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IMPROVEMENT OF HYDRAULIC OPERATIONS OF A STATIONARY INDUSTRIAL MACHINE WITH DECENTRALIZED HYDRAULIC SYSTEM UTILIZING ROTATIONAL SPEED-CONTROLLED PUMPS

Master’s Thesis 2020
Examiners: Professor Heikki Handroos
M. Sc. (Tech.) Iuliia Malysheva

TIIVISTELMÄ

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Hydraulisten toimintojen kehittäminen kiinteässä teollisuuslaitteessa hyödyntäen hajautettua hydrauliikkajärjestelmää ja pyörimisnopeusohjattuja pumppuja

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Tämä diplomityö käsittelee hydraulisten toimintojen kehittämistä kiinteässä teollisuuslaitteessa, jota käytetään esimerkiksi kaivosteollisuudessa. Diplomityön tarkoituksena oli tutkia mahdollisuuksia parantaa laitteen neljän hydraulisylinterin synkronointia ja parantaa turvallisuutta, energiatehokkuutta ja varaosien saatavuutta laitteen elinkaaren aikana. Muutos hydrauliikkajärjestelmään toteutettiin hajautetulla hydrauliikkajärjestelmällä, joka hyödyntää pyörimisnopeusohjattuja pumppuja.

Kirjallisuusselvityksessä perehdyttiin aiempiin tutkimuksiin hajautetuista hydrauliikkajärjestelmistä sekä pyörimisnopeusohjatuista hydrauliikkajärjestelmistä. Laitteen koko nykyinen hydrauliikkajärjestelmä analysoitiin ja hajautetusta hydrauliikkajärjestelmästä muodostettiin konsepti diplomityön kohteena olevalle teollisuuslaitteelle. Laitteen neljä hydraulisylinteriä valittiin konseptitasoa tarkempaan tarkasteluun. Neljälle sylinterille muodostettiin uusi hydrauliikkajärjestelmä, joka erotettiin muista laitteen toiminnoista. Testijärjestelmä rakennettiin, jotta uutta tekniikkaa voitiin vertailla nykyisen järjestelmän kanssa ja saada kokemusta tekniikasta jatkokehitystä varten. Sylinterien synkronointia ja pätötehon kulutusta mitattiin testauksen aikana. Samanlaiset mittaukset toteutettiin myös saman laitteen nykyiselle hydrauliikkajärjestelmälle.

Tulosten perusteella testijärjestelmässä sylinterit olivat paremmin synkronoituja kuin nykyisessä järjestelmässä. Testijärjestelmä vähensi tehonkulutusta kuitenkin vain yhdessä kolmesta vertailusta. Tämä johtui suurimmaksi osaksi siitä, että testijärjestelmän hydraulipumput eivät toimineet optimaalisella painealueellaan. Matalan painetason takia pumppujen hydromekaaninen hyötysuhde oli heikko. Jatkokehitystä varten esitettiin, että järjestelmän painetasoa kasvatetaan pienentämällä sylintereiden männän pinta-alaa. Testijärjestelmän kaltaisella järjestelmällä voidaan parantaa varaosien saatavuutta ja huollettavuutta, koska käytetään standardikomponentteja ja samanlaisia pumppuyksiköitä.

ABSTRACT

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Improvement of Hydraulic Operations of a Stationary Industrial Machine with Decentralized Hydraulic System Utilizing Rotational Speed-Controlled Pumps

Master's thesis

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98 pages, 45 figures, 23 tables and 7 appendices

Examiners: Professor Heikki Handroos
M. Sc. (Tech.) Iuliia Malysheva
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Keywords: decentralized hydraulic system, rotational speed-control, product development

This master's thesis discusses improvement of hydraulic operations of a stationary industrial machine that is used in mining industry. Objective of the thesis was to study possibilities to improve synchronization of four cylinders of the machine and improve safety, energy efficiency and spare part availability during the lifetime of the machine. Improvement of the system was done with a decentralized hydraulic system that utilizes rotational speed-controlled pumps.

