TAMPEREEN TEKNILLINEN YLIOPISTO
TAMPERE UNIVERSITY OF TECHNOLOGY

JAAKKO GRÖNLUND

ENDURANCE TEST OF HYDRAULIC PISTON AND ROD SEALS

Master of Science Thesis

Examiner: Professor Kari Koskinen
Examiner and topic approved by the Faculty Council of the Faculty of Engineering Sciences on 7th October 2015

TIIVISTELMÄ

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Lentokoneen hydraulijärjestelmässä on havaittu dynaamisten ja staattisten tiivistimien kestävyyden vaihtelua. Tämän uskotaan johtuvan vaihteluista materiaalissa ja laatueroista eri valmistajien välillä. Näiden vaihtelujen selvittämiseksi ja todentamiseksi suoritettiin sarja vertailevia testejä erityisesti tätä tarkoitusta varten valmistetulla testi-laitteistolla. Tämän diplomityön keskeisin tarkoitus oli suorittaa testit tiivistimille ja tulkita tuloksia.

Testijärjestelmä koostui hydraulisesti käytettävästä testisylinteristä, jolla oli mahdollista testata sekä staattisia että dynaamisia tiivistimiä. Testisylinteri oli suunniteltu niin, että sillä voitiin testata kahta tiivistinjärjestelmää samanaikaisesti. Testi suoritettiin varsinaisia käyttöolo-suhteita vastaavissa oloissa. Keskeinen asia oli, että paine, lämpötila ja liikenopeus vastasivat mahdollisimman hyvin käyttöä alkuperäisessä sovelluksessa. Ohjauslogiikkaa käytettiin testin sylinterin jatkuvaksi liikuttamiseksi edestakaisin.

Testauksen aikana sylinterin liikuttamiseen tarvittavaa voimaa monitoroitiin. Voimasta pystyttiin laskemaan kitka ja kitkakäyttäytymisen muutos dynaamisessa tiivistimessä testauksen aikana. Vuotoa seurattiin sekä dynaamisessa että staattisessa tiivistimessä. Painetta, asemaa ja lämpötilaa seurattiin testin aikana. Asemasta saatiin laskettua hetkellinen liikenopeus testisylinterille. Järjestelmä oli toteutettu niin, että kahta männäntiivistintä, kahta varrentiivistintä sekä kahta O-rengasta voitiin testata samaan aikaan. Sekä männän että varren tiivistimet olivat T-tiivistimiä. T-tiivistin koostuu t:n muotoisesta elastomeerisestä tiivistinelementistä, jota tukevat tukirenkaat molemmilta puolin. Tiivistimien kuluminen määritettiin painon muutoksena, kun verrattiin tiivistimen painoa testin jälkeen testiä edeltävään painoon. Mahdollinen nesteen imeytymisen vaikutus tiivistimen massaan ja ominaisuuksiin selvitettiin erillisellä testillä.

Tiivistimien ominaisuudet valmistajien välillä poikkesivat tässä tutkimuksessa ja valmistajan 2 tiivistimet kestivät paremmin kulutusta kuin valmistajan 1. Tämä tulos perustuu vuotoon testin aikana, punnitukseen testin jälkeen ja kitkakäytökseen testin aikana. Myös visuaalinen tiivistinten tarkastelu tukee tätä havaintoa.

ABSTRACT

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There is observed deviation of endurance in dynamic and static sealings in the hydraulic system of an airplane which is suspected to be caused by differences in material and quality between different manufacturers. In order to find out and verify these differences series of comparing tests were conducted on specifically made test equipment. The main goal of this Master's Thesis was to perform the needed tests on the seals and to evaluate the test results.

The test system consisted of a hydraulically driven test cylinder which was able to test both the dynamic and static seal. The test cylinder was designed in such a way that it could be used to test two sealing systems at the same time. The test was conducted in conditions that resembled real life usage. The main points were that pressure, temperature and movement speed were the same as in the original application. Control logic was used to run the test cylinder continuously from one end to the other.

During the test the force needed to move the cylinder was monitored. From this force the friction and the deviation of friction behavior in dynamic seal during the test were computed. Leakage was monitored both on the dynamic seal and on the static seal. Pressures, position and temperature were also used as survey measures. Speed of the test cylinder was calculated from position. The system allowed testing of two piston seals, two rod seals and two O-rings simultaneously. Both piston seal and rod seal were T-seals. T-seals consist of a T-shaped elastomer sealing element supported by back-up rings on both sides. The wearing of seals was determined by change of weight before and after the experiment. The possible effect of absorption of fluid on the mass and properties of the sealing was examined with separate tests.

The properties of the seals between the two manufacturers differed in the present study and the seals of manufacturer 2 lasted wearing better than the seals of manufacturer 1. This result is based on the leakage during the test, weighting after the test and friction behavior during the test. Also visual observation of the seals supports this finding.

PREFACE

The work of this Master's thesis has been carried out in the Department of Mechanical Engineering and Industrial Systems at the Tampere University of Technology.

I wish to express my sincere appreciation and gratitude to the department's project manager Jussi Aaltonen for guidance and supervision throughout this Master's thesis. I am also grateful for my examiner Professor Kari Koskinen for his constructive comments.

I am grateful to my family members. They have always been very supportive and encouraged me to pursue and achieve my goals. Special thanks to my father Juhani for helping me with the language and structure of academic writing in this thesis. I also wish to thank my fiancée Susanna for her loving support.

In Tampere, Finland, on the 20th of November 2016

Jaakko Grönlund

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LIST OF SYMBOLS AND ABBREVIATIONS

NBR	Nitrile butadiene rubber
PTFE	Polytetrafluoroethylene
ACN	Acrylonitrile
η	Dynamic viscosity
ρ	Density
ν	Kinematic viscosity
q	Flow
d	Diameter
h	Height
l	Length
p	Pressure

1. INTRODUCTION

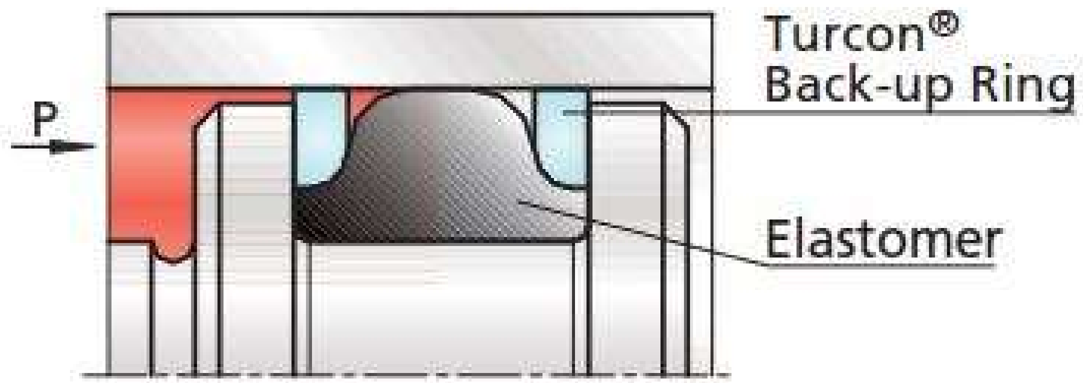
There is observed deviation of endurance in both dynamic and static seals in the hydraulic system of an airplane which is suspected to be caused by differences in material and quality between manufacturers. In order to find out and verify these differences comparing tests were conducted on specifically made test equipment. The main goal of this Master's Thesis was to find out the endurance differences between the seals using the previously designed test equipment. Before the actual test the measurement system and control logic had to be designed. The test process consisted of absorption tests and endurance tests. The test system consisted of a hydraulically driven test cylinder which was able to test the dynamic and static seal. The test was conducted in conditions resembling real life usage of the seals. The main point was that pressure, temperature and movement speed were the same as in the original application and remained the same throughout the testing process.

Kanters and Visscher (1989) state that leakage, friction and wear are the most important practical parameters to classify the performance of a seal. During the test the force needed to move the cylinder was monitored. After test the friction and the deviation of friction behavior in dynamic seal during the test were computed. Leakage was also monitored both on the dynamic seal and on the static seal. Pressures, movement speed and temperature were used as survey measures. The system allowed testing of two piston seals, two rod seals and two O-rings simultaneously. The wearing of seals was determined by change of weight before and after the experiment. This Master´s thesis also aimed to find out information about the chemical resistance of the sealing materials. The possible effect of absorption of fluid on the mass and properties of the sealing was examined with separate tests.

2. RECIPROCATING SEALS IN HYDRAULIC CYLINDERS

According to Koskinen and Aaltonen (2013, p. 1711) Reciprocating seals (sliding seals) are widely used in hydraulic cylinders. There are a wide variety of different reciprocating seal designs available. Even though variation in shapes, material, and functional characteristics is vast the basic operation principle of all reciprocating seals is mostly the same. Koskinen and Aaltonen also state that t-ring is a specially profiled derivative of an O-ring and it is relatively widely used. It is stated that there is a wide selection of applicable materials.

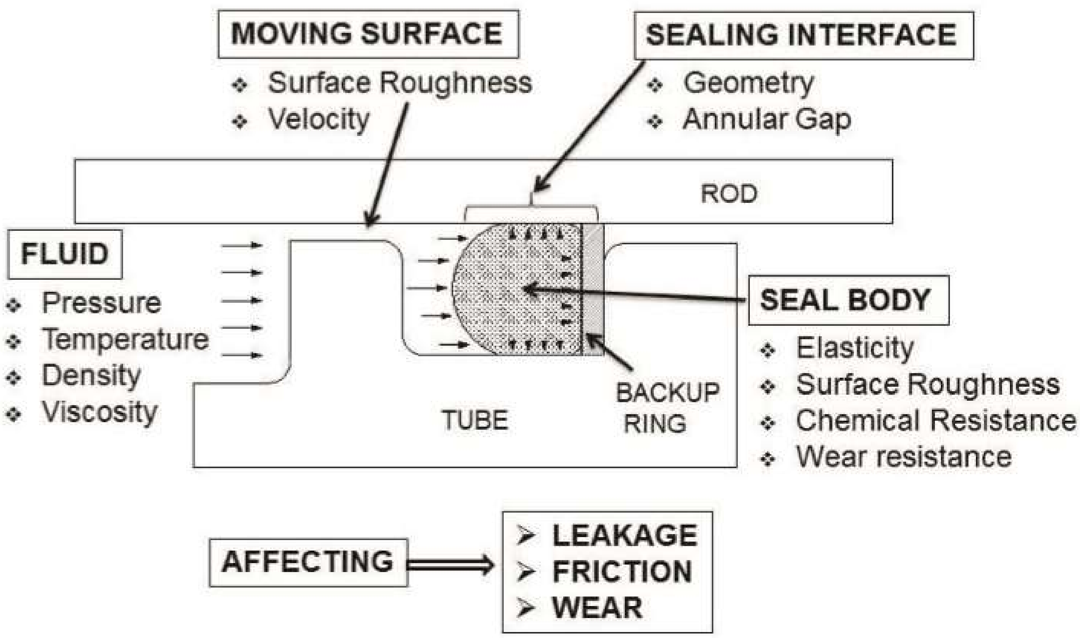
Koskinen and Aaltonen (2013, p. 1713) state that when hydraulic seals are not under pressure they are leak-tight because of their initial compression. Pressure acting on the seal is added to the initial compression and thus surface pressure on seal contact area is always higher than sealing pressure. According to an inverse hydrodynamic theory, surfaces moving relative to each other are pulling fluid film under the contact area and hydrodynamic pressure is built up. This causes the separating force to the surfaces. The thickness of the fluid film depends on the maximum pressure of the pressure field in the contact area, dynamic viscosity of the fluid and the relative speed. The fluid film generated remains on the sliding surface behind the seal and is returned during the return stroke. During the return stroke the same phenomena of the film formation occur. Due to pressure field usually being asymmetrical, the film thickness is different in different directions of movement. Ratio of film thicknesses in different directions is referred as the dynamic tightness. Leakage flow is caused by this difference in film thickness. Leakage flow is inevitable.



Picture 1 Cross section of T-seal. (Aerospace Sealing Systems product Catalog & Engineering Guide 2011, p. 138)

Picture 1 shows a cross section of a T-seal. T-seal consists of a T-shaped elastomer sealing element supported by back-up rings on both sides. This combination results in a stable seal and the semicircular lip configuration ensures positive sealing. The side flanges, which form the seal’s base offer excellent resistance to rolling and act as an effective platform to position and energize the anti-extrusion rings. According to Trelleborg the T-seal type has an excellent extrusion resistance. That is why T-seal is the preferred option for media separation as in this test system. (Aerospace Sealing Systems product Catalog & Engineering Guide 2011, p. 137)

Bhaumik et al (2013, p. 835) state that the sealing performance is affected by type of fluid, seal material, interacting surface and geometry of sealing interface as shown in picture 2.



Picture 2 Factors affecting sealing performance. (Bhaumik et al. 2013, p. 836)

Picture 2 shows what kind of factors have an effect on the leakage, friction and wear of the seal. The things that can separate the seals of different manufacturers in this test are the properties of the seals itself. The main properties are elasticity, surface roughness, chemical resistance and wear resistance. Also the seal geometries can vary slightly between manufacturers and thus the annular gap may differ.

3. BACKGROUND RESEARCH

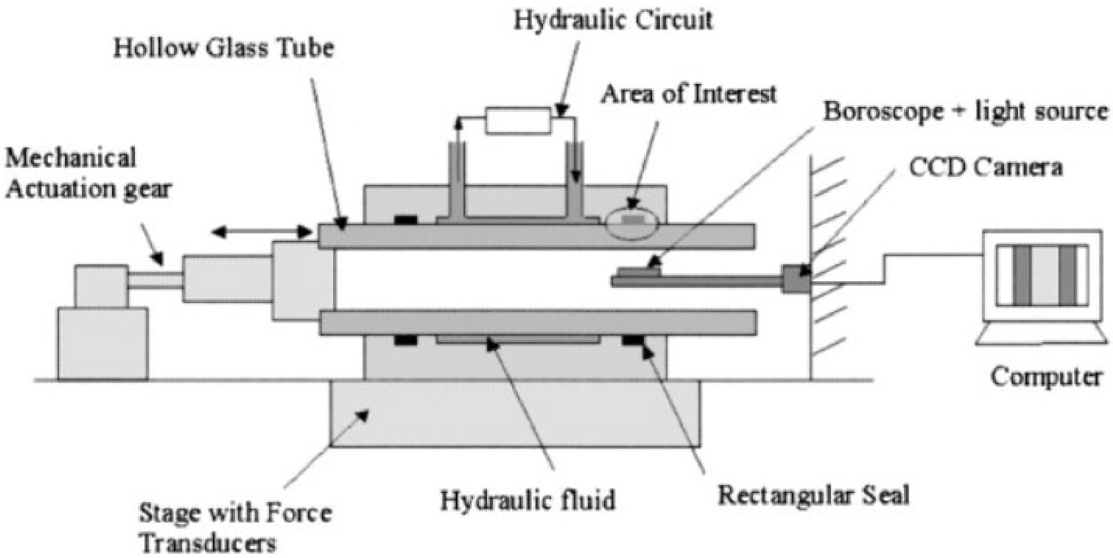
Nikas (2010) state that hydraulic reciprocating seals are critical machine elements used in a variety of industrial, automobile, aerospace and medical applications that involve linear and rotational motion such as in hydraulic actuators.

3.1 Hydraulic seal testing

Nikas (2010) states that the research on hydraulic reciprocating seals has started roughly in the 1930s. Since then the research has achieved a basic understanding of performance issues related to seals. The Work of White and Denny between 1944 and 1946 is probably the first major experimental research contribution on seals. White and Denny studied seals of various design and material. They conducted tests on different pressures, speeds and temperatures and measured friction force, leakage rate, seal wear and failure mechanism involving abrasion and extrusion. A hydraulic reciprocating seal is a neglected machine element in the scientific literature, in spite of its vital role in many applications. The neglect is partly attributed to the complexity of seal behavior. The difficulty is seals flexibility, which precludes obtaining analytical solutions and complicates any numerical solution processes. Moreover typical seal material such as elastomers obey highly complex non-linear stress-strain laws of finite elasticity or thermoviscoelasticity, which are strongly affected by temperature. Basic mechanical properties of seals such as the moduli of elasticity and rigidity, poisons ratio, hardness and compressibility all depends strongly on temperature. Additional influential factors such as chemical interaction with hydraulic fluids, material oxidation, and ageing play major roles in sealing performance. In spite of the difficulties in sealing performance evaluation seals are met in many critical applications with machinery costing hundreds to millions times more than the seals. A characteristic example was the destruction of the NASA space shuttle Challenger in 1986 which was attributed to the loss of sealing ability of a static elastomeric O-ring because of a low ambient temperature the night before the launch. That was and engineering error which cost several human lives. That is why the importance of correct engineering design and evaluation of hydraulic seals cannot be underestimated to avoid costly mistakes.

Seal study of Rana and Sayles (2005) was motivated by the aerospace industry where the seals are used in aircraft actuators. The actuators control several types of aircraft operation systems, such as the landing gears, wing flap controls, suspension systems and other aircraft utility systems. These seals demand very reliable and continuous operation through their entire lifetime through some of the most extreme physical conditions such as low temperatures experienced during flight and high temperatures for actuators in the proximity of aircraft engines. A point of great importance is to ensure that there is no loss of hydraulic fluid into the atmosphere both while the aircraft is static on the runway and airborne. This requires that the seal has maximum reliability under a wide range of operating conditions.

Rana and Sayles (2005) state that there have been several numerical studies of reciprocating seals under real operating conditions but relatively very little experimental work has been carried out on seals. A fuller understanding of seals is still unclear given the lack of experimental work using modern experimental techniques. Friction and lubrication data exist only for very simple loading and geometry cases of the seals. Real seals experience much higher loads and operate in confined spaces. Rana and Sayles constructed an experimental rig, which simulated realistic geometric and operating conditions experienced by the seal that was used. The rig provided friction data while at the same time valuable optical information on the seals operation was provided.



