \sqrt{R4^2 - y4^2}$$

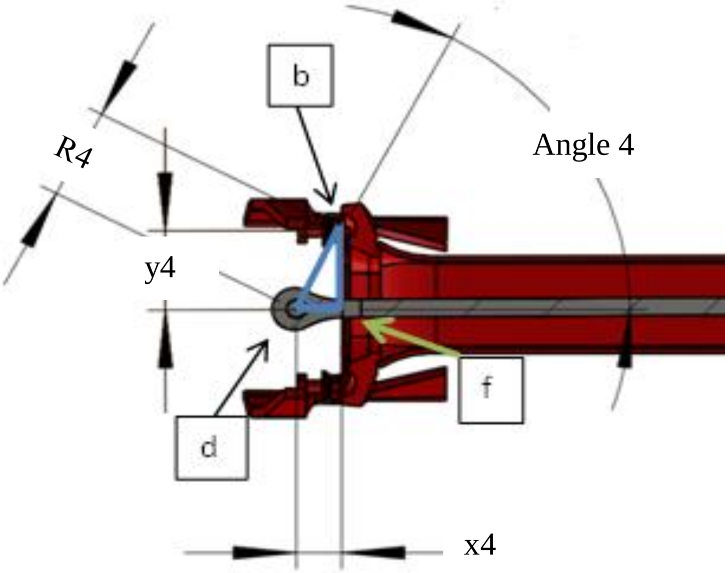


Figure A1.3: *Angle 4, as illustrated, can be obtained from angle 3, which in the next instance allows for a calculation of y4 and x4.*

Trigonometric relation 5

Relation 5 consist of triangle (g, h, i), where x5 and z5 are variables, while R5 is constant.

Obtain x5 from x4:

$$x5 = x5_{initial\ value} - (x4 - x4_{initial\ value})$$

Obtain z5 by Pythagoras:

$$z5 = \sqrt{R5^2 - x5^2}$$

Calculate angle 5:

$$Angle\ 5 = \left(tan^{-1}\left(\frac{x5}{z5}\right)\right) * \left(\frac{180}{\pi}\right)$$

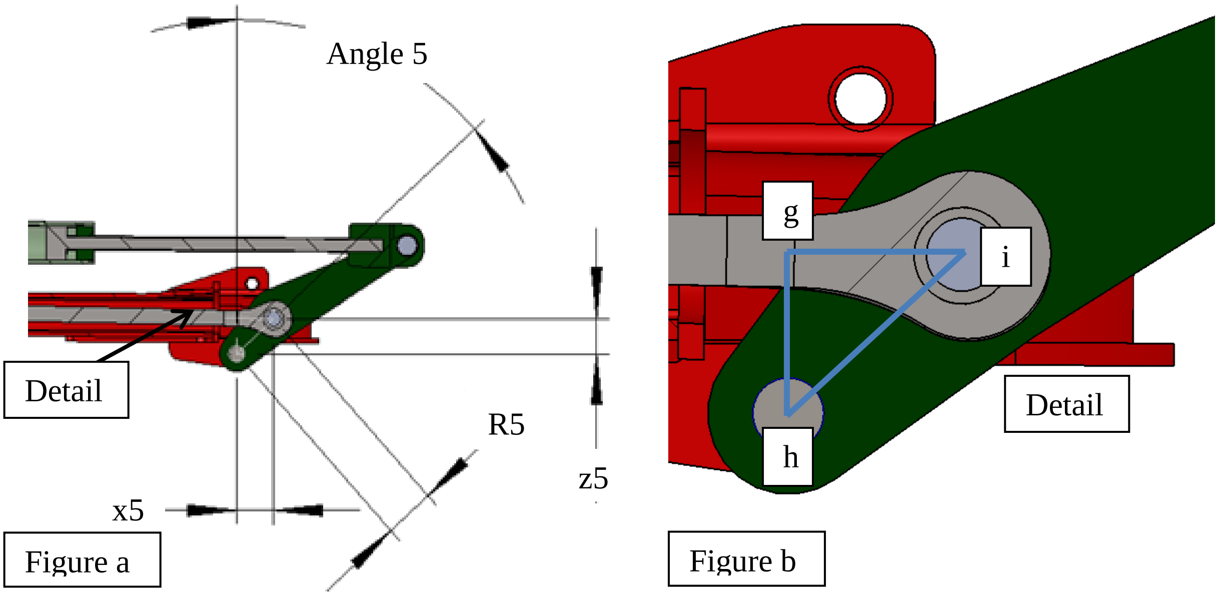


Figure A1.4 a: *In addition to x5 and z5, angle 5 is naturally a variable as well.*

Figure A1.4 b: *Triangle (g, h, i) shown in more detail.*

Trigonometric relation 6

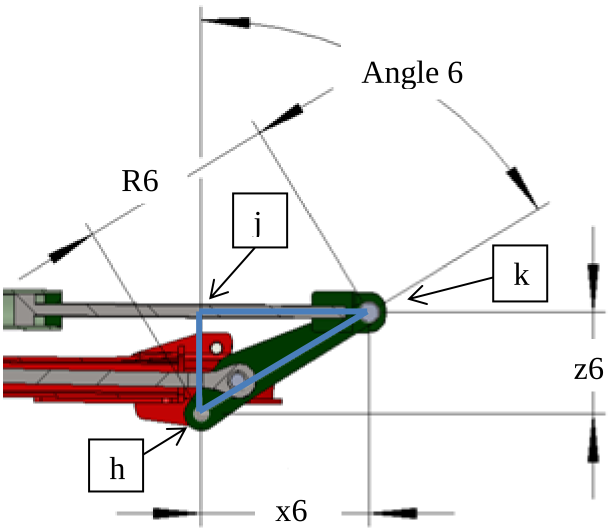


Figure A1.5: *Illustration of the geometry relation (h, j, k) consisting of variables x6, z6 and angle 6. R6 is constant.*

Obtain angle 6 from angle 5:

$$Angle\ 6 = Angle\ 6_{initial\ value} - (Angle\ 5_{initial\ value} - Angle\ 5)$$

Calculate x6:

$$x6 = \sin\left(\frac{Angle\ 6 * \pi}{180}\right) * R6$$

Calculate z6:

$$z6 = \sqrt{R6^2 - x6^2}$$

Correction of z5

When we calculated z5 we didn't take consideration of the fact that the tie rod is in a certain angle with the x-axis. Consequently there is a smaller error in the calculation that we are going to correct in the following calculations.