In literature review of the thesis previous studies of decentralized hydraulic systems as well as rotational speed-controlled hydraulic systems were studied. Based on the findings the whole hydraulic system of the machine was analyzed and a new decentralized hydraulic system was constructed on a concept level. Four cylinders of the machine were chosen for in-depth analysis. Based on literature findings a new concept for the cylinders was developed. A test system was manufactured to gain experience of the technology and compare it with the current hydraulic system. Synchronization of the cylinders was measured as well as active power consumption. The same measurements were done for the same machine with the current hydraulic system.

Based on the results the developed system is more reliable in synchronization of the four cylinders than the current system. However, the test system reduced power consumption only in one of the three comparisons between the systems. This was mainly because of the poor hydromechanical efficiency of the pumps on a low pressure level. It was suggested that the pressure level of the system is increased by reducing the area of the piston. By increasing the pressure level, the new system has potential for energy savings. Additionally, maintenance of the machine can be improved by using standard components and same motor-pump units in all of the cylinders.

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LIST OF SYMBOLS AND ABBREVIATIONS

A	Area of the piston [m^2]
A_{in}	Area of the piston on the working side [m^2]
A_{out}	Area of the piston opposite to the working chamber [m^2]
η_{hm}	Hydromechanical efficiency
η_t	Total efficiency
η_v	Volumetric efficiency
F	Load [N]
P	Power [W]
p	Pressure [Pa]
p_{out}	Pressure of the chamber opposite to the working chamber [Pa]
q	Volume flow [m^3/s]
v	velocity [m/s]

DDH	Directly driven hydraulics
DV block	Directional valve block
EHA	Electro-hydraulic actuator
EHA-FD	Fixed displacement electro-hydrostatic actuator
EHA-VD	Variable displacement electro-hydrostatic actuator
EMA	Electro-mechanical actuator
EMAS	Electro-mechanical actuator solution
HAS	Hydraulic actuation system
HPU	Hydraulic power unit
HSA	Hydraulic servo-actuator
LC block	Locking cylinder block
LEHG	Local electro-hydraulic generation
PF	Pressure filter
QAC	Quick action cylinder
QAC block	Quick action cylinder block
SC block	Sealing cylinder block
VFD	Variable-frequency drive

1 INTRODUCTION

This master's thesis' background is the need to develop the hydraulic system of Outotec Larox PF type filters. There have been challenges with the current hydraulic system even though the current hydraulic system generally functions properly. The most difficult challenge with the current hydraulic system of PF filters is the synchronization of four quick action cylinders. It is essential that the cylinders are fully synchronized to ensure the reliable and safe operation of the pressure filter. Other targets of development include improving energy efficiency, safety, maintenance and spare part availability.

Improving energy efficiency of industrial machines is becoming more important as concern of high energy usage in industrial plants increases. It is common for PF type filters to operate around the clock. Therefore even a small reduction in electricity consumption can make a difference. By improving the hydraulic system's energy efficiency, Outotec can help their customers to save energy during the lifetime of the filter.

Safety of PF type filters can be improved with changes to hydraulic system. At the moment, hydraulic system includes a lot of pipelines. If the system can be modified so that hydraulic unit is closer to the application where hydraulic power is needed, the length of hydraulic pipelines can be significantly reduced. Another safety aspect is the hazard that is caused by asynchronization of cylinders during opening or closing the plate pack. New hydraulic system should be designed so that there is no possibility for asynchronization. In addition, the current separate hydraulic power unit (HPU) requires its own place somewhere close to the pressure filter. The amount of hydraulic fluid in the HPU's tank is significant (800 liters). Therefore, reducing the amount of hydraulic fluid is an important basis for re-designing the hydraulic system as used hydraulic fluid disposal can cause an environmental hazard.