Picture 3 Seal test rig by Rana and Sayles. (2005)

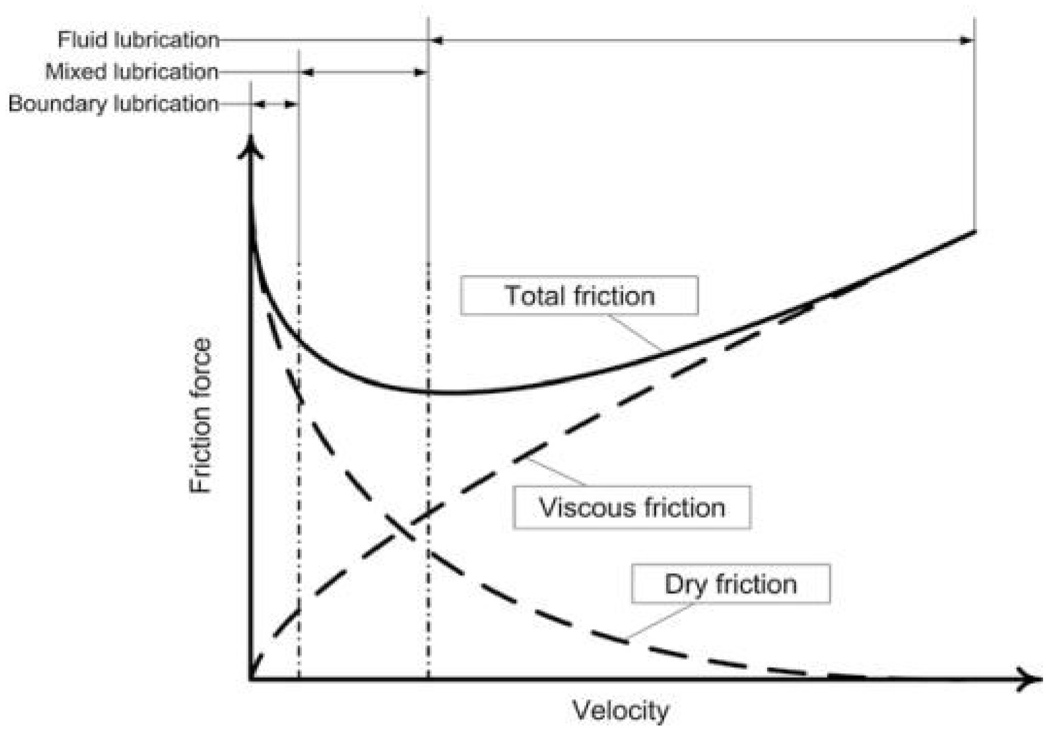
The seal test system by Rana and Sayles (2005) is shown in picture 3. The rig used glass piston instead of a steel one. Instead of a fluid pressure to move the piston they used an external set of gears and motors attached to the piston which drives the glass piston. This kind of setup simulates the actuation cycles of a real actuator while simulating the high fluid pressure a seal experiences in a sliding contact. The hollow glass piston allows the observation of seal to glass contact with a rigid boroscope. Friction force is measured by piezoelectric transducers which measure the combined frictional resistance from both of the seals when the glass shaft is moved.

3.2 Properties of NBR and PTFE in seals

One part of this Master's study was to examine the hydraulic fluid absorption of the used sealing materials. All tested seals were made of nitrile butadiene rubber which is more commonly known as NBR. Support rings of rod and pistons seals are made of Virgin polytetrafluoroethylene (PTFE) grades A and B. PTFE is more commonly known as Teflon.

3.2.1 Seal friction

According to Koskinen and Aaltonen (2013, p. 1713) seal friction is the most important factor in overall friction force formed in a hydraulic cylinder. For optimum function, seal elements should allow such fluid film thickness under them that friction, the level of leakage, and transport of contaminants are simultaneously at an acceptable level. These three Phenomenon's all are dependent from the fluid film thickness. For example if the fluid film is thicker larger contaminants may travel in the film. Thicker fluid film also results in larger leakage. Lubrication in a seal contact generally follows three basic lubrication phenomena: fluid (hydrodynamic) lubrication, mixed lubrication, and boundary lubrication. These different types of lubrications effect on friction can be seen on picture 3. Koskinen and Aaltonen also state that the formation of this previously mentioned hydrodynamic lubricating film is influenced by the structure of counter surface and properties of hydraulic fluid. Also a stick-slip phenomena may occur at slow speeds. Stick-slip is a friction-borne vibration which is caused by an alternating static and Coulomb force.



Picture 4 Friction forces relation to velocity. (Koskinen & Aaltonen 2013, p. 1713)

According to Koskinen and Aaltonen (2013, p. 1714) friction in the seal contact is primarily influenced by the thickness of the lubricating film and characteristics of surfaces. At zero speed the dominant friction is static friction, which is primarily dependant of charasteristics of surfaces. As speed increases, the boundary friction becomes the dominant factor, and finally as the speed has increased enough for high enough hydrodynamic pressure to form, the dominant friction term is viscous friction. In picture 4 the dry friction curve presents the boundary friction. Boundary friction occurs when surface is not so lubricated that there would be no contact between the moving surfaces.

Rana and Sayles (2005) state that the stroke needs to be longer than the width of the seal for viscous friction to form. In this case the stroke is much longer so viscous friction can be expected.

3.2.2 Chemical and wear resistance

Knowledge of chemical resistance of the seals used is crucial in explaining the behavior of the seals. According to Nikas and Sayles (2006) elastomeric seals can suffer from swelling due to fluid absorption or react chemically with hydraulic fluids. Nikas (2008) states that swelling from fluid absorption affects sealing performance. The change in sealing performance is due to change in sealing dimensions which affects the contact pressure between the seal and the counter surface.

Table 1 General properties of NBR by Emerson (Chemical Compatibility of Elastomers and Metals , p. 660)

Property	Nitrile (NBR)
Tensile strenght, Psi (bar) Pure Gum	600 (41)
Tensile strenght, Psi (bar) Reinforced	4000 (276)
Tear Resistance	Fair
Abrasion Resistance	Good
Aging: Sunlight; Oxidation	Poor; Fair
Heat (Maximum Temperature)	250 F (121 C)
Static (Shelf)	Good
Flex Cracking Resistance	Good
Compression Set Resistance	Very Good
Solvent Resistance: Aliphatic hydrocarbon	Good
Aromatic Hydrocarbon	Fair
Oxygenated Solvent	Poor
Halogenated Solvent	Very Poor
Oil Resistance: Low aniline Mineral Oil	Excellent
High Aniline Mineral oil	Excellent
Synthetic Lubricant	Fair
Organic Phosphates	Very Poor
Gasoline Resistance: Aromatic	Good
Non-Aromatic	Excellent
Acid Resistance: Diluted (Under 10%)	Good
Concentrated	Poor
Low Temperature	- 40 F
Flexibility (maximum)	-40 C
Permeability to Gasses	Fair
Water Resistance	Very Good
Alkali Resistance: Dilutes (Under 10%)	Good
Concentrated	Fair
Resilience	Fair
Elongation (Maximum)	500 %

All tables found about NBR materials oil resistance showed that NBR has a good resistance to oil. This is shown in table 1. The fluid used on the test is MIL-PRF-83282 which is more specific a synthetic hydrocarbon based hydraulic fluid. Trelleborg suggests NBR to be used as elastomer material with this fluid. According to Trelleborg (Materials Chemical Compatibility Guide 2012) "The properties of the Nitrile Rubber depend mainly on the ACN content which ranges between 18 % and 50 %. In general they show good mechanical properties. The operating temperatures range between -30 °C/-22 °F and +100 °C/+212 °F (for a short period of time up to +120 °C/+248 °F). Suitable formulated NBR can be used down to -60 °C/-76 °F. NBR is mostly used with mineral based oils and greases." In the real usage of the seals the temperature can rise to a 100 degrees Celsius. Based on this the NBR material should be suitable for the usage. Manufacturers of the test seals do not provide an ACN content of their NBR material so that can result in differences between the test seals. ACN is more commonly known as acrylonitrile.

According to Trelleborg PTFE material exhibit the lowest coefficient of friction of any known solid and have a uniquely low static coefficient friction. This results in an extremely low breakout friction and as the materials do not adhere to their mating surfaces, stick-slip in dynamic applications is eliminated. PTFE material is chemically inert in virtually all media, even at elevated temperature and pressures. They are therefore compatible with an extremely wide range of solvents, acids and other aggressive media. PTFE compounds are available to operate in temperatures from -196 Celsius degrees to +260 Celsius degrees and will endure spiked of up to +360 Celsius degrees. PTFE material are not altered or adversely affected by cycling temperatures. PTFE does not absorb any other media than water to a significant level. One exception is fluorinated-cooling media, for example Freon. These can cause a reversible weight increase. The used virgin PTFE is physiologically inert. (Aerospace Sealing Systems product Catalog & Engineering Guide 2011, p. 18)

4. SEAL TEST SETUPS

The chemical resistance of the seals was tested before the endurance test. The chemical resistance was tested separately to find out if the seals of different manufacturers do react with the used hydraulic fluid differently. The seals of different manufacturers were placed on separate bowls. This was done to make sure that nothing could dissolve from one manufacturer seals to affect the other manufacturers seals test results. For this same reason the chemical resistance test in elevated temperature was done separately for different manufacturers seals. The hydraulic fluid used for the elevated temperature test was changed between the tests for new to ensure the cleanliness of the hydraulic fluid used.

Endurance tests were performed so that on each end of the test cylinder were installed seals of the same manufacturer. This was done to make it possible to see if there is difference in the seals of same manufacturer. This test setup also made it possible to compare the friction results of different manufacturer's seals. If seals of different manufacturers would have been installed on different ends of the test cylinder the fiction behavior of one manufacturer could not have been separated from the other. Also there could not have been certainty from which manufacturers seal does the fluid leak which leaks from the leakage hole in the middle of the cylinder pipe.

4.1 Chemical resistance in room temperature test setup

The first absorption test was carried out in room temperature. The test time was according to standard 72 hours (ISO 1817 2005, p. 5). The seals were on an open aluminum bowl immersed in hydraulic fluid. The fluid is the same that is used on the actual usage of the seals. It is synthetic micro filtered hydraulic fluid and is know more specifically as "MIL-PRF-83282" Bowls were covered with plexiglas to prevent any external material to get in to the fluid during the test.

Picture 5 Setup of absorption test in room temperature

The test equipment used in room temperature test can be seen in picture 5.

4.2 Chemical resistance in elevated temperature test setup

The second absorption test was carried out at elevated temperature to find out does the higher temperature effect on the fluid absorption of the seal. The sealing used in this test was the same that was used in the previous test at room temperature. Absorption test in elevated temperature was carried out by warming the fluid with water bath. The water bath system consisted of an electrical hotplate to boil water in a pot and an open aluminum bowl that was set in the boiling water. The bowl was filled with the test fluid and seals were laid in to the bowl. The test took four hours. The seals of different manufacturers were tested on different days. Temperature changed during the test according to the table 2.

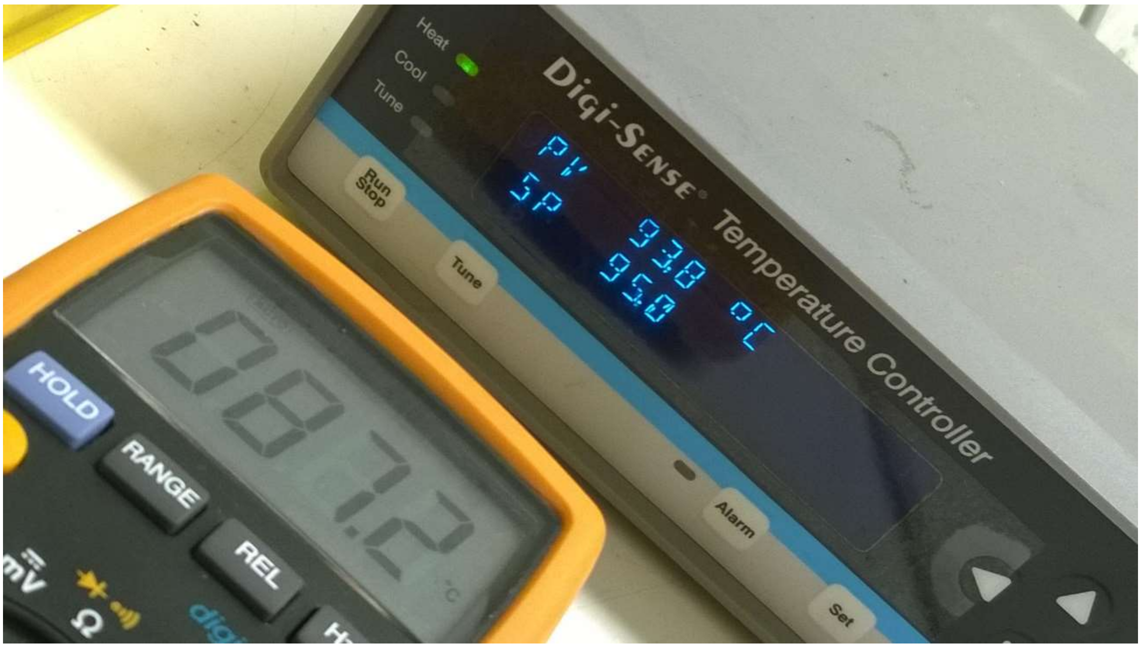
Picture 6 Test setup of elevated temperature test.

The test equipment used in elevated temperature test can be seen in picture 6.

Table 2 Test temperature on absorption test on elevated temperature

Wanted test temperature [C]	90
Temperature in endurance test 1 [C]	85-93,2
Temperature in endurance test 2 [C]	84,4-91,9

Table 2 shows the test temperatures during absorption tests on elevated temperature. In test 1 the temperature changed between 85 and 93,2 degrees Celsius. In test 2 the temperature was between 84,4 and 91,9 degrees Celsius. The differences in maximum and minimum temperature and temperature change cycle times can be explained by the altering amount of water in the pot. The water evaporated during the test and therefore the cycle time became faster and when water was added the cycle time became longer. The maximum temperature in test 1 was higher because of an addition of large water quantity. The hotplate stopped heating when water temperature reached 95 degrees and started heating again on 93 degrees. In test 1 the hotplate stayed on longer and heated more than in test 2 yielding more heating energy and leading to the 93,2 degrees temperature pike. The normal temperature cycle can be seen during the first 60 minutes of the test. After that the temperature values were not taken up constantly but were observed for the whole time of the test.



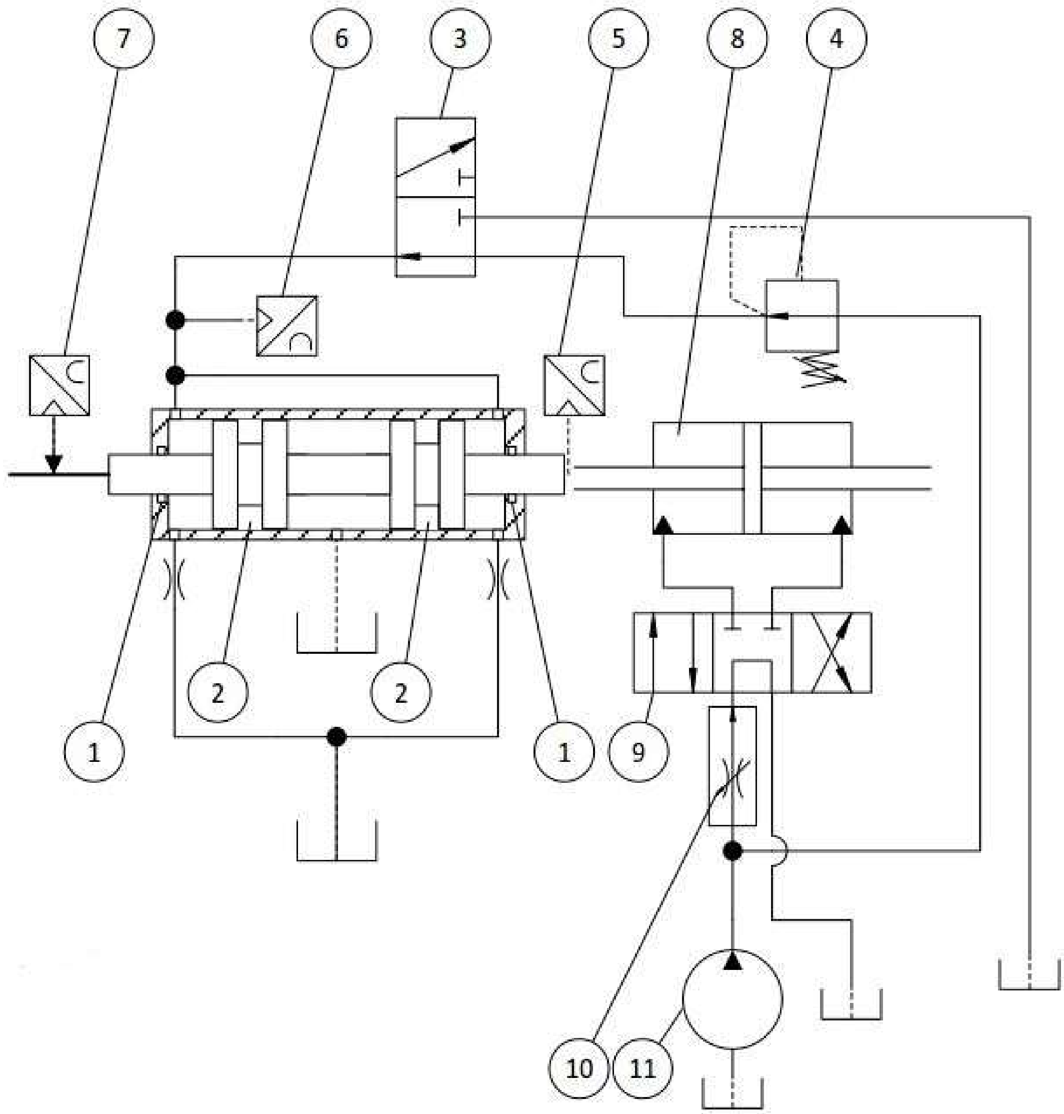
Picture *7* Elevated temperature test temperature observation.