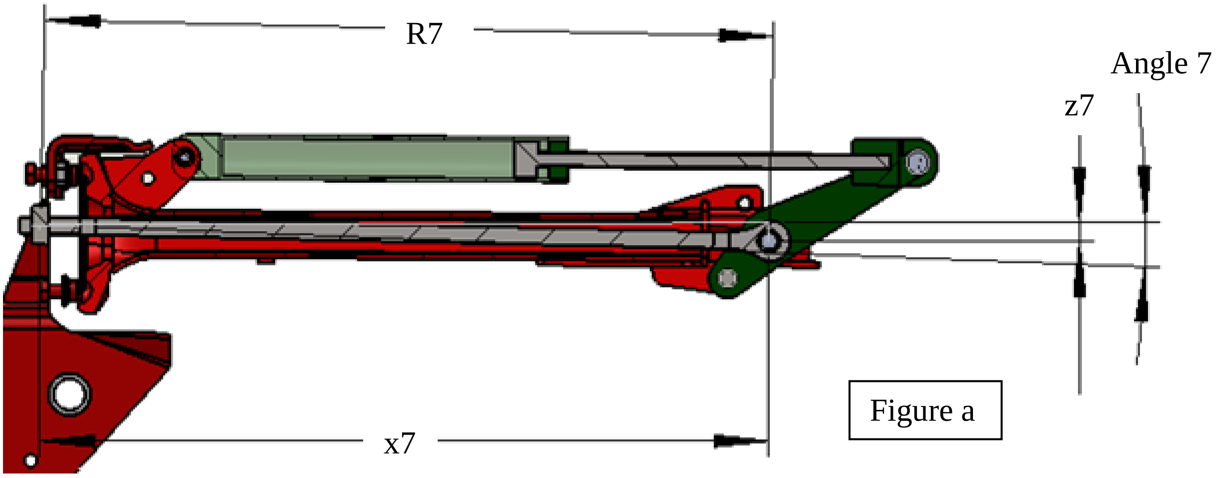


Figure A1.6 a: *This illustration shows the geometry relation of which moment arm z5 is corrected. R7 is constant while x7, z7 and angle 7 are variables.*

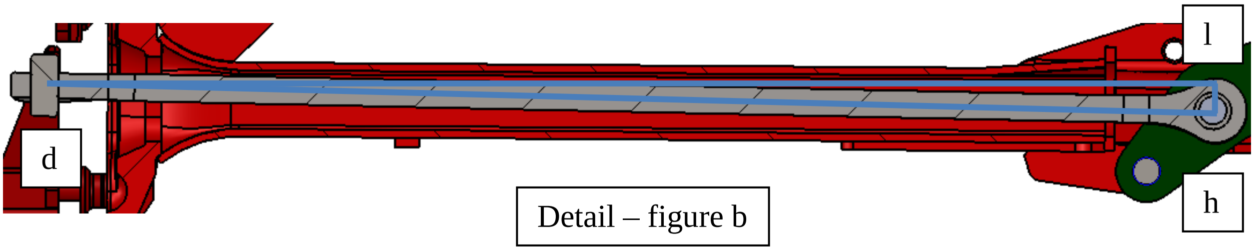


Figure A1.6 b: *Detailed view from Figure A1.6 a, illustrating triangle (d, h, l).*

Obtain z7 from z5:

$$z7 = z7_{initial\ value} - (z5 - z5_{initial\ value})$$

Calculate x7:

$$x7 = \sqrt{R7^2 - z7^2}$$

Calculate angle 7:

$$Angle\ 7 = \left(tan^{-1}\left(\frac{z7}{x7}\right)\right) * \left(\frac{180}{\pi}\right)$$

In order to use the previous numbers to correct the error in z5, we calculate some trigonometric relations which make out the correction factor of z5:

$$factor_{z5,corr} = \left(\frac{x7}{z7} \Big/ \frac{x5}{z5} \right)$$

Calculate the corrected value of z5:

$$z5_{corr} = z5 + \left(\frac{z5}{factor_{z5,corr}} \right)$$

Correction of z6

In the same way as for moment arm z5 there is a smaller error in the calculation of moment arm z6, since the cylinder will rarely be in line with the x-axis. We will do the same type of correction in this case as well.