The availability of spare parts for maintenance is another aspect that could be improved. The current hydraulic system contains components that are special, so the availability of spare parts is challenging. The best option for new system would be to use standard components with good availability. Perspective of maintenance operations must be also taken into account, such as location of hydraulic components. In addition, troubleshooting of the system should be improved.

1.1 Research problem, goals and framing

Research problem is the utilization of decentralized hydraulic system to solve the challenges of the current hydraulic system of PF filters. Research question is:

- Does the decentralized hydraulic system with utilization of speed-controlled hydraulic pumps change PF filters into more energy efficient?

In addition to the main research question, there are subsidiary research questions. The subsidiary research questions are:

- Is it possible to design the hydraulic circuit of opening and closing the plate pack so that the cylinders are always synchronized and operation is accurate?
- Can the number of different spare parts in hydraulics be reduced?
- Is troubleshooting during maintenance operations faster compared to the current system?

Aim of the thesis is to solve the challenges the current hydraulic system has. A new concept of the hydraulic system is developed theoretically. To frame the subject, the circuit of opening and closing the plate pack is chosen for in-depth analysis. For this hydraulic circuit a new construction is created and a method to measure the synchronization of four cylinders is developed. Finally a test assembly of the new hydraulic circuit is manufactured for the plate pack opening and closing movement. This test assembly is then fitted to a PF filter and consumption of power is analyzed. Similar analysis is done for the same PF filter with the current hydraulic system.

1.2 Research methods

First, literature is studied to find the theory related to speed controlled and decentralized hydraulic systems. Based on literature findings a new concept is developed for hydraulic system of PF filters. The aim is to find out the methods to reduce energy consumption and improve operation of the filter. New circuit of opening and closing the plate pack is tested on practice and active power is measured. Same measurements are done for the current hydraulic system of the same machine. Finally, the results are compared to find out the differences between the systems. The tests are conducted with the same machine, so the results can be compared. Standard deviations of the measured values are calculated and presented along the results.

1.3 Structure of the thesis

First, literature is studied to find the related studies and theories. Literature studies include rotation speed-controlled hydraulic systems and decentralized hydraulic systems. Then, the current hydraulic system of PF filters is explained. A new concept for hydraulics is then assembled based on previous studies and related theories. The new concept is developed furthermore for the circuit of opening and closing the plate pack. A test unit is manufactured for testing the new system in practice. The purpose for experimental tests is to compare the energy consumption of the new circuit and the current circuit. The possible benefits for synchronization of four cylinders of the plate pack are also evaluated during the tests. Finally the results are analyzed and conclusions are made. Recommendations are given for further studies.

2 MODERN HYDRAULIC SYSTEMS

This chapter of the thesis includes literature research related to rotation speed-controlled hydraulic systems and decentralized hydraulics.

2.1 Rotation speed-controlled hydraulic systems

Traditionally, the pump of hydraulic system produces too much volume flow into the system than it is required. This is because pressure control valves and flow control valves are used to control hydraulic actuators' force, torque and speed. When pressure and flow control valves are used, the excess volume flow is guided back to the oil reservoir. This then causes a power loss because of the excess volume flow. More losses usually occur from required cooling power that is needed for active cooling of hydraulic fluid. As efficiency of hydraulic systems is important, producing excess volume flow should be avoided. (Kauranne, Kajaste & Vilenius 2013, p. 458.)

A variable-displacement pump can be used to avoid producing excess volume flow to the hydraulic system. A variable-displacement pump has been a common alternative for replacing valves to create a pump-controlled system. Another method for producing a pump-controlled system is to adjust the rotation speed of the motor that drives the hydraulic pump. Rotation speed-controlled hydraulic pumps have become increasingly popular because of their advantages. These advantages include the following viewpoints:

- Possibility to increase efficiency compared to a system utilizing a variable-displacement pump
- Efficiency can be high in a wide range of rotation speeds
- Electrical pump-controls are versatile compared to hydraulic pump-controls
- Regeneration of hydraulic energy is possible
- Low-cost fixed displacement pumps can be used
- Noise levels can be lower than in a variable-displacement pump system because average rotation speeds are lower
- Energy costs can be 30 % to 80 % lower than in variable-displacement pump systems
- Complex control system enables versatile controls and adjustment. (Kauranne et al. 2013, p. 458 – 461.)