A Digi-Sense Temperature Controller was used to control the temperature of the hotplate. Hydraulic fluid temperature was measured with a P3400 K type thermocouple connected to a digimess HM200 multimeter. These equipments that were used to follow and control the test temperature are shown on picture 7.

4.3 Endurance test system

The test cylinder was designed on the basis of the existing seals. The cylinder was designed to be able to test two sets of sealings at the same time. One set consisted of a piston seal, a rod seal and an O-ring seal. Both piston seal and rod seal were T-seals. T-seals consisted of a T-shaped elastomer sealing element supported by back-up rings on both sides. A suitable cylinder type was the tie rod cylinder. Tie rod cylinder is the used cylinder type in the ISO 7986 standard for seal test methods. Linear actuator is used to move the cylinder in the standard as well as in this test system. The fluid leakage drains are positioned similarly as in the standard. One in each cylinder end and one in the middle of the cylinder pipe. (ISO 7986 1997) The design of test cylinder is accurately described in the bachelor’s thesis by Grönlund (2015).

A test system that was able to test two rod seals at the same time was used in endurance test performed by Larsen et al. (2013, p. 66) The test cylinder was moved by a driving cylinder as in the test done in this thesis project. A test system with a separate driving cylinder was also used by Mao et al. (2012), Nikas et al. (2014) and Bhaumik et al. (2014). Yang (1995) used a crank block mechanism to make a reciprocating movement for cylinder.



Picture 8 Hydraulic circuit of the test system.

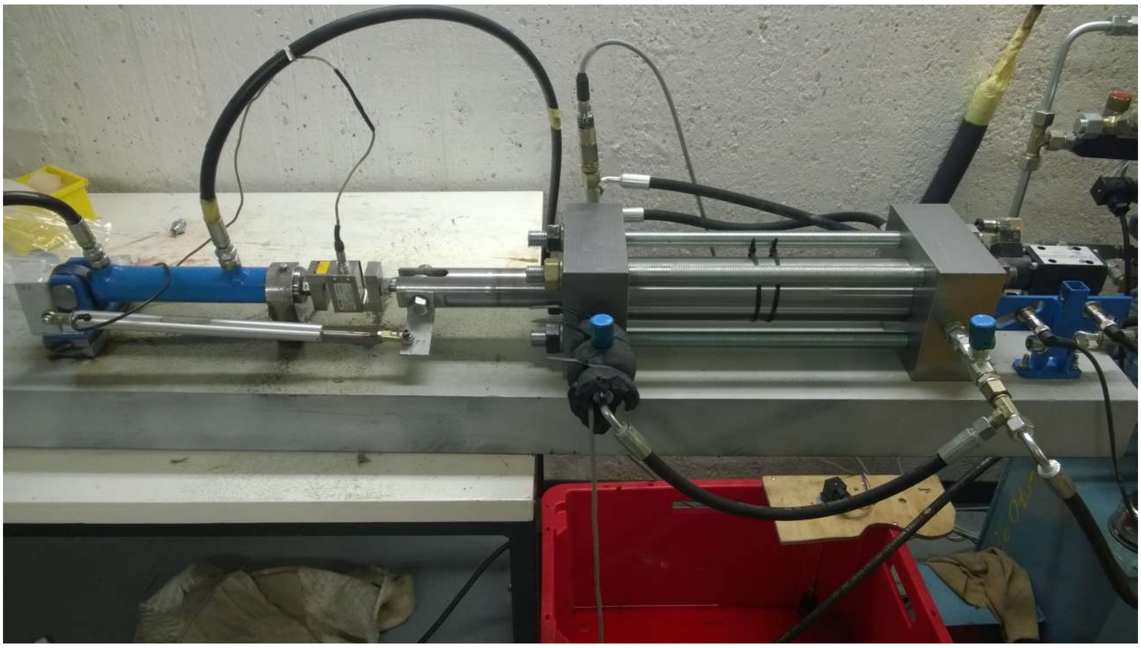
The hydraulic test circuit seen in picture 8 is built on the basis of an existing hydraulic power unit.

Table 3 Components in hydraulic circuit diagram.

1	Rod sealing	
2	Piston sealing	
3	2/2 Directional control valve	
4	Pressure reducing valve	
5	Force transducer	
6	Pressure transducer	
7	Displacement sensor	
8	Driving cylinder	
9	4/3 Directional control valve	
10	2-Way flow control valve	
11	Hydraulic pump	

Table 3 states the components of the hydraulic circuit. From the actual system the pressure reducing valve and the 2-way flow control valve were left out.

Test system consists of a test cylinder, driving cylinder, hydraulic power unit, measurement system and control logic.



Picture 9 Test system.

The picture 9 shows the blue driving cylinder attached to the test cylinder through a force transducer. Along the driving cylinder is a silver colored position transducer that is connected to the test cylinder. On the right side of the test cylinder the induction transducers fixed on a blue rig can be seen.

4.4 Measurement system of the endurance test

Bisztray-Balku (1995) states that the essential operating conditions in an endurance test are the working pressure, speed, temperature and the medium, the friction surface quality and seal-space design. All these conditions are taken in notice when designing the endurance test system for this project.

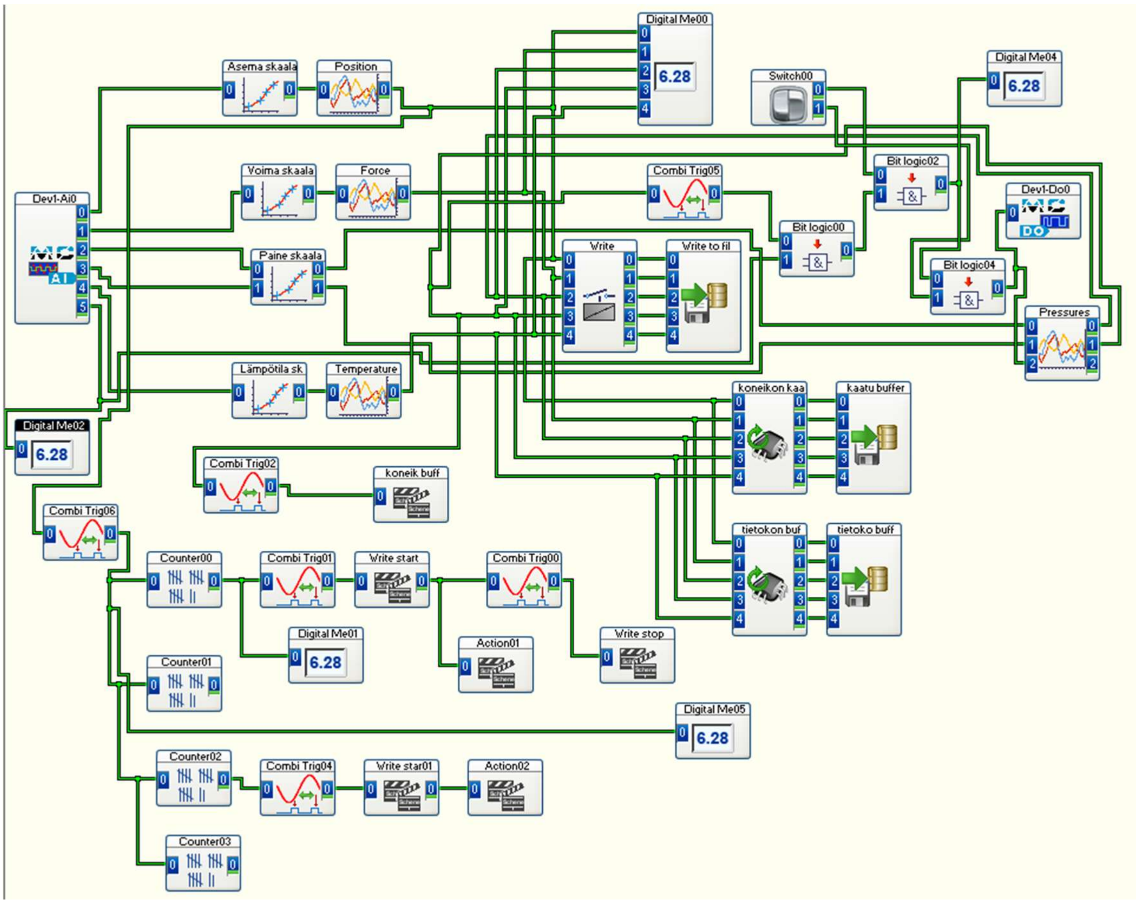
The measuring equipment consists of a force transducer which is attached between the test cylinder and the driving cylinder. The seal friction and the deviation of friction behavior during the test are calculated from the force. Pressure transducers monitor the pressures in the hydraulic circuit. If sudden drop on pressure level is observed the system is automatically stopped because the pressure drop may be caused by a leak. There are two pressure transducers in the system. One pressure transducer measures the supply pressure of the hydraulic power unit and the other measures the test pressure from the fluid inlet of the test cylinder. Displacement sensor measures the position of the cylinder. Movement speed of the cylinder can be calculated from the displacement data. The test temperature needs to be as close as possible to the real usage conditions and it is monitored with a temperature transducer. Leakage container is installed under the leakage holes to catch the leaking fluid. Fluid level sensor is installed in the leakage container so the system will stop if the leakage is too much. This enables the test usage without continuous supervision.

Pressure transducer, force transducer, temperature transducer and position transducer are advised in the standard (ISO 7986 1997) to be used to monitor seal testing.

Table 4 Transducers used on the test system.

Transducer type	Manufacturer	Transducer model
Pressure	trafag	8253.74.2317
Temperature	-	pt 100
Force	FUTEK	LSB350
Position	Penny & Giles Controls Ltd	HLP190SA11506K
Induction	SICK	IMF12-04NNSVC0S

The transducer models and manufacturers that were used on the test system are shown on table 4. A computer program called DASYLab was used to build the data acquisition system. Between transducers and the computer using DASYLab was used a MEASUREMENT COMPUTING™ USB-1208HS-4AO data acquisition device.

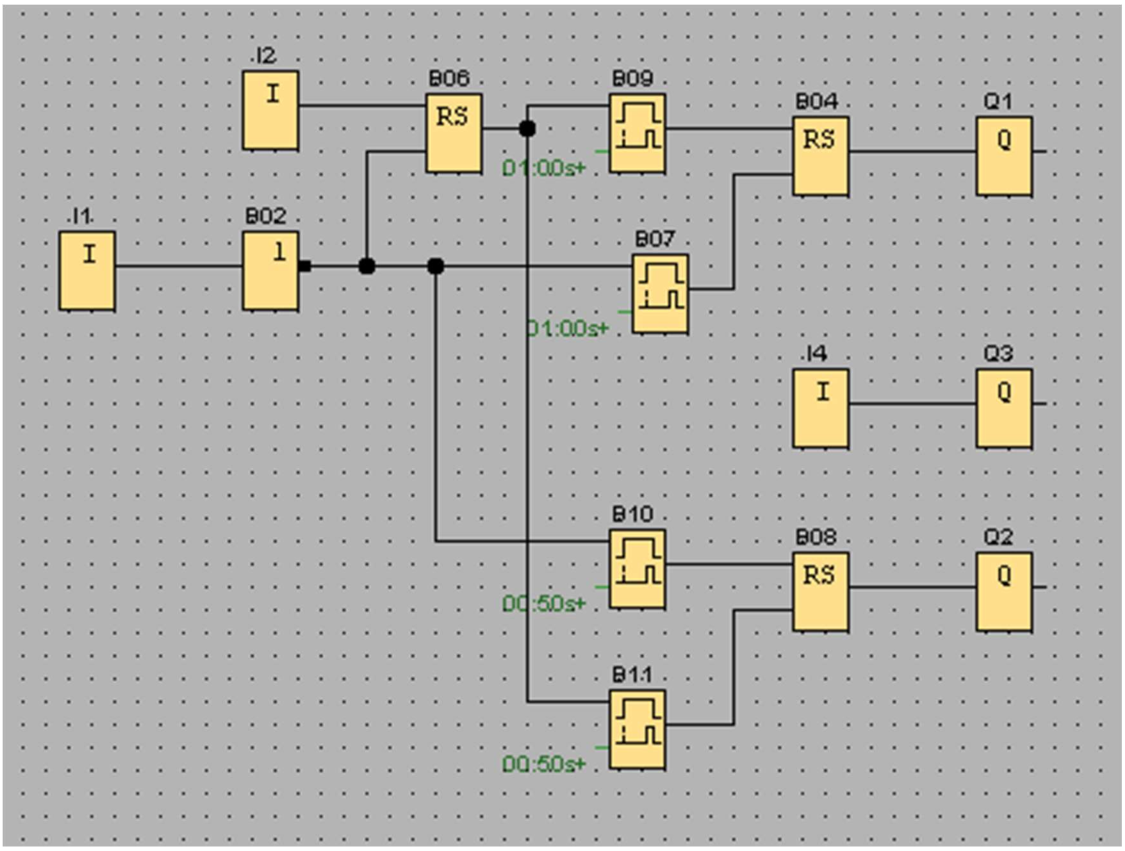


Picture 10 DASYLab program used for data acquisition.

The DASYLab program was used to read data and save it to a file every 1000 cycles. Each file consists of 6 seconds of data from every transducer. The sampling frequency from transducers was 500 Hz. If the hydraulic power unit shuts down the DASYLab will store the last 120 seconds of data to a new file from the buffer. This will happen when system pressure goes under 5 bar and the data is saved to get an idea what has happened before the malfunction. The DASYLab program is shown on picture 10.

A control logic was used to run the test in conditions that resemble real usage conditions. In the test cycle the cylinder is run continuously from one end to the other. Pressure is differentiated so that it resembles normal cylinder usage. Pressure differentiation is done so that when cylinder changes direction the 2/2 directional valve changes position. The reason for this is that the test would resemble better the actual usage conditions of the sealings. Test cylinder is pressurized when it moves to positive direction and it is depressurized when it moves to negative direction. Cylinder's change of direction is done by the 4/3 directional valve. The change of direction occurs when the driving cylinder is fully open or closed. These end positions are monitored with induction transducers. The amount of leak hydraulic fluid is monitored with a liquid level sensor. When the level exceeds certain value the whole system stops for safety reasons. Reaching the designed safety limit stops the operation because it means a big compromising leak. This automatic system allows unobserved use of the test system.

Control logic is built with Siemens LOGO! soft.



Picture 11 Control logic used on the endurance test.

Picture 11 shows the used control logic. Inputs to the logic in picture 11 are signals from the induction transducers and fluid level sensor. I1 and I2 are the induction sensors and I4 is fluid level sensor. Outputs are the power of the hydraulic power unit and control of both directional valves. Q1 and Q2 are control signal of valves and Q3 controls the hydraulic power unit.

5. SEAL TEST RESULTS AND DISCUSSION

Before and after immersing the seals in the hydraulic fluid the seals were weighted using a Precisa EP 420A analytical balance. Readability of the balance was 0.1 mg.

Picture 12 Precisa EP 420A analytical balance.

Picture 12 shows the analytical balance used. After the immersion test the seals were lifted out from the fluid and were laid on a nearly lint free paper towels. After drying for an hour the seals were dried well with paper towels and weighed.

5.1 Absorption test in room temperature

The seals were unused and straightly taken from manufacturer's plastic bags.

Table 5 Weighting results of 72h immersion test at room temperature.

72h Room temperature	Mass before [g]	Mass after [g]	Change of mass percent
Manufacturer 1 rod seal			
Teflon rings	5,004	5,0031	-0,017985612
seal	6,0366	5,9955	-0,680846834
Manufacturer 2 rod seal			
Teflon rings	5,1894	5,194	0,088642232
Seal	5,915	5,892	-0,388841927
Manufacturer 1 Piston seal			
Teflon rings	0,532	0,5327	0,131578947
Seal	1,0265	1,016	-1,022893327
Manufacturer 2 Piston seal			
Teflon rings	0,551	0,5516	0,108892922
seal	0,981	0,97	-1,121304791
Manufacturer 1			
O-ring	0,6094	0,6181	1,427633738
manufacturer 2			
O-ring	0,5411	0,5527	2,143781186

In table 5 the weighting results of 72 hour immersion test performed at room temperature can be seen. The table shows that the mass of all T-seals made of NBR lose slightly their mass. The mass changes of the Teflon rings were insignificantly small. On the contrary to the T-seals the O-rings gained mass on the immersion test. Also the proportional change on mass was larger than with the T-seals. One explanation to this is that the O-rings are made from a different material compound.

5.2 Absorption test in elevated temperature

The seals used on the elevated temperature absorption test are the same that were used on the test performed in room temperature.

Table 6 Weighting results of 4h immersion test at elevated temperature.

4h Elevated temperature	Mass before [g]	Mass after [g]	Change of mass percent
Manufacturer 1 rod seal			
Teflon rings	5,0008	5,001	0,00399936
seal	6,001	5,9478	-0,886518914
Manufacturer 2 rod seal			
Teflon rings	5,1913	5,1902	-0,021189297
Seal	5,896	5,8641	-0,541044776
Manufacturer 1 Piston seal			
Teflon rings	0,5324	0,5326	0,03756574
Seal	1,0162	1,0153	-0,088565243
Manufacturer 2 Piston seal			
Teflon rings	0,5513	0,5516	0,054416833
seal	0,9703	0,9698	-0,051530454
Manufacturer 1			
O-ring	0,6187	0,6171	-0,258606756
manufacturer 2			
O-ring	0,5542	0,5601	1,064597618

On table 6 the weighting results of 4 hour immersion test performed at elevated temperature can be seen. The mass loss of rod seals was greater in the 4h elevated temperature test than in the 72 hour room temperature test. The mass loss of pistons seals on the other hand was smaller than in the previous test. The mass change of Teflon rings stayed insignificantly small. The largest difference between these two immersion test results was the change in the manufacturer 1 O-ring. In test 1 the mass increased but in test 2 the mass decreased. The mass of the O-ring of manufacturer 2 increased also on the test 2 such as in test 1.