Obtain z8 from z6:

$$z8 = z8_{initial\ value} - (z6_{initial\ value} - z6)$$

Calculate x8:

$$x8 = x8_{initial\ value} - (x6_{initial\ value} - x6)$$

Calculate angle 8:

$$Angle\ 8 = \left(tan^{-1}\left(\frac{z8}{x8}\right) \right) * \left(\frac{180}{\pi}\right)$$

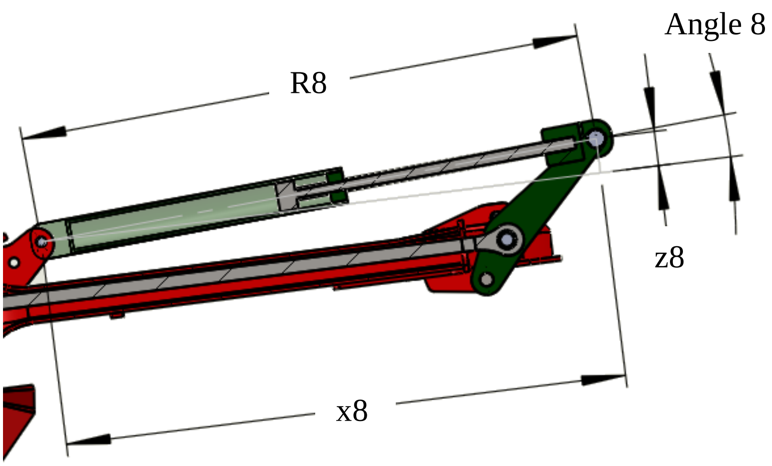


__Figure A1.7:__ This illustration shows the geometry relation of which moment arm z6 is corrected. Use the relation between variables x8, z8 and angle 8.

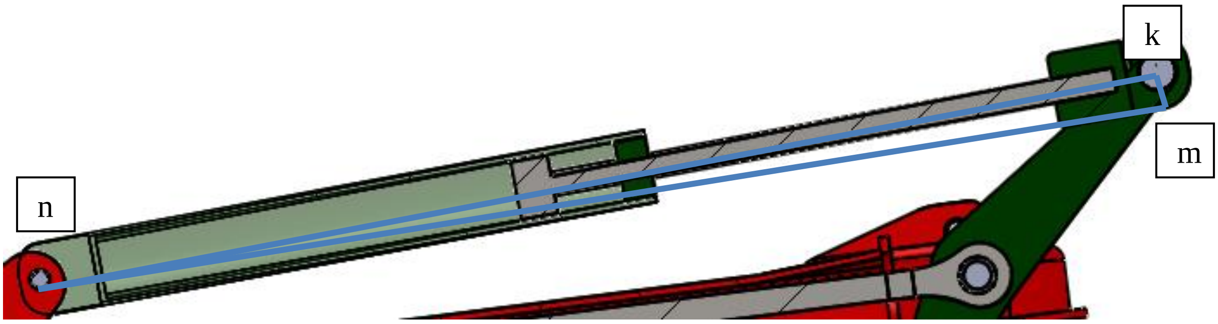


Figure A1.8: *Detailed view of triangle (k, m, n) making the geometric relations shown in figure A7.*

In order to use the previous numbers to correct the error in z6, we calculate some trigonometric relations once again, in order to make out the correction factor of z6:

$$factor_{z6,corr} = \left(\frac{x8}{z8} \Big/ \frac{x6}{z6} \right)$$

Calculate the corrected value of z6:

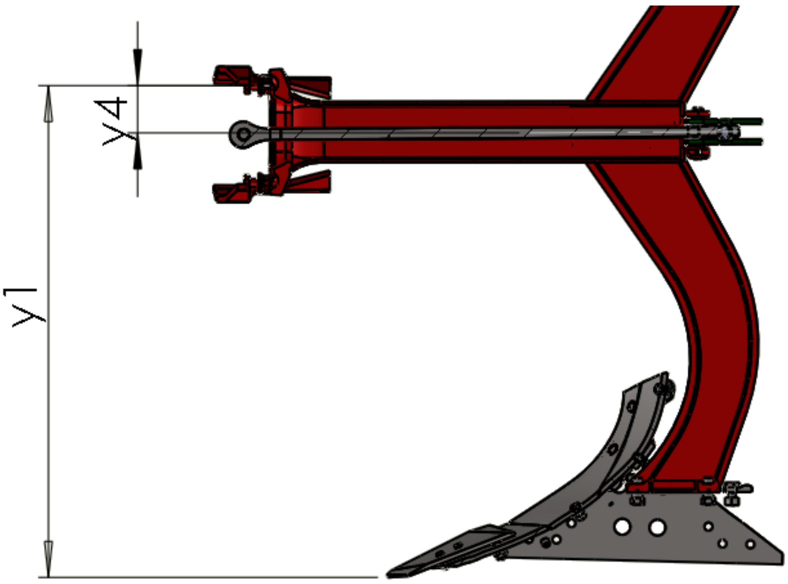
$$z6_{corr} = z6 + \left(\frac{z6}{factor_{z6,corr}} \right)$$

Final moment arm relations

Finally; "moment arm relation 1", "moment arm relation 2" and the total release characteristics can be calculated and obtained:

$$Moment\ arm\ relation\ 1 = \frac{y4}{y1}$$

Figure A1.9: *Show the moment arms connecting upper ball bearing with tie rod center axis and plough point respectively.*



$$Moment\ arm\ relation\ 2 = \frac{z6_{corr}}{z5_{corr}}$$

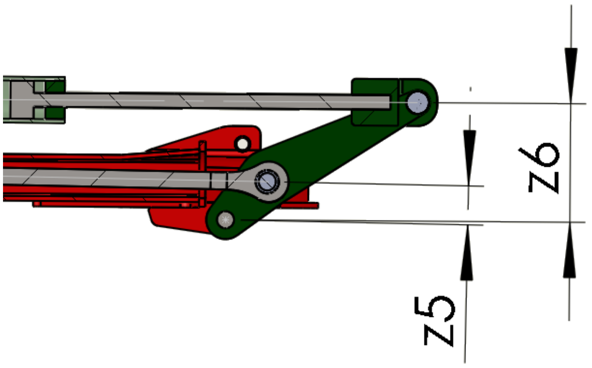


Figure A1.10: *Show the two moment arms for the tie rod and the cylinder rod respectively.*

Release characteristics

The final release characteristics are the product of the two moment arm relations:

$$Release\ characteristics = \left(\frac{y4}{y1}\right) * \left(\frac{z6_{corr}}{z5_{corr}}\right)$$

The release characteristics are used to calculate the required cylinder counterforce (F_{cyl}) towards the force on the plough point (F_A) to keep the system in balance.

$$F_{cyl} = \frac{F_A}{Release\ characteristics}$$

Doing the calculation with regards to the actual resistive force on plough point (F_A), gives:

$$F_A = F_{cyl} * Release\ characteristics$$

Calculating cylinder velocity and oil flow

The cylinder velocity is obtained by the relationship between cylinder displacement along the calculated x8 and the time elapsed within given time interval, which in the release characteristics (section A1.1) has been set to 5 ms intervals.