Despite advantages, rotation speed-controlled hydraulic pump systems have some weaknesses. Complex control system can be seen as an disadvantage, depending on application. Dynamic properties of the system can be minor compared to variable-displacement pump systems because of ratio of control elements and actuating forces. (Kauranne et al. 2013, p. 458 – 461.) Lovrec and Ulaga (2007) studied the dynamic behavior of variable displacement pump system and constant displacement and variable speed pump system. As a result of the study it was stated that the dynamic response ratio does not correspond the ratio between control elements (rotational parts in variable-speed system and swash plate in the variable-displacement pump). According to the study, the response time of the variable-displacement pump system was four to five times faster compared to the system utilizing variable-frequency drive (VFD) and constant displacement pump. (Lovrec & Ulaga 2007, p. 33 – 41.)

Sometimes when the pump is rotation speed-controlled, the pump can be oversized. Oversizing is due to the required volume flows that occur during operation. The volume flows must be feasible between the permissible rotating speeds. This problem can be eliminated if it is allowed instantaneously to exceed the permissible rotating speed. (Kauranne et al. 2013, p. 458 – 461.)

2.1.1 Drives for speed-controlled pumps

There are several possibilities for the type of the driving motor. Driving motor can be a servomotor, asynchronous servomotor or asynchronous motor. If asynchronous motor is used, motor is controlled with a frequency converter. Servomotors are controlled with servo controllers. (Kauranne et al. 2013, p. 461.) Asynchronous motors (induction motors) are used in the industry on a large scale. Induction motors can be divided into two types: wound-rotor motors and squirrel-cage motors. (Tan & Putra 2011, p. 57.)

Squirrel-cage motors are popular AC motors. They are durable and the construction is simple. (Tan & Putra 2011, p. 60 – 61.) Squirrel-cage motors can be controlled with VFDs. Normally the rotating speed of a squirrel-cage motor is proportional to the input power's frequency. One factor that must be reduced is the slip that changes according to torque load. With a VFD, it is possible to alter the rotating speed of the motor. VFD helps to save energy, as the speed of the motor can be chosen based on the actual need. VFD is a device that utilizes power electronics. A typical VFD consists of four sub-systems. The sub-systems are an AC/DC converter, a DC bus, an inverter and a control system. The output is AC at a specific frequency and voltage. (Dieckmann, McKenney & Brodrick 2010, p. 60 – 61.)

2.1.2 Comparison of different systems

The differences between the widely used variable-displacement pump systems and increasingly popular variable-speed pump systems have been studied by Lovrec and Ulaga (2007). In their experimental study, two systems were compared; fist system had a constant speed asynchronous motor and a variable axial piston swash plate pump, the second system had the same asynchronous motor that is driven by a VFD and the pump was a constant external gear pump. The other parts of the test gear were the same in both systems, to find the actual differences between the two pump systems. Three different pipeline lengths were used in the experiment, to demonstrate hydraulic capacity and inductivity. The authors focused on dynamic response differences between the two systems. Based on the study, the authors impugn the general assumption of the dynamic response of the variable-speed system. The dynamic response time of the first system was 4–5 times faster than the second system's response time, as mentioned earlier. The response time of the first system should have been much faster if the general assumption of the ratio of rotating masses was accurate. (Lovrec & Ulaga 2007, p. 33 – 41.)