5.3 Changes in seal behavior during the endurance test

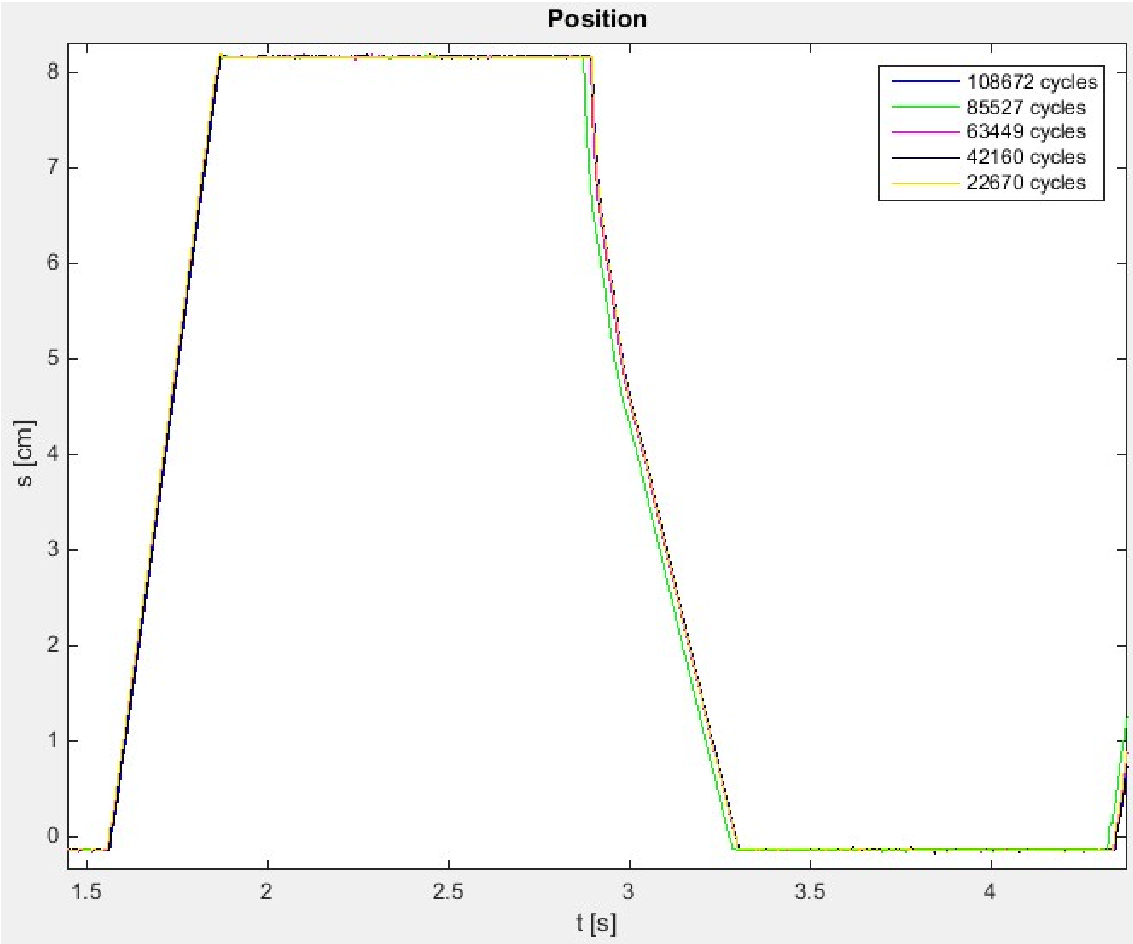
Before the endurance test the hydraulic test system was flushed with clean oil. For the endurance test the test system was filled with clean unused oil from manufacturer's containers and the oil filter was changed to a new one.

The test system was basically designed so that it did not need continuous supervision. However, the endurance test needed continuous supervision because of a malfunction which leaded the computer to send the "everything ok" signal to the logic system regardless of the state the testing system was in.

The continuous usage times for the system were between 45 minutes and 7 hours and 30 minutes. This leads to that the continuous cycle amounts between 1000 and 10000 cycles per day. This run time is due to saving the data every 1000 cycles which takes about 45 minutes to run

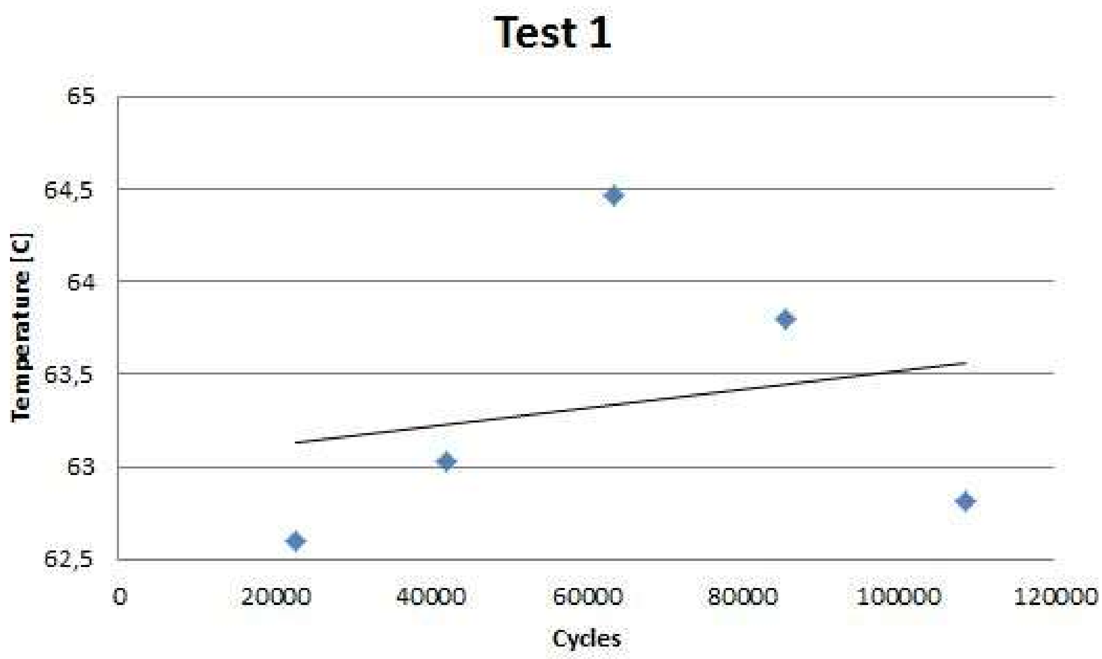
Bisztray-Balku (1995) states that qualifying seal for life with endurance test means that the test is continued until permitted friction or leakage limitation is exceeded or until minimum required service hours. The endurance test for the seals of Manufacturer 1 was stopped after 108698 cycles due to a leak. The leak was observed to be about 2-6 drops of hydraulic fluid per minute from the middle leaking hole of the cylinder resembling a 12 ml leak per hour at an average. Because the leak was from the middle hole the leaking seals were the pistons seals. This amount of leakage was more that acceptable in the actual usage system so the test was stopped at this point. Also the rod seals leakaged but the leakage was estimated to be less than 1 drop per minute.

The endurance test for the seals of Manufacturer 2 was stopped after 214907 cycles. The amount of leakage was about 0.5 drops per minute resembling a 1.5 ml leak per hour at an average. The amount of leakage did not grow after about 70000 cycles. At this point the test was stopped. The amount of cycles for the seals of manufacturer 2 was nearly two times the respective amount for Manufacturer 1. This was considered to be reliable enough and the test would not be continued to the same amount of leak than in previous test.



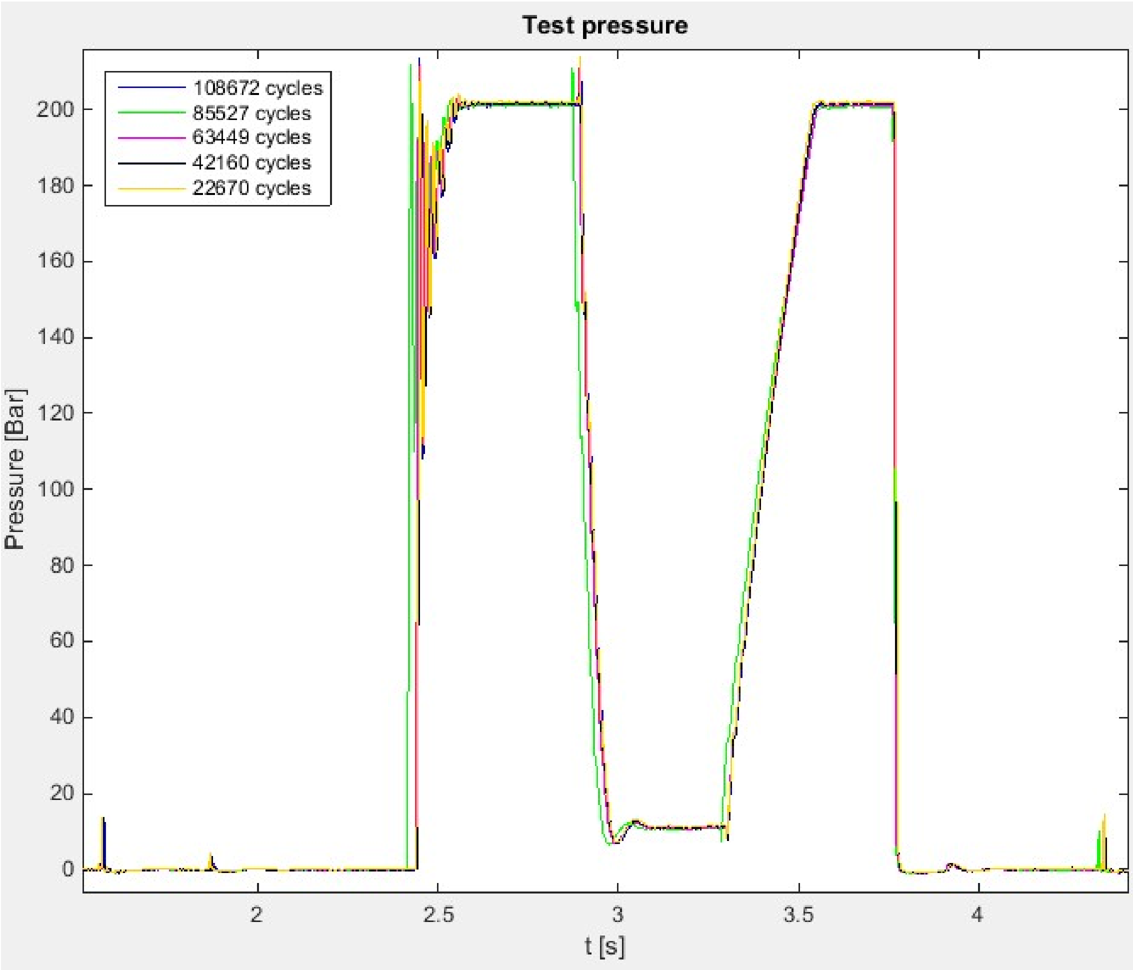
Picture 13 Cylinder position on endurance test 1.

Picture 13 shows the cylinder position in endurance test 1. There is no visible difference between the cycles. This is in accordance with the test hypothesis because if the cycle would have changed it could have effect on the test results.



Picture 14 System temperature on endurance test 1.

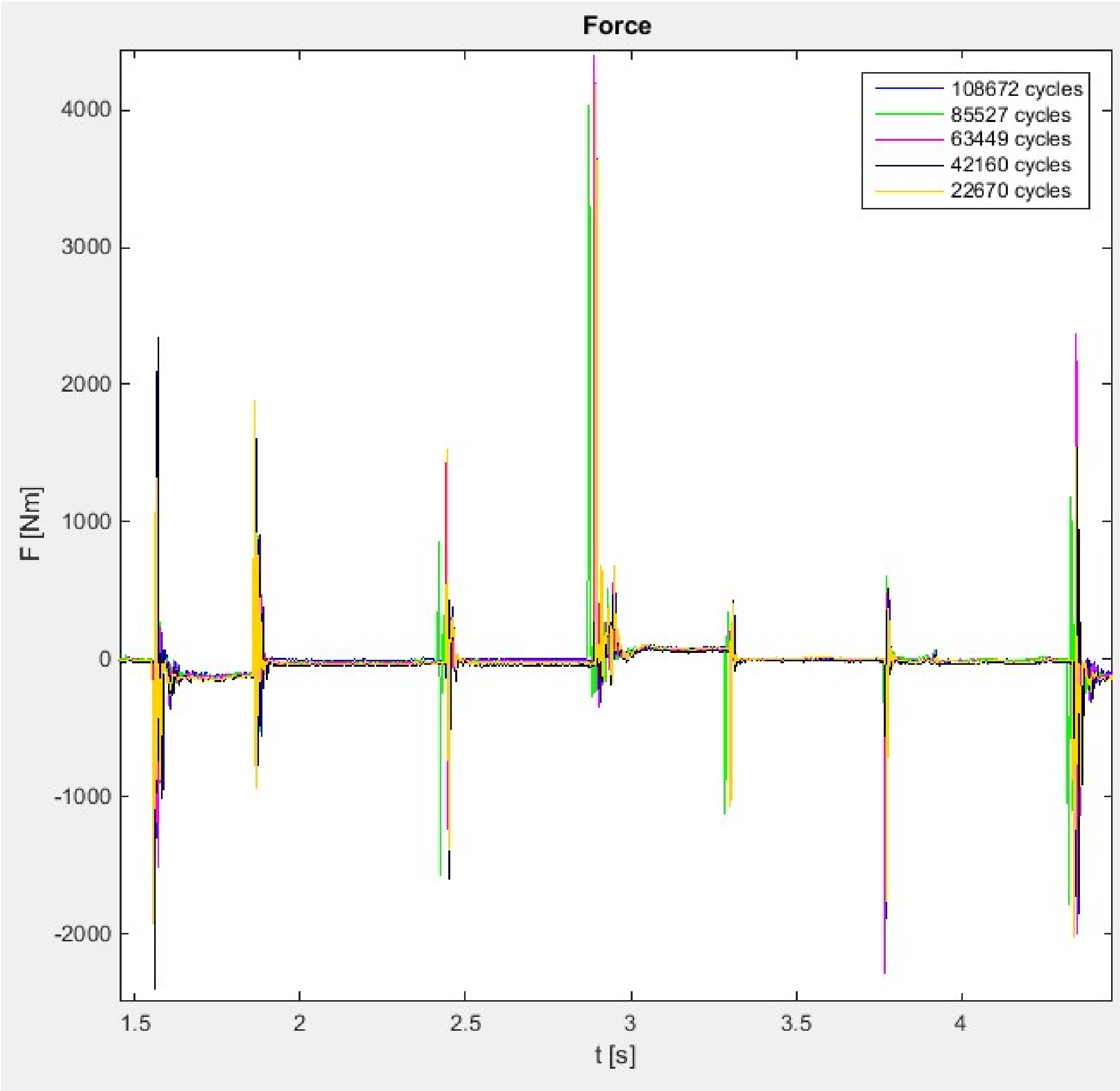
The system temperature is measured with pt100 transducer. The transducer is fixed on the hydraulic outlet of the test cylinder. The transducer measures the surface temperature which should be close to the hydraulic fluid temperature. Picture 14 shows the temperature of the system. These are maximum temperatures the test system reached after several hours of use. It would have been more optimal to have higher temperature to get closer to the real usage conditions. The hydraulic system did not have a radiator but the temperature stayed stable at the temperatures seen on picture 14. The test temperature does effect the pressure curves due to change on the viscosity of the hydraulic fluid used. The regression line on picture 14 shows that the maximum temperature increases during the test. The heating of system is due to energy loss in the system. This increasing of temperature back the observed increasing leakage.



Picture 15 Test pressure on endurance test 1.

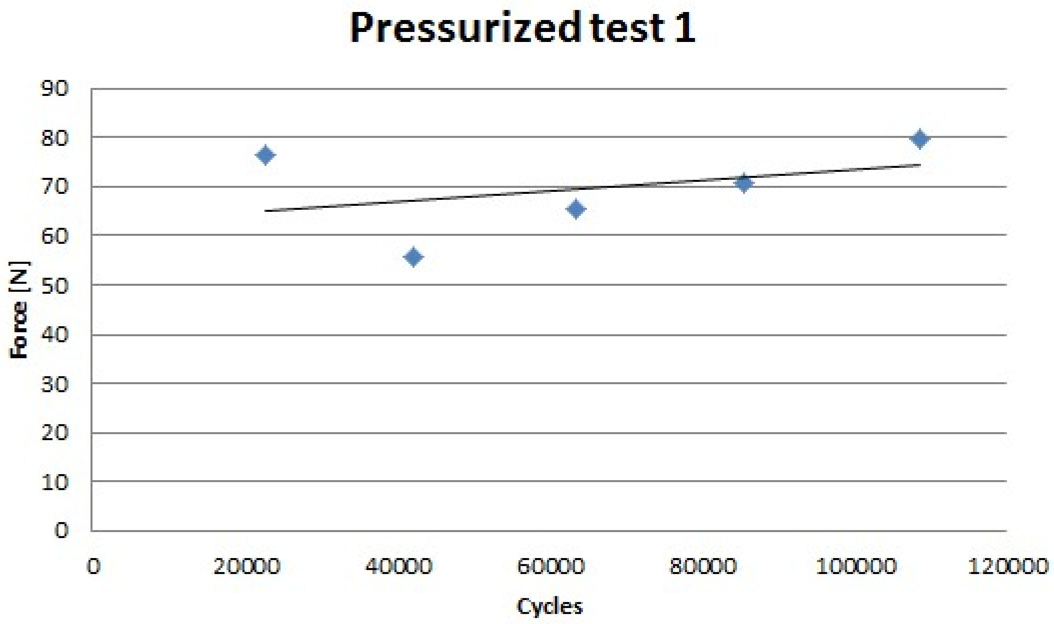
Picture 15 shows the test pressure of endurance test 1. The picture describes how the pressurizing of the test cylinder is controlled with the 2/2 valve. The cylinder is pressurized when moving to the negative direction and depressurized when moving to positive direction. When depressurized the pressure is 0 bar. When pressurized the test cylinder gets the full supply pressure between 10 and 202 bar. Largest pressure spikes are around 213 bar which is 8 bars more than on the other pressure transducer.