Cylinder velocity:

$$v_{cyl} = \frac{x8}{t}$$

Furthermore the oil flow is obtained from the cylinder velocity as follows:

$$Q = v_{cyl} * A_{cyl}$$

A1.1 Excel documentation of calculated release characteristics

The calculations on the mechanical release characteristics have all been worked out in Microsoft Excel. As a last part of dealing with the mechanical release characteristics there has been done a verification of characteristics and comparison between calculated and measured release characteristics, which can also be found in a separate document. The calculation of the release characteristics for the different designs outlined during the development process can be found as digital copy in compact disc accompanying this report, along with a calculation basis for the calculations performed in chapter 7. An overview of the attached Excel documents relative to the different design presented throughout this project is listed below:

- "Calculated mechanical release characteristics concept designs.xlsx"
- "Calculation basis chapter 7.xlsx" (on basis of release characteristics)
- "Calculated release characteristics function model.xlsx"
- "Verification of release characteristics (Test 2).xlsx"

A2: Pressure drop along pipeline relative to time

This calculation follows the calculation of pressure loss along pipeline, performed in chapter 7. Instead of "only" calculating the pressure loss for each element along the pipeline, this calculation is taken a bit further by calculating the pressure at each node along pipeline, on the basis of the pipeline illustrated in Figure 7.9. In this calculation the pressure at each node is calculated in terms of time for one release sequence, where the specific momentary oil flow velocity at each given time instant is implemented into this calculation. Furthermore it is calculated with how far the oil as reached through the pipeline system in order to predict the pressure loss for the oil at nodes at each given time instant, on basis of the pressure losses already calculated along the pipeline elements. Note that pressure losses along pipeline are carried out with regards to average oil flow in previous calculations, and will not comply a 100 % with the momentary oil flows used for the calculations at each given time instant in this calculation. The calculation background for the following calculations can be found in Excel document named "Calculation basis chapter 7.xlsx". Calculations are performed by using "bar" as unit of pressure, equivalent to 10^5 Pascal in terms of SI-units. Equations used can be found in Table 3.3 and 3.4 in chapter 3, along with the respective references.

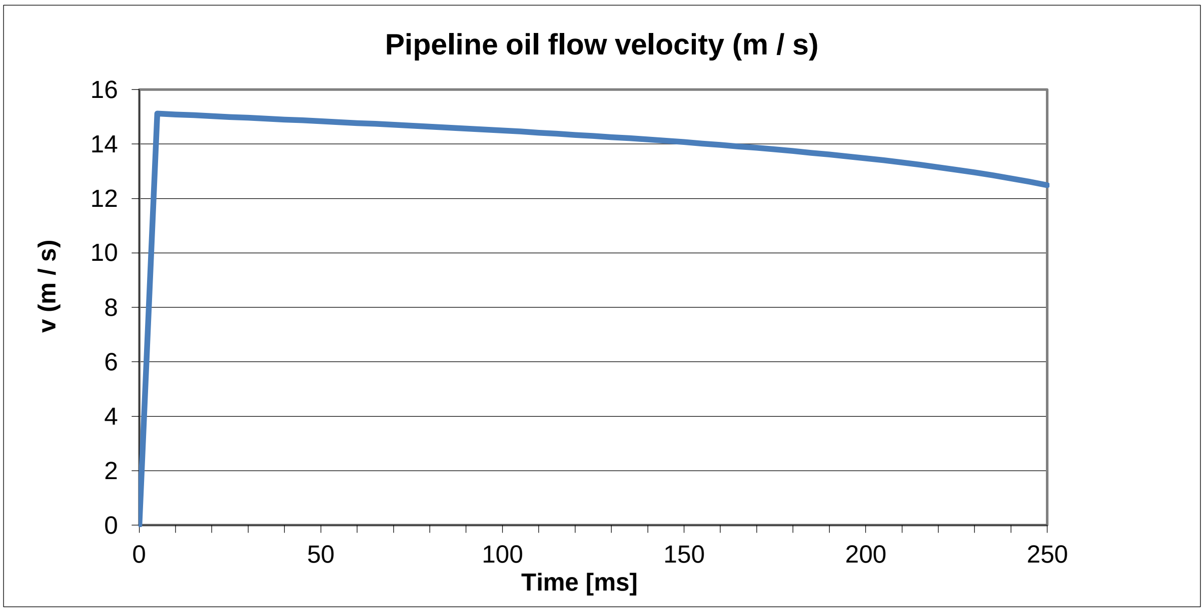


Figure A.11: *This is the calculated pipeline oil flow velocity with regards to time during a release sequence, which forms the basis for the further calculations of pressure at the specified measuring points along the pipeline. Graphical output is obtained from calculations carried out as in appendix A1 and can once more be found in Excel document related to calculations for this chapter.*