According to Lovrec and Ulaga (2007), the second system had higher pressure pulsations, but it was because of the applied signal conditioning. The authors suggested to use a VFD that allows an external control loop. Energy savings are possible with the second system because efficiency of constant pumps is higher than of variable pumps and rotating speed of the motor can be decreased as required during the work cycle. The authors suggested the use of internal gear pumps because of their reliability and relatively low price. It should be noted that if there is pressure load present during the idle running phases, extra wear of the pump is possible if there is no hydrodynamic oil film. (Lovrec & Ulaga 2007, p. 33 – 41.)

2.2 Decentralized hydraulic systems

This chapter focuses on decentralized hydraulic systems. Principle of a decentralized hydraulic system is to break the hydraulic system into individual sub-systems. A conventional centralized hydraulic system has one hydraulic power unit that delivers the hydraulic fluid to the metering valves. However, metering valves produce excessive heat to the hydraulic fluid. Cooling of the fluid consumes energy, that can be seen as wasted energy. Efficiency of industrial machines is important because of environmental impacts and energy costs. In centralized hydraulic systems, a lot of energy is often wasted. (Jalayeri, Imam, Tomas & Sepehri 2015, p. 1 – 2; Quan, Quan & Zhang 2014, p. 337.)

Decentralized hydraulic systems have been developed in mobile machines for a long time, but they have become increasingly popular in stationary industrial machines as well. Decentralizing the hydraulic system means that the system is actuator specific and has individual motors and pumps for each actuator. A decentralized hydraulic system is often pump controlled. (Koitto et al. 2018, p. 348 – 349.)

2.2.1 Open and closed pump controlled circuits

Hydraulic circuits can be valve controlled or pump controlled. Valve controlled circuits have been widely used, but they consume a lot of energy. Pump controlled circuits have become the focus of development, because energy efficiency can be increased compared to valve controlled circuits. Pump controlled circuits can be divided into open and closed circuits. In an open circuit, pump operates only in one direction. (Quan et al. 2014, p. 337.) Basic structures of an open circuit are presented in figure 1.

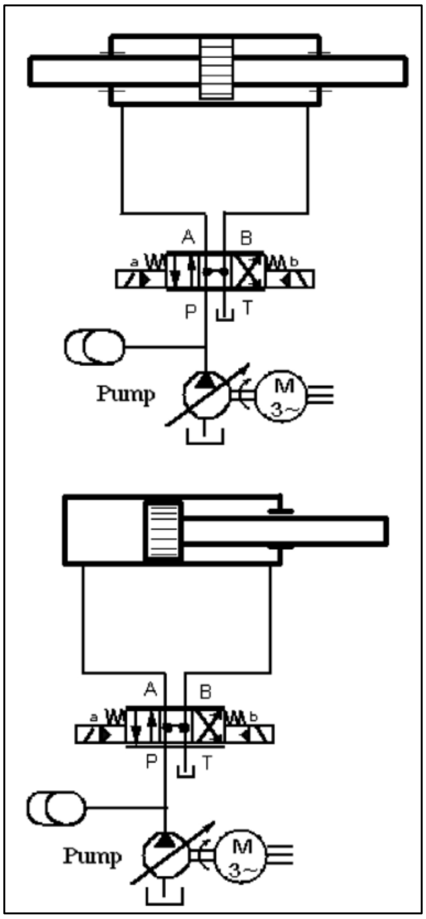


Figure 1. Basic structures of an open circuit (modified Quan et al. 2014, p. 338).

The topmost scheme in figure 1 presents an open circuit for a double-rod hydraulic cylinder. The lowest scheme in the same figure is for a single-rod hydraulic cylinder. An open system requires valves to change the direction of flow. Therefore, the efficiency of an open circuit is reduced. Closed circuit (direct pump-controlled system) has a pump that can operate in two directions. This, in case of a cylinder, enables that cylinder’s chambers are directly connected to the pump. (Quan et al. 2014, p. 337.) Basic structures of a closed circuit are presented in figure 2.