There is no change on supply pressure cycle during the endurance test 1 so it does not effect on test results. The constant supply pressure is 202 bar. Largest pressure spikes are 205 bar. During the cylinder movement to negative direction the pressure dropped to 10 bar. During the movement to a positive direction the pressure dropped to 58 bar. The movement cycle of the cylinder can be seen in picture 13. When changing the pressurization of the test cylinder with 2/2 valve it causes the supply pressure quickly decrease to 130 bar and raise back to 202 bar.



Picture 16 Force from endurance test 1.

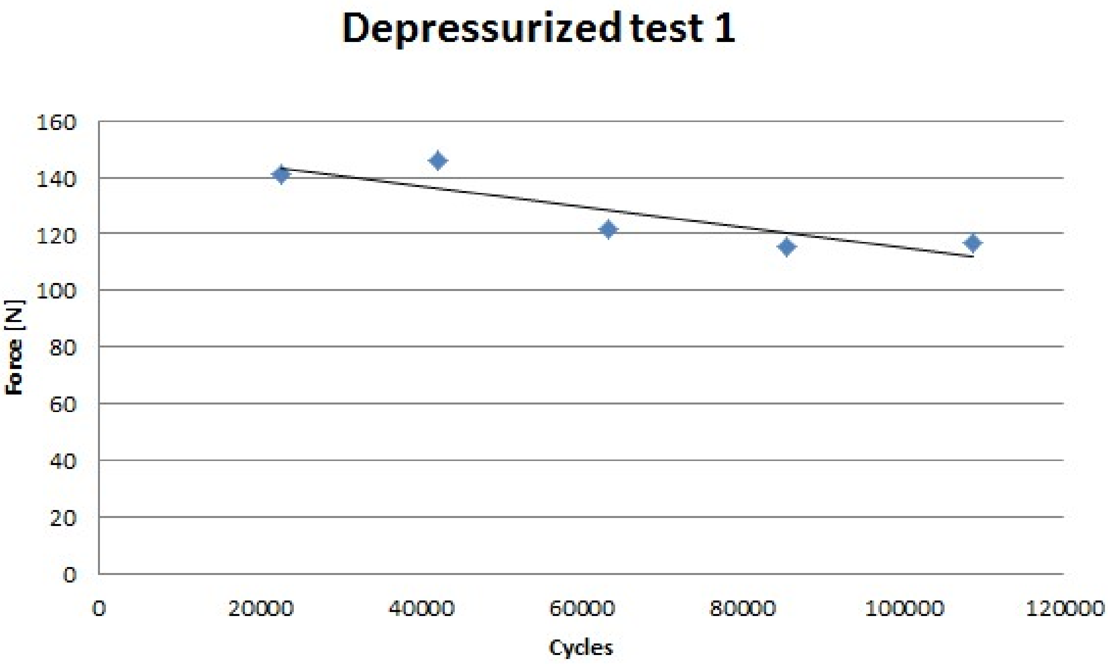
Picture 16 shows the force needed to move the test cylinder in endurance test 1. The force needed to move the test cylinder depends on the friction on the test seals. There was no visible development on the friction behavior in relation to cycle amount. The force spiked at 4500 Newtons when the cylinder started to move to the negative direction. The force spiked at -2000 Newton when the cylinder started to move to the positive direction and when 2/2 valve changed position. The constant force needed to move the test cylinder to the negative direction was 75 Newtons. The force needed to move the test cylinder to the positive direction was -135 Newtons.



Picture 17 Behavior of force at pressurized state during the endurance test 1

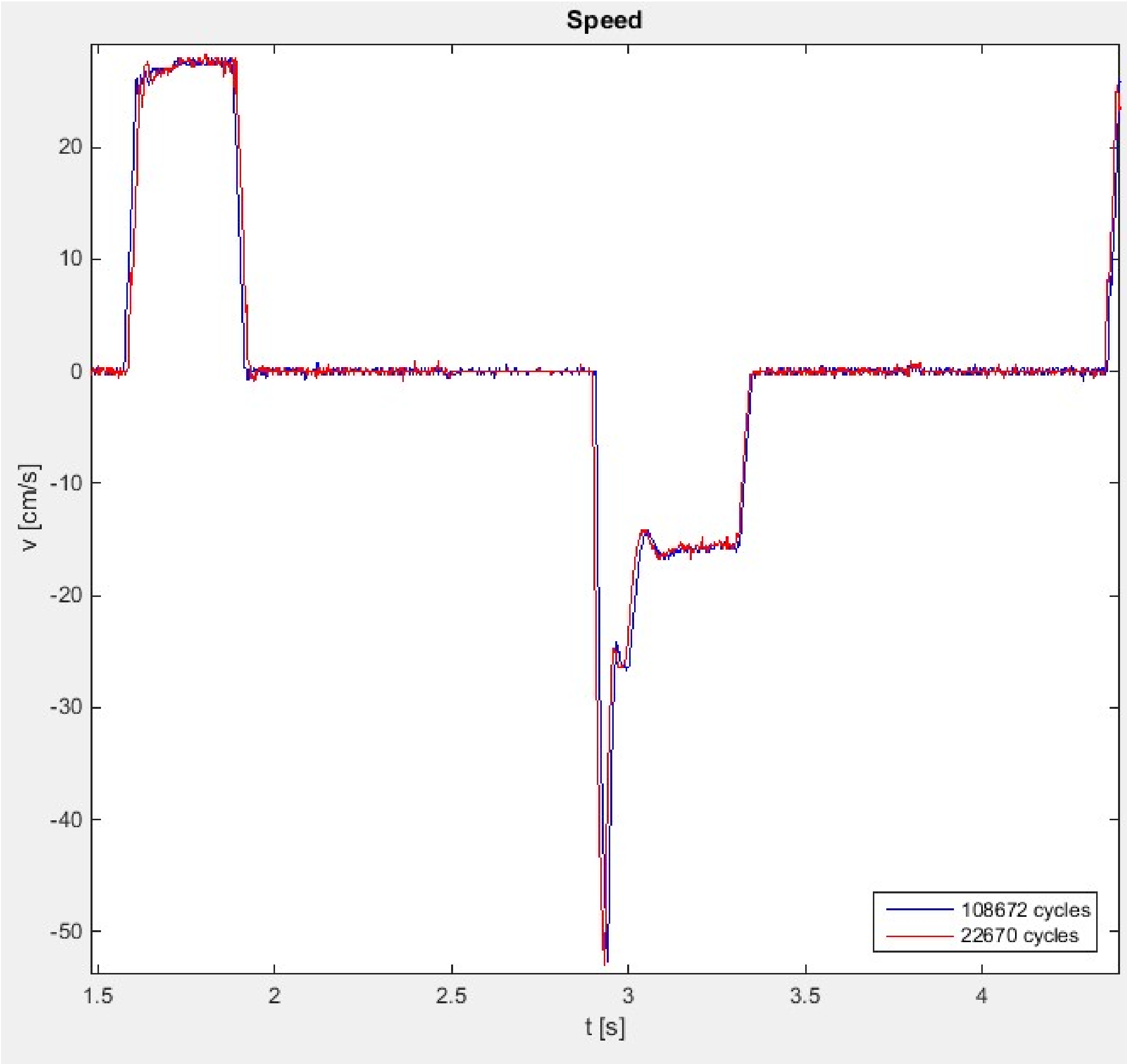
The data points are averages of 50 measurement points. Points were selected so that the movement speed and force were nearly constant for the 0.1 seconds. This 0.1 seconds is the time that 50 measurement points shows. Test cylinder was pressurized when cylinder moved to a negative direction. The position from where the movement speed and force are constant is 4.4 cm. A linear regression line was computed between the points in picture 17. The regression line shows that the amount of friction increases during the endurance test. Yet while comparing the first two data points it seems that between 20000 and 40000 cycles the friction decreases.

That may be caused because the seal has worn and that decreases the precompression of the seal. After 40000 cycles the friction starts to increase. When seal wears more it may start to affect on the hydrodynamic lubrication which results in rise of friction. Hydrodynamic lubricating film is dependent on the seal design, seal surface and the counter surface. (Koskinen & Aaltonen 2013, p. 1713)



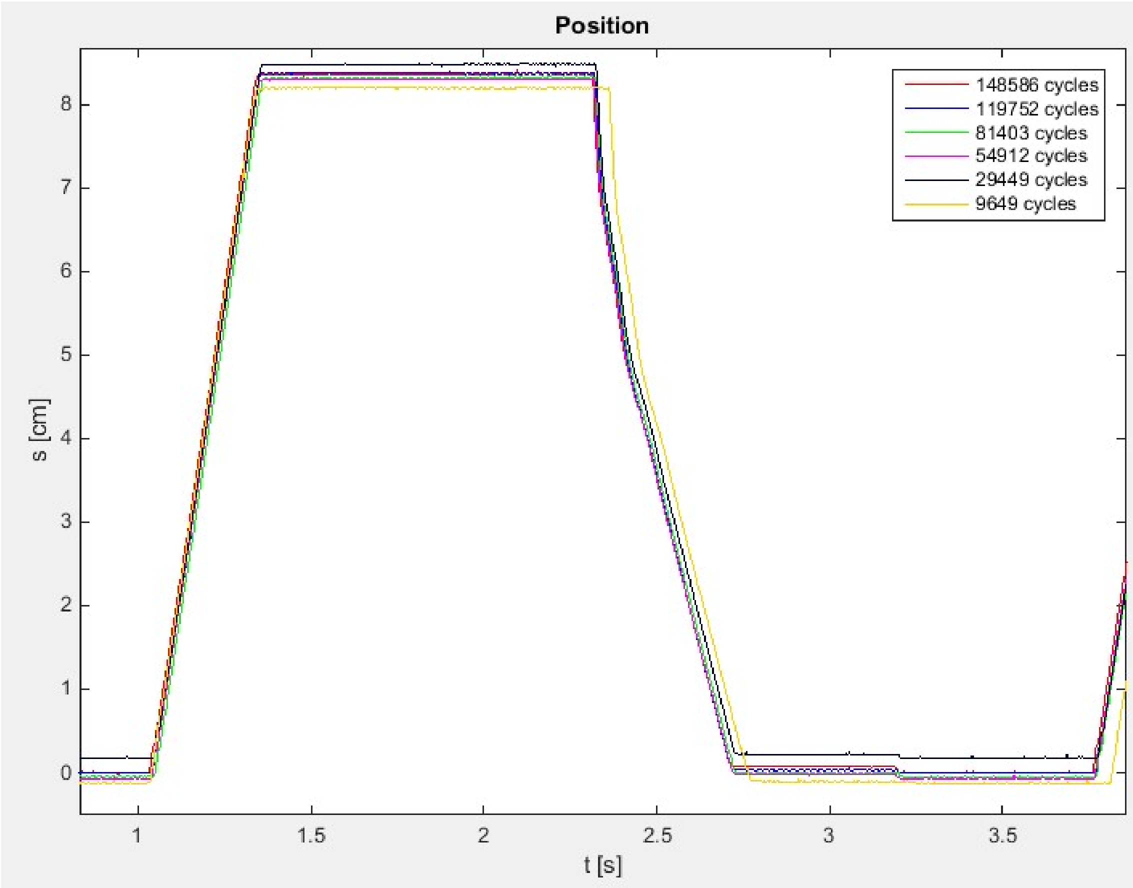
Picture 18 Behavior of force at depressurized state during the endurance test 2

The data points are averages of 50 measurement points. Points are selected so that the movement speed and force were nearly constant for the 0.1 second time that 50 measurement points show. Test cylinder is depressurized when cylinder moves to a positive direction. The position from where the movement speed and force are constant is 2.2 cm. The regression line in picture 18 shows that the amount of friction decreases during the endurance test while the system is depressurized. Wearing of the seal decreases the precompression and that may be the explanation to the decrease of the friction



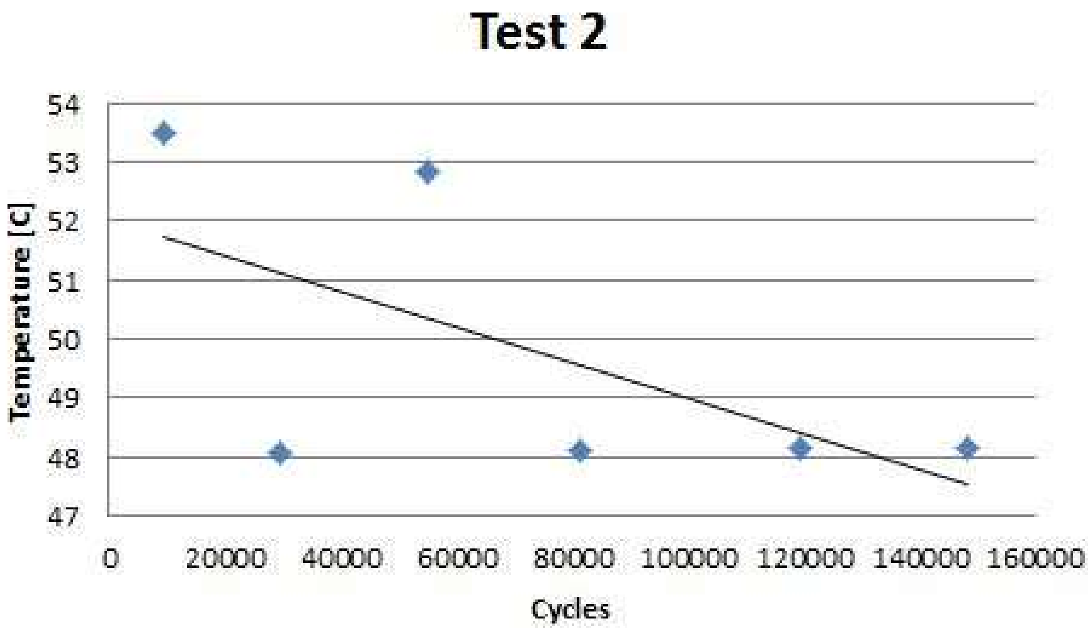
Picture 19 Speed in endurance test 1.

Picture 19 shows the speed of test cylinder in endurance test 1. The movement speed to negative direction spikes at 52 cm/s and then stays constant at 16 cm/s. The movement speed to positive direction is 27 cm/s. The movement speed cycle did not change during the endurance test 1 as is shown in the picture 19. This means that the system works as planned and is not broken.



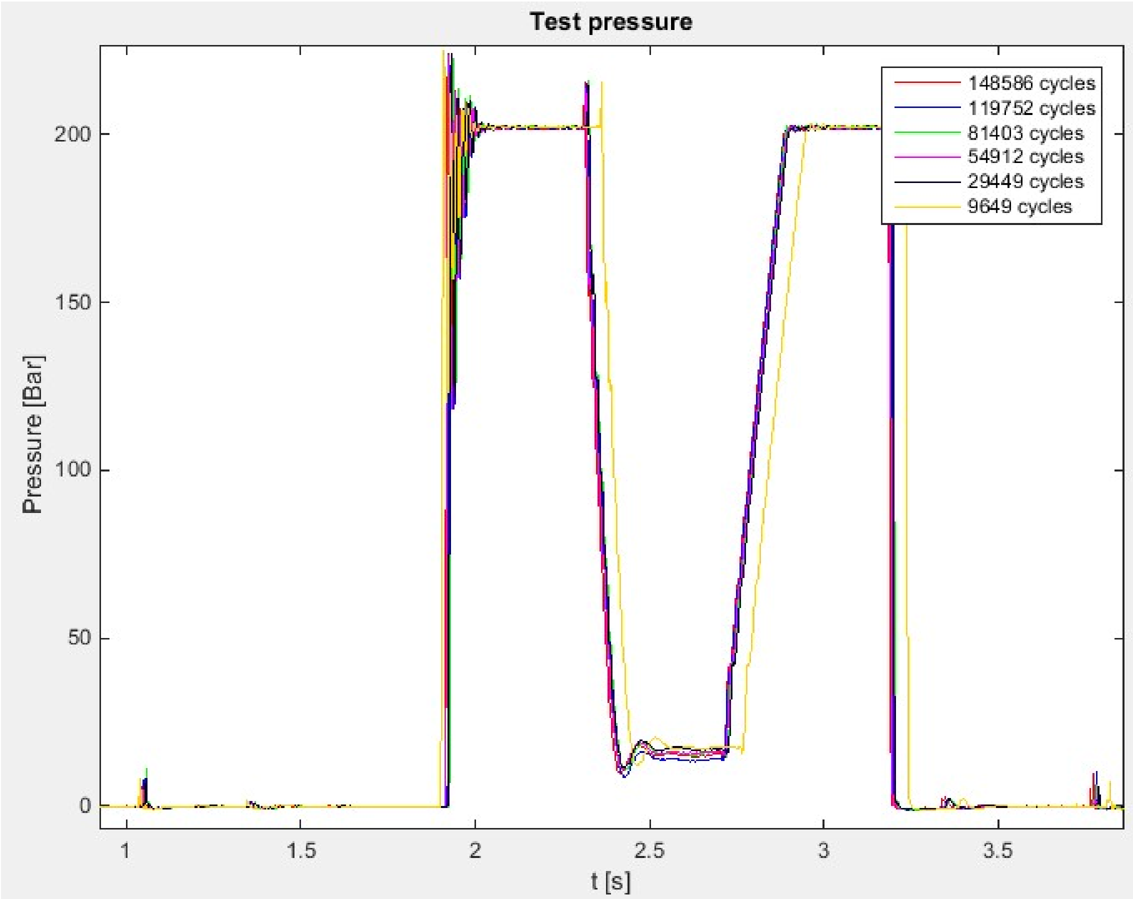
Picture 20 Cylinder position on endurance test 2.