The illustrated pipeline oil flow velocity with regards to time is obtained from the oil flow characteristics by looking at the pipeline diameter:

$$v_{oil} = \frac{Q}{\left(\frac{\pi * d^2}{4}\right)} \quad (Eq\ 9)$$

Pressure characteristics at node a

Assume there is no pressure loss from the oil flow travel through the cylinder. The initial static pressure of the system as a whole is the following:

$$p_{initial} = 170\ bar$$

Calculate pressure related tothe present oil flow in terms of the Bernoulli equation obtained from (Brautaset 1983), where no friction effect yet applies at node a:

$$p = p_{initial} - \rho(rho) * \frac{v^2}{2} \quad (Reworked\ from\ eq\ 12)$$

$$p_{node\ a,time=50\ ms} = 170\ bar - \left(\left(865\frac{kg}{m^3}\right) * \frac{\left(14.84\frac{m}{s}\right)^2}{2}\right) * 10^{-5} = 169.05\ bar$$

$$p_{node\ a,time=100\ ms} = 170\ bar - \left(\left(865\frac{kg}{m^3}\right) * \frac{\left(14.50\frac{m}{s}\right)^2}{2}\right) * 10^{-5} = 169.09\ bar$$

$$p_{node\ a,time=150\ ms} = 170\ bar - \left(\left(865\frac{kg}{m^3}\right) * \frac{(14.07\frac{m}{s})^2}{2}\right) * 10^{-5} = 169.14\ bar$$

$$p_{node\ a,time=200\ ms} = 170\ bar - \left(\left(865\frac{kg}{m^3}\right) * \frac{(13.48\ m/s)^2}{2}\right) * 10^{-5} = 169.21\ bar$$

$$p_{node\ a,time=250\ ms} = 170\ bar - \left(\left(865\frac{kg}{m^3}\right) * \frac{(12.48\ m/s)^2}{2}\right) * 10^{-5} = 169.32\ bar$$

According to the Bernoulli equation the variations in oil flow during the release sequence is not sufficient to cause any significant changes to the pressure conditions. When calculating the pressure for the further nodes, it will only be performed with regards to the frictional energy loss that the inlet oil has been exposed to at the time being of each measurement. Note that all further values of the flow for each given time interval are obtained from the oil flow calculated in Excel document for calculations related to this chapter, where flow values at each 5 ms interval is specified. For further reference to that Excel document, see appendix A1.1.

Pressure characteristics, node b

Calculate volume of elbow, element 1, by obtaining cross-sectional area and calculated length from previous calculation in relation to element 1:

$$V_{element\ 1} = l * A$$

$$A_{element\ 1} = \frac{\pi * d^2}{4} = \frac{\pi * 13^2}{4} = 132{,}73\ mm^2$$

$$V_{element\ 1} = 47.125 * 132.73 = 6\ 255\ mm^3 = 0.006\ L$$

Volume displacement at t = 0 to t = 50 ms:

$$V_{displacement,t=50\ ms} = Q * t = \frac{(120.44\ L + 118.16\ L)}{2 * 60\ s} * 0.05\ s = 0.099\ L$$

Hence; all inlet oil at t = 50 ms has passed through entire element 1, and we can assume frictional forces and pressure drop thereafter.

Pressure at node b, t = 50 ms:

$$p_{node\ b,time=50\ ms} = p_{node\ a,t=50\ ms} - p_{loss,1} = 169.05\ bar - 1.17\ bar = 167.88\ bar$$

$$p_{node\ b,time=100\ ms} = p_{node\ a,t=100\ ms} - p_{loss,1} = 169.09\ bar - 1.17\ bar = 167.92\ bar$$

$$p_{node\ b,time=150\ ms} = p_{node\ a,t=150\ ms} - p_{loss,1} = 169.14\ bar - 1.17\ bar = 167.97\ bar$$

$$p_{node\ b,time=200\ ms} = p_{node\ a,t=200\ ms} - p_{loss,1} = 169.21\ bar - 1.17\ bar = 168.04\ bar$$

$$p_{node\ b,time=250\ ms} = p_{node\ a,t=250\ ms} - p_{loss,1} = 169.32\ bar - 1.17\ bar = 168.15\ bar$$

Even if taking the small pressure differences due to the oil flow variations into account, the pressure will remain constant at around 167-168 bar at node 2 during the release sequence.

<u>Pressure characteristics, node c</u>

Sum volume of element 1 and 2, where volume of element 2 can be calculated from the already known duct cross sectional area and previously calculated element length:

$$V_{element\ 2} = l * A$$

$$V_{element\ 2} = 500\ mm * 132.73\ mm^2 = 66\ 365\ mm^3 = 0.066\ L$$

$$V_{element\ 1+2} = 0.006\ L + 0.066\ L = 0.072\ L$$

Since the displaced oil volume at t =50 ms was calculated to 0.099 L, we can assume practically constant pressure conditions during the whole release sequence at node c.
Pressure at node c at every given time instant can then be calculated:

$$p_{node\ c} = p_{node\ a} - p_{loss\ 1} - p_{loss\ 2}$$

$$p_{node\ c,time=50\ ms} = 169.05\ bar - 1.17\ bar - 1.32\ bar = 166.56\ bar$$

$$p_{node\ c,time=100\ ms} = 169.09\ bar - 1.17\ bar - 1.32\ bar = 166.6\ bar$$

$$p_{node\ c,time=150\ ms} = 169.14\ bar - 1.17\ bar - 1.32\ bar = 166.65\ bar$$
$$p_{node\ c,time=200\ ms} = 169.21\ bar - 1.17\ bar - 1.32\ bar = 166.72\ bar$$
$$p_{node\ c,time=250\ ms} = 169.32\ bar - 1.17\ bar - 1.32\ bar = 166.83\ bar$$

Pressure at node c remains at 166-167 bar for the entire release sequence.

Pressure characteristics, node d

Sum volume of element 1, 2 and 3:

$$V_{element\ 1+2+3} = 0.072\ L + 0.006\ L = 0.078\ L$$

$$0.078\ L < V_{displacement,t=50\ ms} = 0.099\ L$$

$$\rightarrow same\ frictional\ conditions\ during\ entire\ time\ interval, since\ oil\ at\ node\ d, at\ t = 50\ ms\ is\ was\ located\ in\ cylinder\ at\ t = 0.$$

Pressure at node d, t = 50 ms to t = 250 ms:

$$p_{node\ d} = p_{node\ a} - p_{loss\ 3} - p_{loss\ 2} - p_{loss,1}$$
$$p_{node\ d,t=50\ ms} = 169.05\ bar - 1.17\ bar - 1.32\ bar - 1.17\ bar = 165.39\ bar$$
$$p_{node\ d,t=100\ ms} = 169.09\ bar - 1.17\ bar - 1.32\ bar - 1.17\ bar = 165.43\ bar$$
$$p_{node\ d,t=150\ ms} = 169.14\ bar - 1.17\ bar - 1.32\ bar - 1.17\ bar = 165.48\ bar$$
$$p_{node\ d,t=200\ ms} = 169.21\ bar - 1.17\ bar - 1.32\ bar - 1.17\ bar = 165.55\ bar$$
$$p_{node\ d,t=250\ ms} = 169.32\ bar - 1.17\ bar - 1.32\ bar - 1.17\ bar = 165.66\ bar$$

Pressure characteristics, node e

Volume, element 4:

$$V_{element\ 4} = 2000\ mm * 132.73\ mm^2 = 265\ 460\ mm^3 = 0.266\ L$$

At t = 0 ms, the oil present at node e, at t = 50 ms was in element 4, since:

$$0.266\ L > V_{displacement,t=50\ ms} = 0.099\ L$$

Calculate volume partition of element 4 to be neglected:

$$V_{diff} = 0.266\ L - 0.099\ L = 0.167\ L$$
$$V_{partition} = \frac{0.167\ L}{\frac{0.266\ L}{100}} = 62.78\ \% = 0.63$$

Calculate pressure at node e, at t = 50 ms:

$$p_{node\ e,t=50\ ms} = p_{node\ a,t=50\ ms} - p_{loss\ 4} * \left(1 - V_{partition}\right)$$
$$p_{node\ e,t=50\ ms} = 169.05\ bar - 5.3\ bar * (1 - 0.63) = 167.09\ bar$$

Volume displacement at t = 0 to t = 100 ms:

$$V_{displacement} = Q * t = \frac{(120.44\ L + 115.44\ L)}{2 * 60\ s} * 0.1\ s = 0.2\ L$$

At t = 0 ms, the oil present at node e, at t = 100 ms was still in element 4, since volume displacement is 0.2 l.

Calculate volume partition of element 4 to be neglected:

$$V_{diff} = 0.266\ L - 0.2\ L = 0.066\ L$$

$$V_{partition} = \frac{0.066\ L}{\frac{0.266\ L}{100}} = 24.81\ \% = 0.25$$

Calculate pressure at node e, at t = 100 ms:

$$p_{node\ e,t=100\ ms} = p_{node\ a,t=100\ ms} - p_{loss\ 4} * \left(1 - V_{partition}\right)$$

$$p_{node\ e,t=100\ ms} = 169.09\ bar - 5.3\ bar * (1 - 0.25) = 165.12\ bar$$

Volume displacement at t = 0 to t = 150 ms:

$$V_{displacement} = Q * t = \frac{(120.44\ L + 112.06\ L)}{2 * 60\ s} * 0.15\ s = 0.29\ L$$

Find the initial location at t = 0 for oil present at node e, at t = 150 ms, by summation of element 4, 3 and 2, with volume displacement subtracted:

$$V_{diff} = (0.266\ L + 0.006\ L + 0.066\ L) - 0.29\ L = 0.048\ L$$

$$\rightarrow Oil\ located\ in\ element\ 2$$

Calculate volume partition of element 2 to be neglected:

$$V_{partition} = \frac{0.048\ L}{\frac{0.066\ L}{100}} = 72.73\ \% = 0.72$$

Calculate pressure at node e, at t = 150 ms:

$$p_{node\ e,t=150\ ms} = p_{node\ a,t=150\ ms} - p_{loss\ 4} - p_{loss\ 3} - p_{loss\ 2} * \left(1 - V_{partition}\right)$$

$$p_{node\ e,t=150\ ms} = 169.14\ bar - 5.3\ bar - 1.17\ bar - 1.32\ bar * (1 - 0.72)$$

$$p_{node\ e,t=150\ ms} = 162.30\ bar$$

Find volume displacement at t = 0 to t = 200 ms:

$$V_{displacement} = Q * t = \frac{(120.44\ L + 107.32\ L)}{2 * 60\ s} * 0.2\ s = 0.38\ L$$

Sum volume of element 1, 2, 3 and 4:

$$V_{element\ 1+2+3+4} = 0.078\ L + 0.266\ L = 0.344\ L$$

$$0.344\ L < 0.38\ L\ \rightarrow oil\ at\ t = 0\ is\ located\ in\ cylinder$$

Calculate pressure at node e, at t = 200 ms and t = 250 ms:

$$p_{node\ e,t=200\ ms} = 169.21\ bar - 5.3\ bar - 1.17\ bar - 1.32\ bar - 1.17\ bar = 160.56\ bar$$

$$p_{node\ e,t=250\ ms} = 169.32\ bar - 5.3\ bar - 1.17\ bar - 1.32\ bar - 1.17\ bar = 160.67\ bar$$

<u>Pressure characteristics, node f</u>

Volume, element 5+6:

$$V_{element\ 5+6} = (50\ mm + 200\ mm) * 132.73mm^2 = 33\ 183\ mm^3 = 0.033\ l$$

$$V_{displacement,t=50\ ms} - V_{element\ 5+6} = 0.099\ L - 0.033\ L = 0.066\ L$$

At t = 0 ms, the oil present at node f, at t = 50 ms was located in element 4.