Picture 20 shows the cylinder position in endurance test 2. The minimum and maximum position values differentiate between different cycles. This happens because the cylinder started slightly to turn around its axis affecting to the distance the position transducer moves. At 0 seconds all the curves are on each other. At the end of the graph it is visible that the yellow and cyan colored lines differ from the other lines. This is possibly because the induction transducers were detached from the system. When the transducers were installed back it was possible that the position was not precisely the same as before. That was possible caused by the design of the rig for the transducers that was taken from another test setup.



Picture 21 Temperature on endurance test 2.

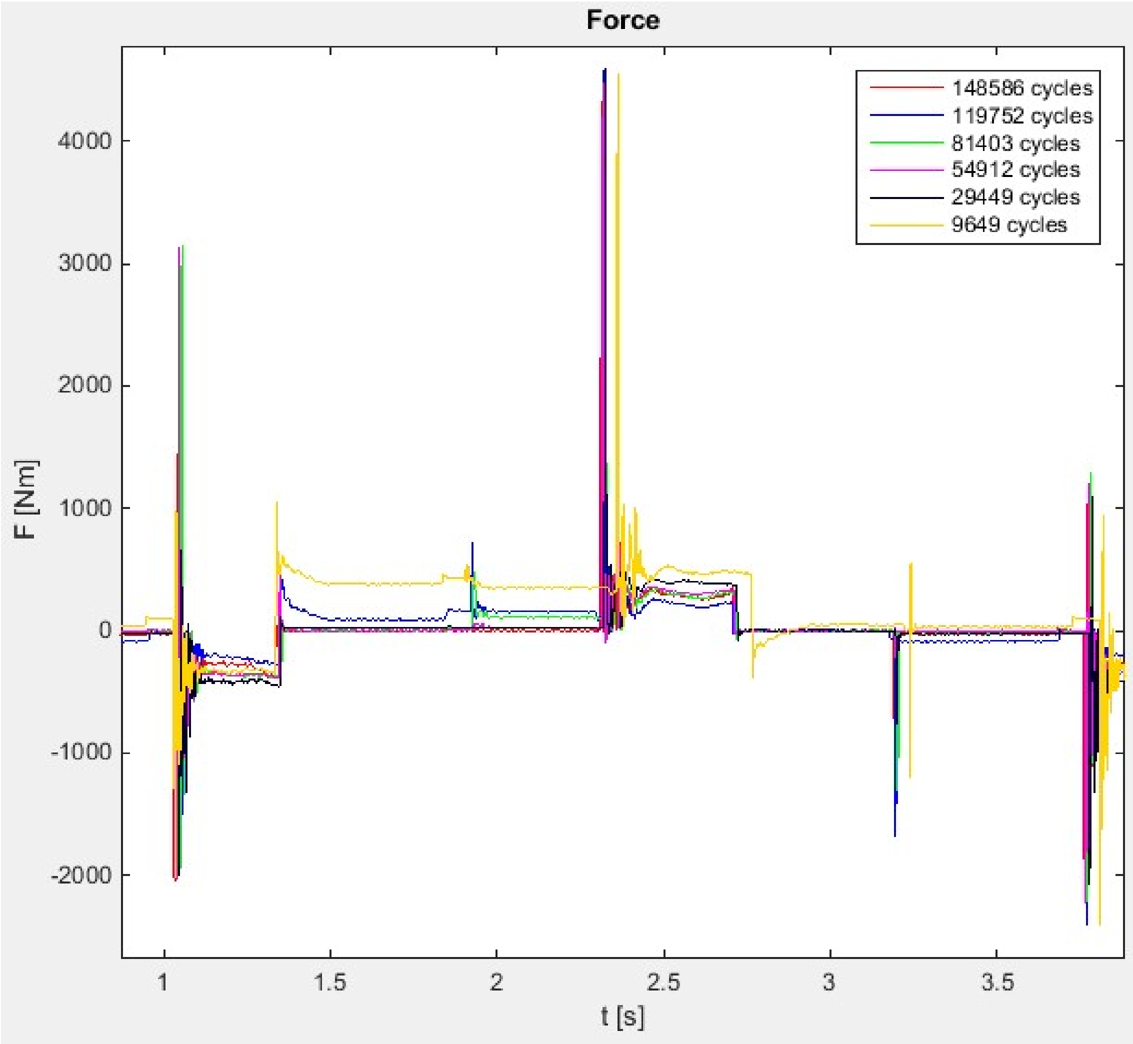
Picture 21 shows the temperature of the system during the other plots of endurance test 2. There is a clear difference between the later cycles of the temperatures between the endurance tests 2 and 1. During the endurance test 2 the system did not warm more than is shown in picture 21. The difference was approximately 16 degrees Celsius. This difference may be caused by the difference in cycle time that was caused by the change of the induction transducers placement. The regression line in picture 21 shows that the maximum temperature decreases during the test. The heating of system is due to energy loss in the system. This decreasing of temperature may back the observation that the leakage does not increase and the friction decreases. That does mean that the energy loss in the system decreases.



Picture 22 Test pressure on test 2.

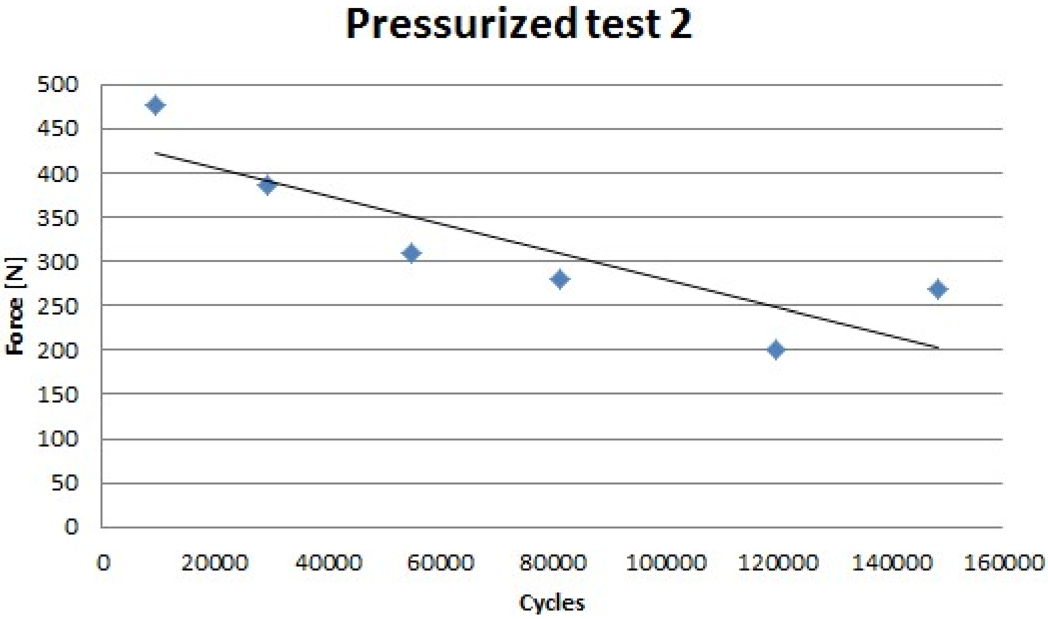
Picture 22 shows the test pressure in endurance test 2. The difference to test 1 was that the minimum pressure on negative direction was 7 bar higher. Also pressure spikes on test 2 were higher spiking at 225 bar.

There is no change on supply pressure cycle during the endurance test 1 so it does not effect on test results. The minimum pressure while the cylinder moved to negative direction was 15 bar. When the cylinder moved to positive direction the minimum pressure was 58 bar. The maximum pressure was 202 bar. Pressure spikes were 208 bar. The only difference between the supply pressures in tests 1 and 2 was 7 bar on the minimum pressure on negative direction and a 3 bar difference on pressure spikes.



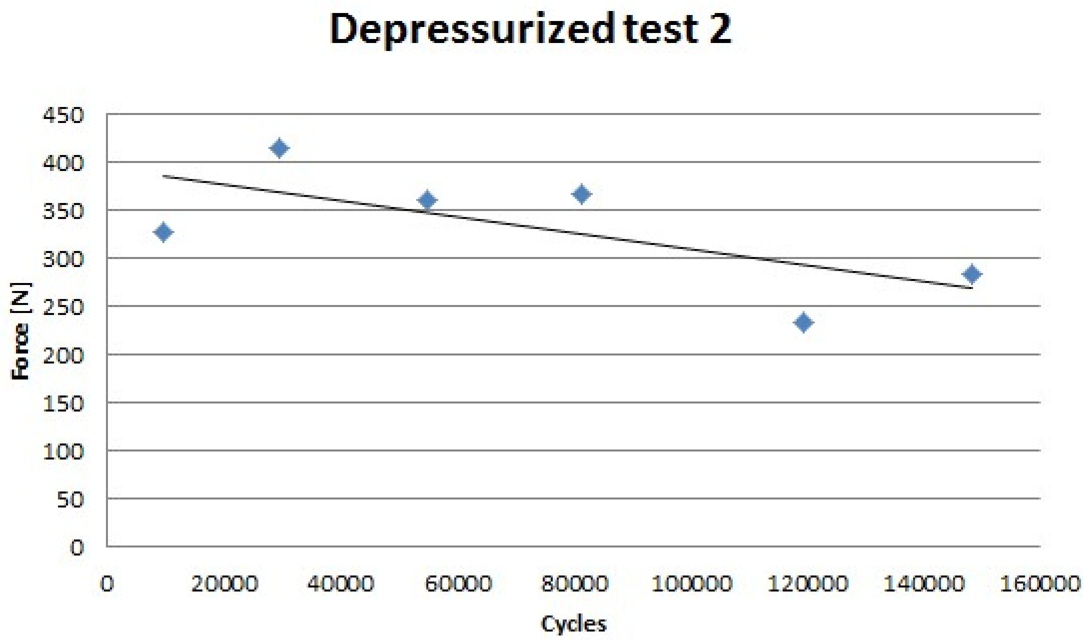
Picture 23 Force in test 2.

Picture 23 shows the force needed to move the test cylinder in endurance test 2. It is visible that the two earlies cycles had most friction but the friction behavior is still not clear. The seals tested in the endurance test 2 also seemed to have more friction than the seals used on test 1. The constant force needed to move the test cylinder to negative direction was between 200 and 470 Newton and spikes at 4600 Newton. The constant force needed to move the test cylinder to positive direction was between -200 and -415 Newton. The force spiked at -2000 Newton when cylinder started to move to a positive direction and when 2/2 valve changed position as it did also in test 1.



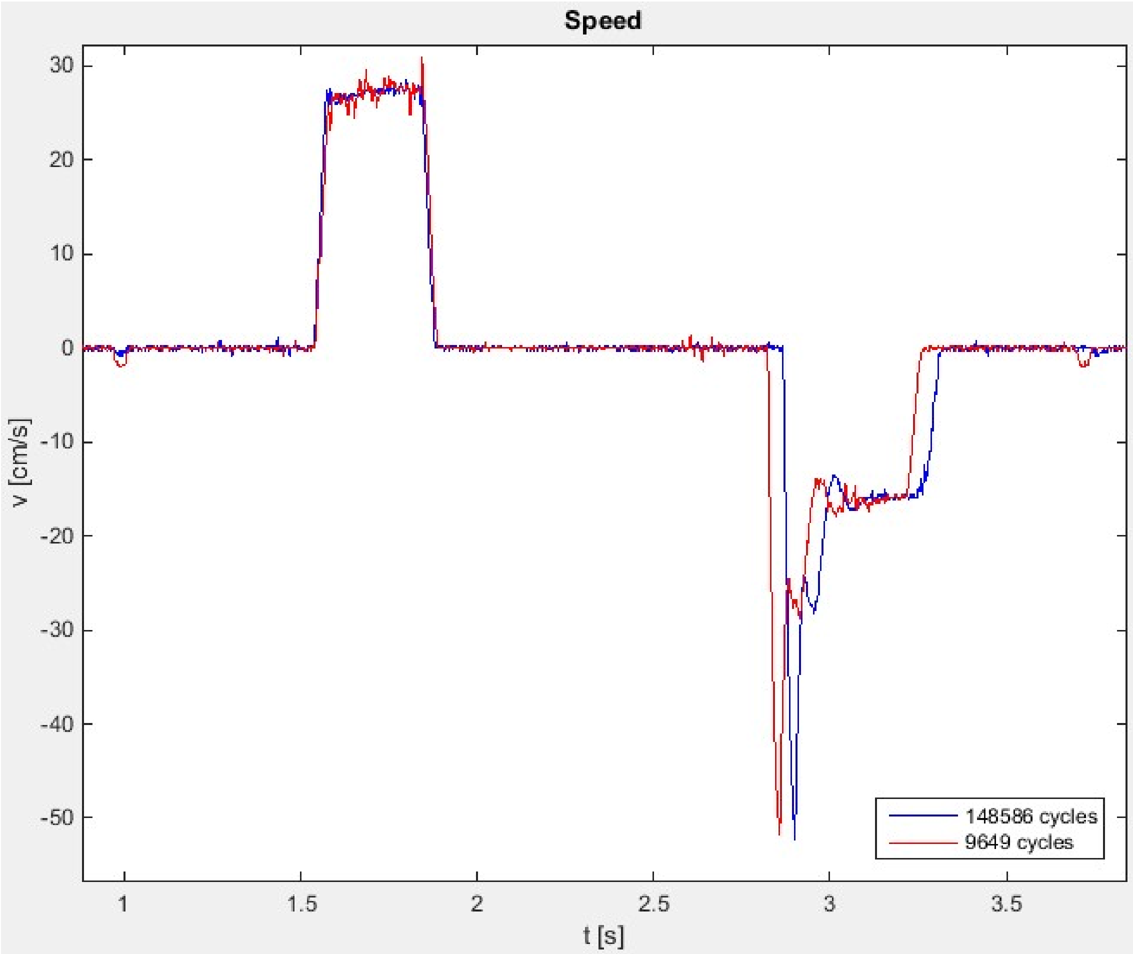
Picture 24 Behavior of force at pressurized state during the endurance test 2

The position from where the movement speed and force are constant is 4.4 cm as in the endurance test 1. The regression line in picture 24 shows that the amount of friction decreases during the endurance test. That may be caused because the seals had worn and that decreased the precompression of the seal as in the test 1. When single data points are looked into the friction increased between 120000 and 148000 cycles. It seems that the same wearing point which occurred in test 1 around 40000 cycles happens on test 2 around 120000 cycles.



Picture 25 Behavior of force at depressurized state during the endurance test 2

The position from where the movement speed and force are constant is 2.2 cm as in the test 1. The black color line in picture 25 is a linear regression line. The regression line shows that the amount of friction decreases during the endurance test but not as quickly as in the pressurized state. Wearing of the seal decreased the precompression and that may be the explanation to the decrease of the friction.



Picture 26 Speed on endurance test 2.

Picture 26 shows the speed of the test cylinder in endurance test 2. The speeds were the same than in endurance test 1. This graph shows that the difference between the lines were due to different time the cylinder stays on end position and not because of different speed the cylinder moves on. The previously made assumption of the changed position of induction transducers is backed by this fact that movement speeds do match. This does explain the minor change in the cycle time.

5.4 Material loss during the endurance test

After stopping the endurance test the test cylinder was opened and seals were detached from the test components. The seals were left to dry on a nearly lint free paper towels for an hour like in the absorptions test. After this the seals were dried carefully with the paper towels before weighting. The seals were weighted with the same analytical balance scale that was used in the absorption test.

Table 7 Weighting results before the endurance test.

Before endurance test	1	2	Mass different between the seals %
Manufacturer 2 rod seal			
Teflon rings	5,004		-100
Seal	6,037		-100
Manufacturer 1 rod seal			
Teflon rings	5,189		-100
Seal	5,915		-100
Manufacturer 2 piston seal			
Teflon rings	0,5314	0,5304	-0,18818216
Seal	1,0258	1,0206	-0,506921427
Manufacturer 1 piston seal			
Teflon rings	0,551		-100
Seal	0,981		-100
Manufacturer 2			
O-ring	0,609		-100
Manufacturer 1			
O-ring	0,541		-100

The only seals that were weighted before the endurance test were the piston seals of Manufacturer 2. For other seals the mass of the seals from the weighting that was done before the absorption test was used. These masses from the previous weighting can be assumed to be sufficiently close to the mass of the actual seals used. In such a small and precise parts as the seals used here it can be assumed that the mass differentiation of seals is really small. This assumption is further emphasized by the weighting result of the manufacturer two's pistons seals before the endurance test. As shown in the table 7 the mass difference between two Teflon rings was under 0.2 % and the mass of the third seal weighted in the absorption test was between the two weighted here. The mass difference between the three pistons seals of Manufacturer 2 is 0.62 %

Table 8 Weighting results after the endurance test.