Calculate volume partition of element 4 to be neglected:

$$V_{diff} = 0.266\ L - 0.066\ L = 0.2\ L$$

$$V_{partition} = \frac{0.2\ L}{\frac{0.266\ L}{100}} = 75.19\ \% = 0.75$$

Calculate pressure at node f, at t = 50 ms, and assume that the flow passing T-junction is only heading towards node f (hence; $p_{loss\ 5,\ case\ 1}$):

$$p_{node\ f,t=50\ ms} = p_{node\ a,t=50\ ms} - p_{loss\ 6} - p_{loss\ 5,case\ 1} - p_{loss\ 4} * \left(1 - V_{partition}\right)$$

$$p_{node\ f,t=50\ ms} = 169.05\ bar - 0.53\ bar - 0.13\ bar - 5.3\ bar * (1 - 0.75) = 167.07\ bar$$

Find the location of oil at node e, at t = 100 ms when t = 0:

$$V_{displacement,t=100\ ms} - V_{element\ 5+6} = 0.2\ L - 0.033\ L = 0.167\ L$$

At t = 0 ms, the oil present at node f, at t = 100 ms was still in element 4, since volume displacement is 0.2 l.

Calculate volume partition of element 4 to be neglected:

$$V_{diff} = 0.266\ L - 0.167\ L = 0.099\ L$$

$$V_{partition} = \frac{0.099\ L}{\frac{0.266\ L}{100}} = 37.22\ \% = 0.37$$

Calculate pressure at node f, at t = 100 ms:

$$p_{node\ f,t=100\ ms} = p_{node\ a,t=100\ ms} - p_{loss\ 6} - p_{loss\ 5,case\ 1} - p_{loss\ 4} * \left(1 - V_{partition}\right)$$

$$p_{node\ f,t=100\ ms} = 169.09\ bar - 0.53\ bar - 0.13\ bar - 5.3\ bar * (1 - 0.37)$$

$$p_{node\ f,t=100\ ms} = 165.09\ bar$$

Volume displacement at t = 0 to t = 150 ms:

$$V_{displacement} = Q * t = \frac{(120.44\ L + 112.06\ L)}{2 * 60\ s} * 0.15\ s = 0.29\ L$$

$$V_{displacement,t=100\ ms} - V_{element\ 5+6} = 0.29\ L - 0.033\ L = 0.257\ L$$

Find the initial location at t = 0 for oil present at node f, at t = 150 ms:

$$V_{diff} = 0.266\ L - 0.257\ L = 0.009\ L$$

$$\rightarrow Oil\ located\ in\ element\ 4$$

Calculate volume partition of element 4 to be neglected:

$$V_{partition} = \frac{0.009\ L}{\frac{0.266\ L}{100}} = 3.38\ \% = 0.03$$

Calculate pressure at node f, at t = 150 ms:

$$p_{node\ f,t=150\ ms} = p_{node\ a,t=150\ ms} - p_{loss\ 6} - p_{loss\ 5,case\ 1} - p_{loss\ 4} * V_{partition}$$

$$p_{node\ f,t=150\ ms} = 169.14\ bar - 0.53\ bar - 0.13\ bar - 5.3\ bar * (1 - 0.03)$$

$$p_{node\ f,t=150\ ms} = 163.34\ bar$$

Find volume displacement at t= 0 to t = 200 ms:

$$V_{displacement} = Q * t = \frac{(120.44\ L + 107.32\ L)}{2 * 60\ s} * 0.2\ s = 0.38\ L$$

Sum volume of element 2, 3, 4, 5 and 6:

$$V_{elements,total} = 0.377\ L$$

$$0.377\ L < 0.38\ L \ \rightarrow measured\ oil\ at\ t = 0\ located\ in\ cylinder$$

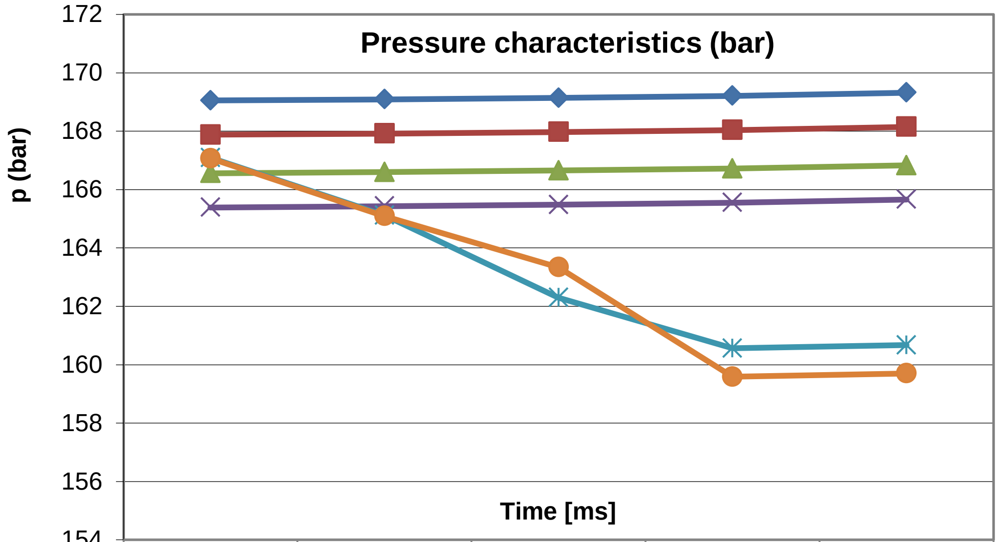
Calculate further pressures at node f:

$$p_{node\ f,t>200ms} = p_{node\ a,t>200ms} - p_{loss\ 6} - p_{loss\ 5,case\ 1} - p_{loss\ 4} - p_{loss\ 3} - p_{loss\ 2} - p_{loss\ 1}$$

$$p_{node\ f,t=200\ ms} = 169.21\ bar - 0.53\ bar - 0.13\ bar - 5.3\ bar - 1.17\ bar - 1.32\ bar - 1.17\ bar = 159.59\ bar$$

$$p_{node\,f,t=250\,ms} = 169.32\,bar - 0.53\,bar - 0.13\,bar - 5.3\,bar - 1.17\,bar - 1.32\,bar - 1.17\,bar = 159.7\,bar$$