1	2	Mass difference between the seals %	Mass change % 1	Mass change % 2
5,0013	4,7244	-5,536560494	-0,053956835	-5,587529976
5,9936	6,0282	0,577282435	-0,718900116	-0,145767765
5,1578	5,1553	-0,048470278	-0,601271921	-0,649450761
5,802	5,811	0,155118925	-1,910397295	-1,758241758
0,5226	0,5197	-0,554917719	-1,656003011	-2,0173454
1,0106	0,9911	-1,929546804	-1,481770326	-2,890456594
0,5222	0,5355	2,54691689	-5,226860254	-2,813067151
0,9663	0,9191	-4,884611404	-1,498470948	-6,30988787
0,6035	0,6057	0,364540182	-0,903119869	-0,541871921
0,5211	0,509	-2,32201113	-3,678373383	-5,914972274

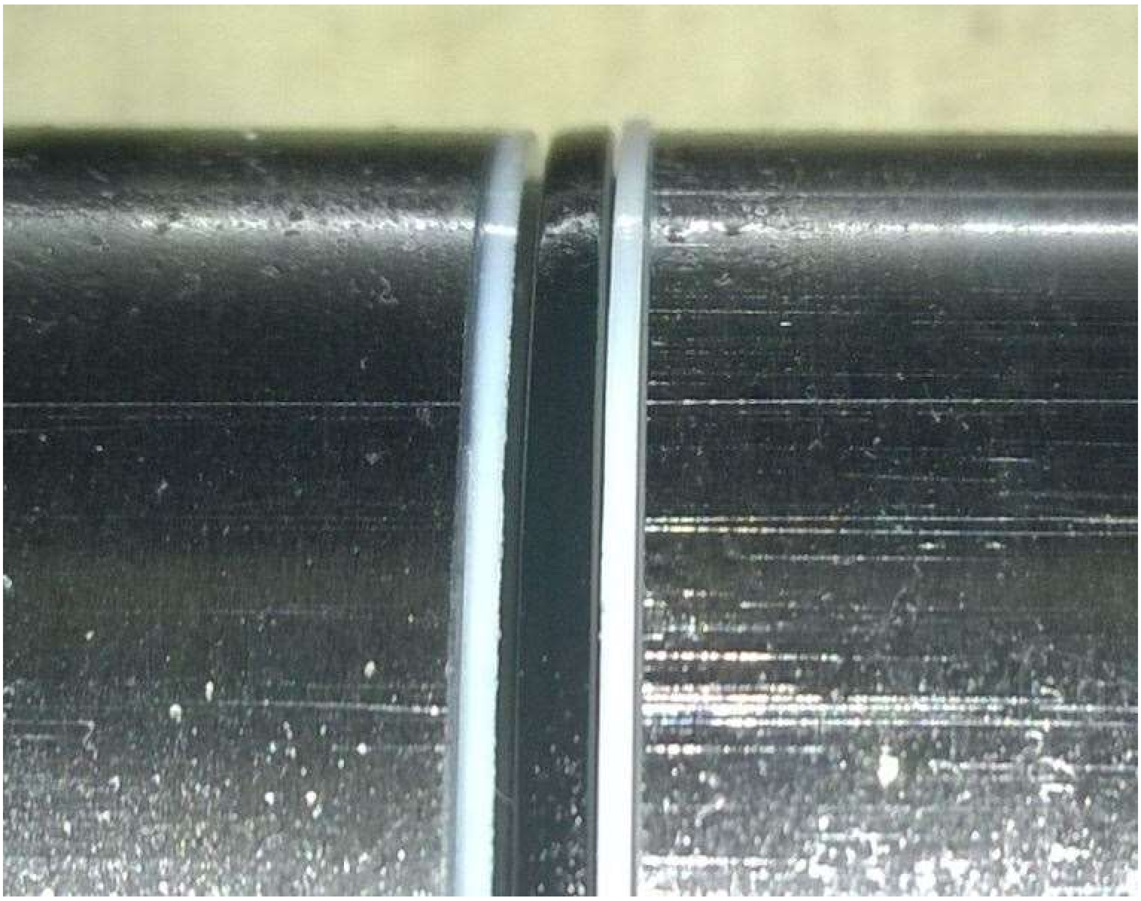
The rows in table 8 are the same as in table 7. The 5.58 % change in mass of the Manufacturer 2's second rod seals Teflon rings can be regarded as substantial. Although this was not visible while looking at the Teflon rings. Other Teflon rings from both manufacturers' rod seals lost under 0.65 % of their mass which is not significant when taking into account the 0.62% differential on the piston seals before the test. Interesting is that the rod sealing from the four tested seals that lost the least mass from elastomer is the one which lost the most mass from the Teflon rings. Table 8 shows that this seal lost only 0.15% of its mass which is an insignificant amount. The other seal lost 0.72% which is still not much and can be explained by the difference between seals. The seals of Manufacturer 1 lost 1.76 - 1.91 % of their mass which is a significant amount. This can explain the observed leakage from the rod seals. The difference of the mass changes between the rod seals of Manufacturers 1 and 2 are clear. When the amount of cycles run is taken into account the difference gets even clearer because the seals of Manufacturer 2 run nearly double amount of cycles compared to the respective amount of Manufacturer 1.

The piston seals of manufacturer 1 did lose a 5.23% and 2.81% of their mass from the support rings. The mass loss is significant. The piston seals of manufacturer 2 lost 1.66% and 2.02% mass from support rings. The support rings of piston seals lost much more mass than the support rings of rod seals. The difference between the mass loss from support rings of manufacturer 1 and 2 is clear. Piston seals of manufacturer 1 lost 1.50% and 6.31% of mass from the elastomer. The 6.31% mass loss is very significant and can clearly explain the leakage. The manufacturer 2's piston seals lost 1.48% and 2.89% mass from the elastomer. The mass loss is significant but the difference to the test 1 is still clear. The differences of mass loss between the tests support the observed difference in leakage.

The mass loss of manufacturer 1's O-rings are 3.68% and 5.91%. The mass losses are very significant. The O-rings of manufacture 2 lost 0.90% and 0.54% of their mass. The difference of mass loss between the manufacturers is clear.

5.5 Visual inspection of material loss

Abrasion is the process of material mechanical wearing down or when material rubs away.



Picture 27 Piston seal 2 from endurance test 1.

Picture 27 shows manufacturer 1's piston seal 2 which lost 6.3 % mass from the seal and 2.8 % mass from the Teflon rings. Evident abrasion can be seen on the seal and the top of the seal has worn flat.

Picture 28 Piston seal 1 from endurance test 1.

Picture 28 shows the piston seal 1 from manufacturer 1. In This seal shows no visible abrasion as on the seal 2 and the seal has kept its round form. Seal one lost only 1.5 % of its mass and thus there was a clear difference when compared to the seal 2. Seal 1 lost a 5.2 % mass from the Teflon rings. The mass loss is nearly double compared to the loss on seal 1. It is not clear how the Teflon rings have lost so much mass.

Picture 29 O-ring 2 from endurance test 1.

Picture 29 shows the O-ring 2 of manufacturer 1 after the endurance test. There is a considerable amount of wearing visible on the seal. This type of wearing is not normal for a static seal. The wearing may be caused by the seal pushing into the clearance between the cylinder pipe and cylinder end. The seal lost 5.9 % of its mass and it can clearly be seen that it is due to abrasion. Next to the seal is a lot of rubber crumb which is shown in picture 29.

Picture 30 O-ring 1 from endurance test 1.

Picture 30 shows the O-ring 1 of manufacturer 1 after the endurance test. Rubber flakes have ripped from the seal. This demonstrates how the seal lost nearly 3.7 % of its mass. This wearing behavior is not normal for seals as in the O-ring 2. The same explanations for the abnormal wearing apply for seal 1 as the seal 2. The abrasion of seal 1 and seal 2 differ from each other. The way the rubber has worn from the seal is different. The O-ring 1 has lost less mass compared to O-ring 2.

Picture 31 Piston seal 1 from endurance test 2.

Picture 31 shows the piston seal 1 of manufacturer 2 after the endurance test. On the seal is visible abrasion because the seal has worn nearly flat. The seal has lost 2.9 % of its mass which is less than half of what the manufacturer 1's seal lost. Teflon rings have lost 2 % of their mass. The mass loss is smaller than on the endurance test 1. Picture 31 shows that the left Teflon ring has been elongated. The wavelike form of the Teflon ring shows the elongation.

Picture 32 Piston seal 2 from manufacturer 2.

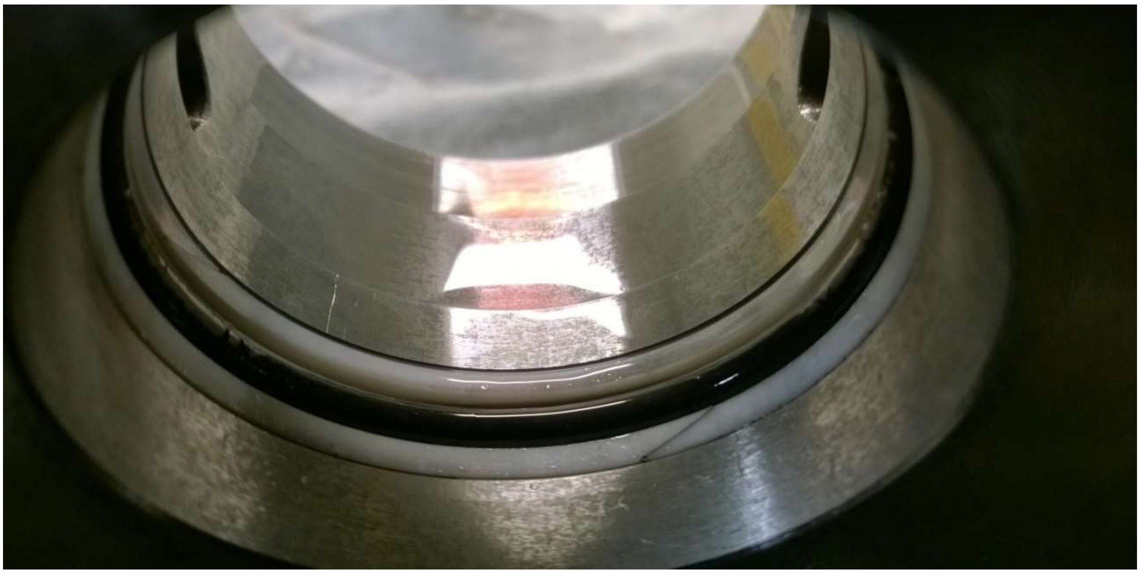
Picture 32 shows the piston seal 2 from manufacturer 2 after the endurance test. Seal 2 has clearly more roundness left than seal 1. Weighting results show that this seal lost 1.5 % of its mass which is the same amount as the seal of manufacturer 1. Teflon rings lost 1.7 % of their mass. This is less than the seal of manufacturer 1 lost. There is also a visible deformation on the Teflon ring. The ring has turned 90 degrees around its axis on the part where is does lay against the cylinder pipe.

Picture 33 O-ring 2 from endurance test 2.

Picture 33 shows the O-ring 1 of manufacturer 2 after endurance test. There is very little visible abrasion on O-ring 1. The O-ring has kept its original round form. This observation is confirmed by the weighting result which shows a 0.54 % mass loss. There is a clear difference when compared to the 5.9% mass loss of the manufacturer 1 O-rings.

Picture 34 O-ring 1 from endurance test 2.

Picture 34 shows the O-ring 2 of manufacturer 2 after endurance test. Some rubber crumbs are visible due to abrasion. The mass loss of the O-ring is 0.9 % which is much less than that of the O-rings of manufacturer 1.

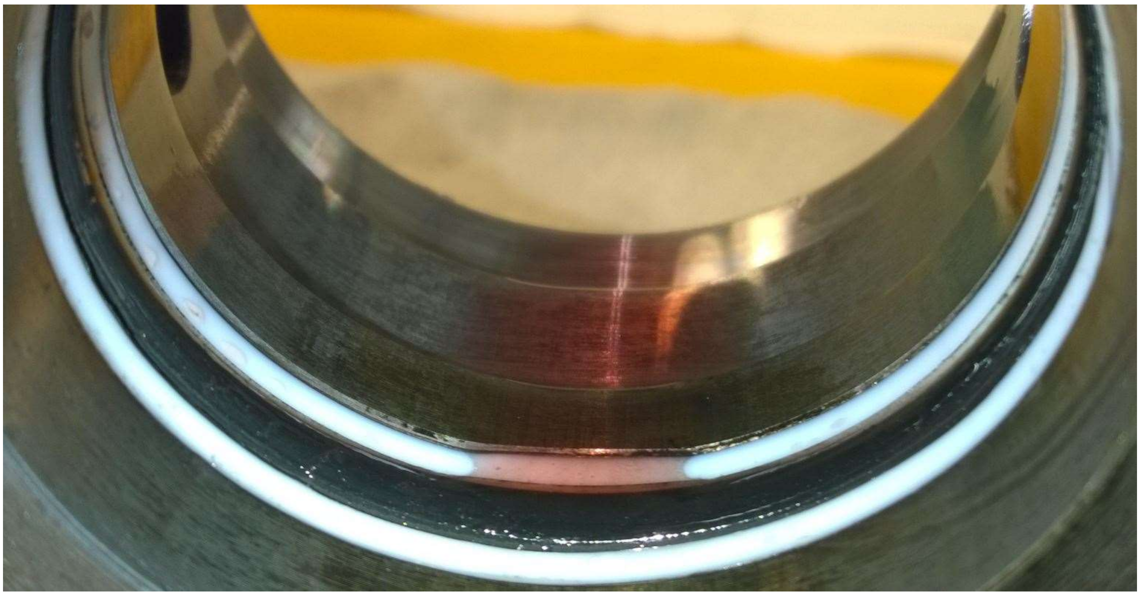


Picture 35 Rod seal 1 from endurance test 1.

Picture 35 shows rod seal 1 from endurance test 1. The seal has lost 0.6% mass from support rings and 1.91% mass from elastomer. There is a little visible abrasion on the elastomer part because it is not as round as a new one. The mass loss from the elastomer is quite significant.

Picture 36 Rod seal 2 from endurance test 1.

Picture 36 shows the rod seal 2 from endurance test 1. The seal has lost 0.64% mass from support rings and 1.76% mass from the elastomer. There is a little visible abrasion on the elastomer part because it is not as round as a new one. The loss of mass from the elastomer is quite significant.



Picture 37 Rod seal 1 from endurance test 2.

Picture 37 shows the rod seal 1 from endurance test 2. There is no visible abrasion on the seal. The seal has lost 0.054% mass from the support ring and 0.72% mass from the elastomer. The loss of mass is insignificant and the visual observation supports the measurements findings.

Picture 38 Rod seal 2 from endurance test 2.

Picture 38 shows the rod seal 2 from endurance test 2. The seal has lost 5.6% mass from support rings and 0.15% mass from the elastomer. There is no visible abrasion on the elastomer part. The mass loss from the support rings is very significant. Visual inspection shows that the surfaces of the support rings are smooth so the wearing seems normal.

5.6 Fluid leakage

The fluid leakage was observed during the endurance test. Test temperature decreases during the endurance test 2 by 5 degrees Celsius. The decrease of temperature is shown in picture 21. The effect of temperature change to the viscosity of the hydraulic fluid needs to be found out. The change of viscosity may effect on the leakage flow.

Dynamic viscosity can be calculated when the kinematic viscosity and the density are known (Kauranne et al. 2008).

$$\nu = \frac{\eta}{\rho} \qquad \text{(Equation 1.)}$$

Where:

η Dynamic viscosity (Pas)

ρ Density (850 kg/m^3)

ν Kinematic viscosity (m^2/s)

The kinematic viscosity of the hydraulic fluid was given at 40 and 100 Celsius degrees.

ν_{40} 14 m^2/s

ν_{100} 3.45 m^2/s

The change of the kinematic viscosity can be assumed to be linear between 40 and 100 Celsius degrees. The kinematic viscosities at 48, 53 and 63 degrees are shown below.

ν_{48} 12.59 m^2/s

ν_{53} 11.71 m^2/s

ν_{63} 9.95 m^2/s

The dynamic viscosities can be calculated when the kinematic viscosities are known.

$$\eta_{48} = \nu_{48}\ \rho \qquad \text{(Equation 2.)}$$

Which gives dynamic viscosity of 0.011 Pas.

$$\eta_{53} = \nu_{53}\ \rho \qquad \text{(Equation 3.)}$$

Which gives dynamic viscosity of 0.009955 Pas.

The flow past the seal can be calculated with the equation of flow in a circular gap.

$$q_v = \frac{\pi d h^3}{12\eta l}(p_{1-}\, p_2) \qquad \text{(Equation 4.)}$$

Where:

q_v Flow in the gap (m^3/s)

d Outer diameter of the flow channel (m)

h Height of the gap (m)

η Dynamic viscosity (Pas)

l Length of the gap (m)

p_1 Pressure in starting end of the gap (Bar)

p_2 Pressure in end of the gap (Bar)

The actual height of the gap is not known but it can be solved from the equation 4. The observed leakage at the end of the endurance test 1 is used as the flow.

$$h = \sqrt[3]{\frac{q_{vo} 12 \eta_{63} l}{\pi d (p_1 - p_2)}} \qquad \text{(Equation 5.)}$$

Where:

q_v Flow in the gap (3.333*10-9 m^3/s)

d Outer diameter of the flow channel (0.0563 m)

h Height of the gap (m)

η Dynamic viscosity (Pas)

l Length of the gap (0.0436 m)

p_1 Pressure in starting end of the gap (210 Bar)

p_2 Pressure in end of the gap (0 Bar)

$$\eta_{63} = \nu_{63}\ \rho \qquad \text{(Equation 6.)}$$

Which gives dynamic viscosity of 0.008459 Pas.

When using the result from equation 6 the equation 5 gives the height of gap $1.917*10^{-6}$ m.

The result from equation 5 is used to solve the equation 4 at temperatures of 48 and 53 Celsius degrees.

The equation 4 gives a flow of $4.672*10^{-9}$ m^3/s at temperature of 48 Celsius degrees.

At 53 Celsius degrees the equation gives a flow of $5.023*10^{-9}$ m^3/s. The change of flow due to temperature change can be calculated.

$$\frac{q_{v53}-q_{v4}}{q_{v53}} * 100\% \qquad \text{(Equation 7.)}$$

Which gives a flow change of 6.989 %

The cooling of 5 degrees does explain a roughly 7 % change on the leakage flow. These calculations show that the cooling does not explain why the leakage does not increase to the same level than on the endurance test 1. The leakage on endurance test 1 was 12 ml per hour and on endurance test 2 1.5 ml per hour. The difference on leakage between the endurance tests is most likely due to differences on wearing of the seals.

5.7 Discussion

When comparing the friction behavior from pictures 17 and 18 to picture 24 and 25 it can be assumed that the seals behaved in a similar way on the endurance tests but the seals of test 1 wore out much faster. From the figures of pressured state it can be assumed that the seals of manufacturer 2 reached the same amount of wearing at 120000 cycles as the seals of manufacturer 1 reached already at 40000 cycles thus suggesting a difference in material quality. From the figures it can be seen that until these cycle amounts the amount of friction keeps decreasing and after that the friction starts to increase. The increase of friction can be assumed to be caused by the changing of hydrodynamic lubrication because of the wearing of the seals. The frictions behavior of depressurized state show decreasing friction on seals of the both manufacturers. In depressured state the hydrodynamic friction is not present because it depends on the pressure. The decreasing of friction can be explained by the decreasing precompression when the seal wears. Similar decreasing of friction during seal endurance test is observed by Bisztray-Balku (1999).

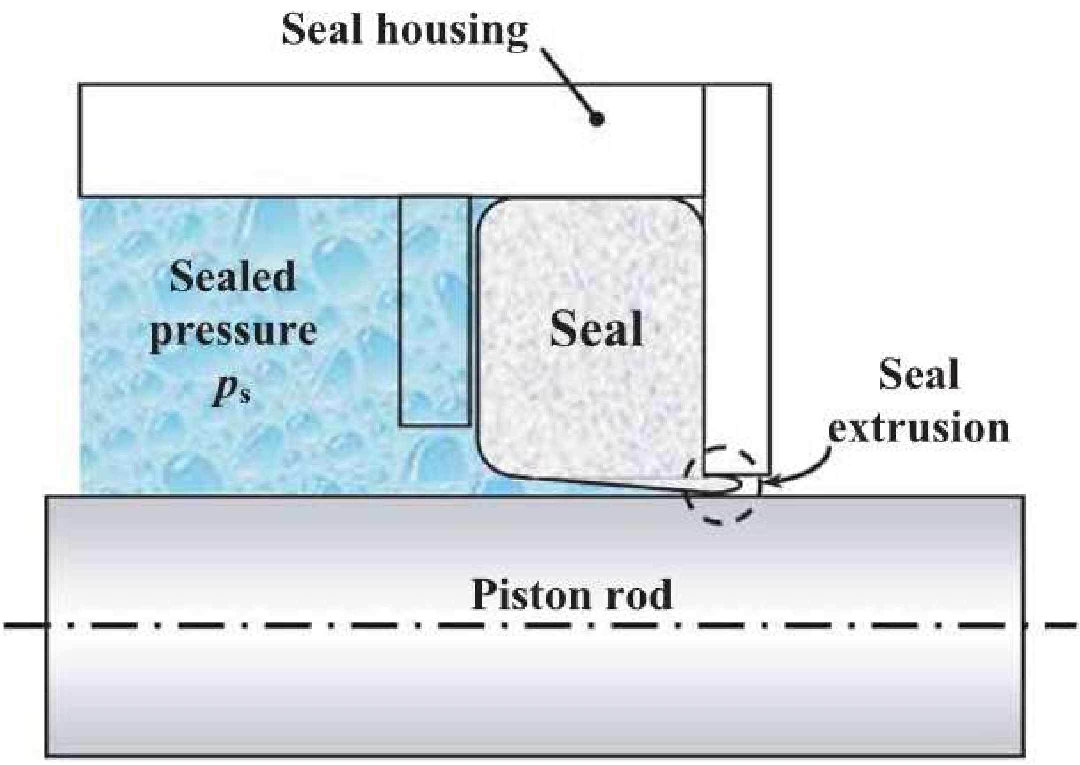
Sui et al. (1999) observed that during the phase of running-in wear leads to a rapid decrease of contact stress. After a period of high wear rate the decreasing of the radial load is almost linear. This behavior of radial load explains the friction behavior shown in picture 24 until cycle amount of 120000. The wearing out of the seals in endurance test one is so fast that this behavior is not clearly visible in picture 17.

Endurance test for PTFE material seals by Larsen et al. (2013, p. 67) showed the same kind of friction behavior as on endurance test 2. The amount of friction force decreases during the test when more cycles are run. The test cycle consisted of 200000 cycles of 280mm strokes at a speed of 0.2 m/s. The pressures were under 50 bar and 300 bar depending on the movement direction. The diameter of the rod was 50 mm and the test temperature 55 degrees Celsius. The test conditions are reasonably same as in the endurance tests that were conducted in this project. The test results can be seen as comparable.

The friction measurements seem to be reliable. The results are constant throughout the endurance tests. Still there can be seen some odd behavior. For example on picture 24 the zero friction level of 9649 cycles keeps changing. The force transducer was tested before the endurance test. To get more accurate result from fast high frequency effects the sampling frequency should be higher. Such effects are for example the force spike after the cylinder starts and stops moving. The measurement system that was available could not have a higher sampling frequency than the 500 Hz which was used. This limited the highest frequency in analysis to 250 Hz according to the Nyquist sampling theorem. The sampling theorem states that "we must sample at least twice as fast as the highest frequency component that we want to maintain after sampling". (Leis 2011, p. 82) Bhaumik et al. (2013, p. 838) state that the used test method for seal friction measurement is useable for comparative study of seal performance but do not provide the friction and leakage values for a single seal. Also the measurement data cannot be obtained separately for in-stroke and out-stroke. This is due to the structure of the test cylinder.

The T-seals made by Trelleborg had maximum continuous operation pressure of 210 bar so the test pressure was near the maximum. Maximum speed was 3 m/s or higher for noncontinuous or intermittent use. (Aerospace Sealing Systems product Catalog & Engineering Guide 2011, p. 138) The maximum speed of the endurance test was 0.52 m/s so it was not close to the maximum usage speed limit. Avoiding of extreme limits was advised. That kept in mind the circumstances in test were not difficult for the seals tested.

Myshkin et al. (2005) state that abrasive wear is related to cutting or plowing of the surface by harder particles or asperities. Abrasion displays scratches, gouges and scoring marks on the worn surface and the debris from abrasive wear is in the form of fine cutting chips similar to machining but only smaller. The visual inspection of the seals shows that the piston seals of manufacture 1 were much more worn than the piston seals of manufacturer 2. The round part of the elastomer that seals against the cylinder pipe has worn more flat on the manufacturer 1's seals. This kind of wearing can explain the leaking that was observed on the test. The O-rings of manufacturer 1 wore really badly. There was a lot of visible rubber material that has been ripped from the seal. On manufacturer 2's O-rings there were a little bit of same kind of wear visible but the seals still seemed fine after the test. This kind of wear can be caused due to seal extruding to the gap between the piston and cylinder pipe. According to Merkel (Merkel Technical manual Hydraulics 2016) seal materials act like a viscous fluid under the influence of the operating pressure. When the pressure is applied the sealing component is pressed closer to the metal housing and to the sealing gap. The ingress of the seal material into the sealing gap is referred to as the gap extrusion. The sealing component is damaged in the area of the metal edge of the installation space by the ingress of the seal material into the sealing gap. The repeated damage eventually causes failure of the sealing component. The seal extrusion is shown in picture 39. The gap extrusion behavior of the seals may be caused due to high pressure pikes or too large clearance between the cylinder pipe and cylinder end. As solutions to stop seal extrusion Flitney (2007) suggest for example smaller sealing gap, harder sealing material, use of backup rings, alternative seal design or alternative design of the equipment.



Picture 39 Seal extrusion (Nikas 2010)

One other explanation for this is that the cylinder pipe and cylinder end may be moving. This may be possible if the pretension of the tightening rods is too small. The needed pretension of the tightening rods was calculated during the designing process of the endurance test system. The tightening rods were tightened with a torque wrench. Still it is possible that the tightening may have been decreased during the endurance test. Either of these possibilities may have caused the behavior that leaded to the O-ring sealing to be torn. There was a very minor wear on the manufacturer 1's rod seals elastomer part. The elastomer was not as round as the new one. The wear was not as large as was visible on the piston seals. On manufacturer 2's rod seal there was no visible wear. The visual inspection and weighting results both show that the seals wore less on the endurance test 2 than on the endurance test 1. When the cycle amounts are taken into account it is clear that the seals used on the test 2 have much more wear resistance because the cycle amount that was run was nearly double on endurance test 2 compared to endurance test 1. One factor that also affects the seal wear is the surface finish quality of the surface that the seal slides on. Rana et al. (2001) state that after a certain quality of surface finish the friction is independent of how smooth the surface is. However if the surface is too rough the friction increases and the seal wears significantly faster. Surface quality of the rod was not measured. There is not observed noticeable smoothening of the rod. Thus the differences between tests one and two is not assumed to be caused by the change of surface quality due to smoothening of the surface.

The weighting results should be very accurate because of a high quality analytical balance. The seals were very carefully cleaned and dried before the weighting with a nearly lint free paper towels. Due to that the weighting results should be comparable. Because all the seals were not weighted before the endurance test their weight was evaluated to be same as another same kind of seal that was weighted for the absorption tests. Yet there can be small weight differences between same kinds of seals. Even when taken this uncertainty into account the differences on weighting results between the test 1 and 2 are so big that it seems clear that the seals of manufacturer 2 last wearing better. The difference of wearing can also be visually observed. The differences may be caused by differences in used material compounds. Other possibility is that the seals have slightly different measurements so it starts to wear differently.

The wear test by Larsen et al. (2013) for the seals lasted for 1000000 cycles and stroke length was 10mm. The velocity was informed as 1 Hz. The rod diameter was same 50 mm. The test cycle was different as on the friction test. The test pressure was a constant 200 bar. Test temperature was 80 degrees Celsius. After the wear test the seal profiles look almost as new without any signs of extrusion even after one million cycles. It is stated by Larsen et al. (2013) that weaknesses of virgin PTFE are the wear and deformation properties. The virgin PTFE was used for the support rings of seals on the endurance tests in this Master's thesis project. There was minor visible extrusion of PTFE on endurance test 2.

6. CONCLUSION

There has been observed deviation of endurance in both dynamic and static seals in the hydraulic system of an airplane which was suspected to be caused by differences in material and quality between manufacturers. In order to find out and verify these differences comparing tests were conducted on specifically made test equipment. The main goal of this Master's Thesis was to find out the endurance differences between the seals using the previously designed test equipment. The test was conducted in conditions resembling real life usage of the seals. The main point was that pressure, temperature and movement speed were the same as in the original application and remained the same throughout the testing process.

This Master´s thesis also aimed to find information about the chemical resistance of the sealing materials. This information is crucial in explaining the behavior of the seals. The fluid used on the test was MIL-PRF-83282 which is a synthetic micro filtered hydrocarbon based hydraulic fluid. Trelleborg suggests NBR to be used as elastomer material with this fluid. According to Trelleborg PTFE material is chemically inert in virtually all media, even at elevated temperature and pressures. (Materials Chemical Compatibility Guide 2012) PTFE does not absorb any other media than water to a significant level. The used virgin PTFE is physiologically inert. (Aerospace Sealing Systems product Catalog & Engineering Guide 2011, p. 18) It is stated by Larsen et al. (2013) that weaknesses of virgin PTFE are the wear and deformation properties. The virgin PTFE was used for the support rings of seals on the endurance tests in this Master's thesis project.

The chemical resistance of the seals was tested before the endurance test. The first absorption test was carried out in room temperature. The test time was according to standard 72 hours (ISO 1817 2005, p. 5). The seals were immersed in the same hydraulic fluid that is used on the actual usage of the seals. The second absorption test was carried out at elevated temperature to find out does the higher temperature effect on the fluid absorption of the seal. This test took four hours. Every T-seals sealings were made of NBR. All sealings lost slightly their mass in the 72 hour immersion test performed at room temperature. The mass changes of the Teflon rings were insignificantly small. On the contrary to the T-seals the O-rings gained mass on the immersion test. Also the proportional change on mass was larger than with the T-seals. One explanation to this is that the O-rings are made from a different material compound. The mass loss of rod seals was greater in the 4h elevated temperature test than in the 72 hour room temperature test. The mass loss of piston seals on the other hand was smaller than in the previous test. The mass change of Teflon rings stayed insignificantly small. The largest difference between these two immersion test results was the change in the manufacturer 1 O-ring. In test 1 the mass increased but in test 2 the mass decreased. The mass of the O-ring of manufacturer 2 increased also on the test 2 such as in test 1. The seals used in tests were unused and straightly taken from manufacturer’s plastic bags. Before and after immersing the seals in the hydraulic fluid the seals were weighted using an analytical balance. After the immersion test the seals were lifted out from the fluid and were laid on a nearly lint free paper towels. After drying for an hour the seals were dried well with paper towels and weighed.

The test cylinder was designed on the basis of the existing seals. The cylinder was designed to be able to test two sets of seals at the same time. One set consisted of a piston seal, a rod seal and an O-ring seal.

The measuring equipment consists of transducers. The seal friction and the deviation of friction behavior during the test were calculated from the force transducer data. Pressure transducers monitored the pressures in the hydraulic circuit. Displacement sensor measured the position of the cylinder. Movement speed of the cylinder was calculated from the displacement data. The test temperature needed to be as close as possible to the real usage conditions and it was monitored with a temperature transducer. Leakage container was installed under the leakage holes to catch the leaking fluid. A control logic was used to run the test in conditions that resemble real usage conditions. During the test cycle the cylinder was run continuously from one end to the other. Pressure was differentiated so that it resembled normal cylinder usage.

The endurance test for the seals of Manufacturer 1 was stopped after 108698 cycles due to a leak. The leak was observed to be about 2-6 drops of hydraulic fluid per minute from the middle leaking hole of the cylinder resembling a 12 ml leak per hour at an average. Because the leak was from the middle hole the leaking seals were the pistons seals. This amount of leakage was more than is acceptable in the actual usage system so the test was stopped at this point. Also the rod seals leakaged but the leakage was estimated to be less than 1 drop per minute. The endurance test for the seals of Manufacturer 2 was stopped after 214907 cycles. The amount of leakage was about 0.5 drops per minute resembling a 1.5 ml leak per hour at an average. The amount of leakage did not grow after 70000 cycles. At this point the test was stopped. The amount of cycles for the seals of manufacturer 2 were nearly two times the respective amount for Manufacturer 1. This was considered to be reliable enough and the test was not continued to the same amount of leak than the previous test. The continuous usage times for the system were between 45 minutes and 7 hours and 30 minutes. This led to the continuous cycle amounts between 1000 and 10000 cycles per day. This run time was due to saving the data every 1000 cycles which takes about 45 minutes to run

From the friction figures it can be assumed that the seals behaved in a similar way on the endurance tests but the seals of test 1 wore out much faster. From the figures of pressured state it can be assumed that the seals of manufacturer 2 reached the same amount of wearing at 120000 cycles as the seals of manufacturer 1 reached already at 40000 cycles. From the figures it can be seen that until these cycle amounts the amount of friction keeps decreasing and after reaching these the friction starts to increase. The increasing of friction can be assumed to be caused by the changing of hydrodynamic lubrication because of the wearing of the seals. The frictions behavior of depressurized state show decreasing friction on seals of the both manufacturers. In depressured state the hydrodynamic friction is not present because it depends on the pressure. The decreasing of friction can be explained by the decreasing precompression when the seal wears.

The friction measurement results are constant throughout the endurance tests. To get more accurate result from fast high frequency effects the sampling frequency should be higher. Such effects are for example the force spike after the cylinder starts and stops moving. The measurement system that was available could not have a higher sampling frequency than the 500 Hz which was used. This limited the highest frequency in analysis to 250 Hz according to the Nyquist sampling theorem (Leis 2011, p. 82).

The T-seals made by Trelleborg had maximum continuous operation pressure of 210 bar so the test pressure was near the maximum. Maximum speed was 3 m/s or higher for noncontinuous or intermittent use. (Aerospace Sealing Systems product Catalog & Engineering Guide 2011, p. 138) The maximum speed of the endurance test was 0.52 m/s so it was not close to the maximum usage speed limit. Avoiding the extreme limits was advised. That kept in mind the circumstances in test were not difficult for the seals tested.

The visual inspection showed that the piston seals of manufacture 1 were much more worn than the piston seals of manufacturer 2. The round part of the elastomer that seals against the cylinder pipe had worn more flat on the manufacturer 1's seals when compared to that of manufacturer 2. This kind of wearing can explain the leaking that was observed on the test. The O-rings of manufacturer 1 wore really badly. There was a lot of visible rubber material that has been ripped from the seal. On manufacturer 2's O-rings there was a little bit of same kind of wear visible but the seals still seemed fine after the test. The wearing on the rod seals was insignificant compared to the wear on the piston seals and O-rings.

The visual inspection and weighting results both show that the seals wore less on the endurance test 2 than on the endurance test 1. When the cycle amounts are taken into account it is clear that the seals used on the test 2 are much more wear resistant because the cycle amount that was run was nearly double on the endurance test 2 compared to the endurance test 1.

The weighting results should be very accurate because of a high quality Precisa EP 420A analytical balance. The seals were very carefully cleaned and dried before the weighting with a nearly lint free paper towels. Due to that the weighting results should be comparable. The differences between the manufacturers 1 and 2 weighting results after the endurance test were evident. Thus it seems clear that the seals of manufacturer 2 last wearing better. The difference of wearing could also be visually observed. The differences may be caused by differences in used material compounds. Other possibility is that the seals have slightly different dimensions so it starts to wear differently.

The fluid leakage was observed during the endurance test. Test temperature decreased during the endurance test 2 by 5 degrees Celsius. The cooling of 5 degrees explains a roughly 7 % change on the leakage flow. Cooling does not explain why the leakage did not increase to the same level than on the endurance test 1. The leakage on endurance test 1 was 12 ml per hour and on endurance test 2 1.5 ml per hour. The difference on leakage between the endurance tests is most likely due to differences on wearing of the seals.

All the exact material compounds and properties of the seals are not necessarily available during the procurement process. It can also be assumed that there are differences between the materials the different manufacturers use for their seals thus affecting the actual use of the seals. Thus critically important seals used in aviation need further research such as the present Master's thesis depicts. The properties of the seals between the two manufacturers differed in the present study and it seemed that the seals of manufacturer 2 lasted wearing better. This result is based on the leakage during the test, weighting after the test and friction behavior during the test. Also visual observation of the seals supports this finding.

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