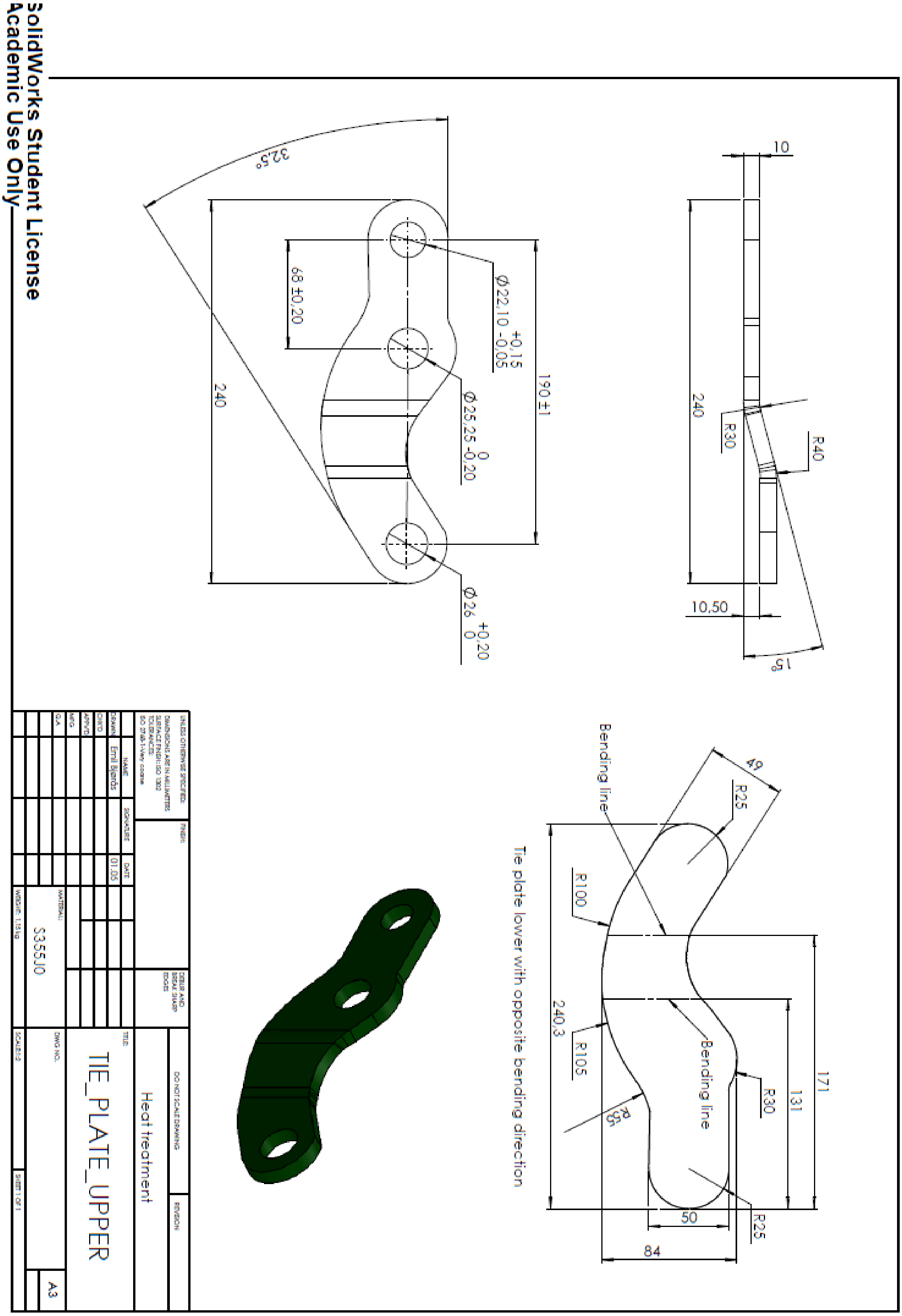
This calculation is carried out under the assumption that we have the designated oil flow without accounting for possible disturbances in the calculation from the behavior of the valve and accumulator in the system. The pressures at the end are decreasing due to the previously calculated pressure loss along the pipeline.



	50	100	150	200	250
Node a	169,05	169,09	169,14	169,21	169,32
Node b	167,88	167,92	167,97	168,04	168,15
Node c	166,56	166,6	166,65	166,72	166,83
Node d	165,39	165,43	165,48	165,55	165,66
Node e	167,09	165,12	162,3	160,56	160,67
Node f	167,07	165,09	163,34	159,59	159,7

Figure A.12: *This chart displays the pressure calculated for every 50 ms time interval at all nodes.*

A3: Technical drawing – Tie plate



A4: Project plan

PROJECT PLAN

Week	January	February	March	April	May
Project specification/planning	20.01.2013				
Research/project introduction	03.02.2013				
Product specification	03.02.2013				
Concept phase		17.02.2013			
Design phase			31.03.2013		
• *Mechanical design*			31.03.2013		
• *Hydro-pneumatic system*			31.03.2013		
• *Mechatronics (outsourced)*			31.03.2013		
• *Final design*			31.03.2013		
Function test			05.04.2013		
Results interpretation and design revision				28.04.2013	
Process-/product evaluation and conclusion					10.05.2013
Finalizing of report					15.05.2013