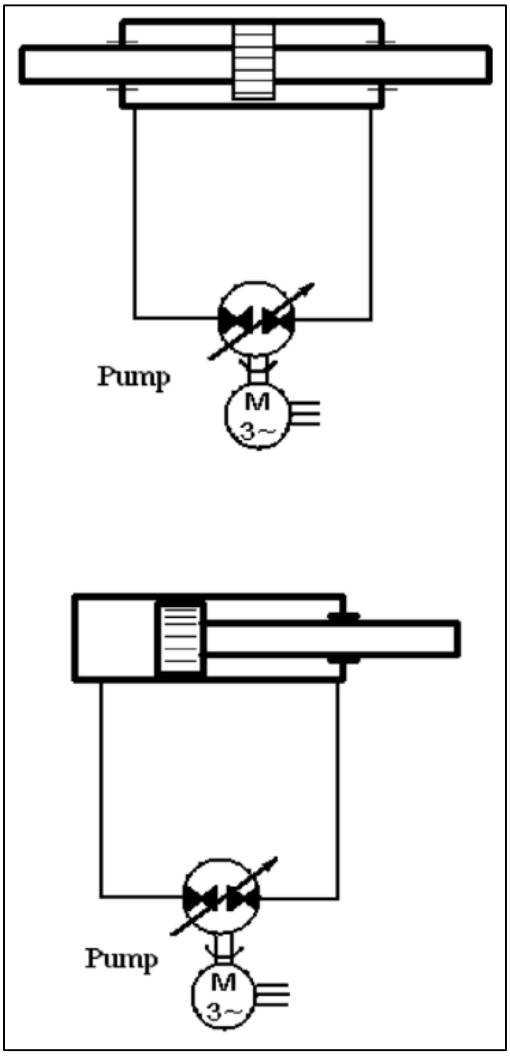


Figure 2. Basic structures of a closed circuit (modified Quan et al. 2014, p. 338).

The topmost scheme in figure 2 presents a closed circuit for a double-rod hydraulic cylinder. The lowest scheme in the same figure is for a single-rod hydraulic cylinder. Efficiency of a closed circuit can be higher compared to an open system because valves are not required to control the direction of flow. (Quan et al. 2014, p. 337 – 338.)

In a decentralized hydraulic system, hydraulic actuators are either variable-displacement pump or variable-speed pump controlled. A decentralized hydraulic system can combine the best characteristics of hydraulic and electric technology. For example, hydraulic piping can be eliminated because power is delivered to the motors by wire. (Jalayeri et al. 2015, p. 1 – 2; Altare & Vacca 2015, p. 8 – 11.) Variable-speed pump controlled hydraulic actuators are also sometimes called throttle-less hydraulic actuators, because no centralized hydraulic power unit is present and metering valves are not required. Each throttle-less hydraulic actuator needs its own hydraulic pump. It is easier to design a throttle-less hydraulic circuit for a double-rod hydraulic cylinder than for a single-rod hydraulic cylinder, because of the equal sectional areas of chambers on the both sides of the cylinder. (Jalayeri et al. 2015, p. 1 – 2.) Single-rod cylinder (differential cylinder) does not have equal flow from the ports. Therefore, research has been focusing on single-rod cylinders. When a single-rod cylinder is direct pump controlled (closed circuit), asymmetric flow can be a problem. In order to avoid problems such as positioning inaccuracy and unsatisfactory control, the flow must be compensated. In addition, energy efficiency decreases because of the asymmetric flow if the flow is not compensated. (Quan et al. 2014, p. 339 – 340.)

2.2.2 Concepts for actuators

Conventional hydraulic actuation systems (HAS) are developing towards more electrical systems. This leads to the development of electro-hydraulic actuators (EHA). EHAs combine the good power to weight ratio of hydraulics and the diverse properties of electrical technologies. A typical EHA contains at least an hydraulic actuator (for example a cylinder), a fixed-displacement or variable-displacement pump and an electric motor. Another direction of development are electro-mechanical actuator solutions (EMAS) that do not have hydraulic components, only mechanical and electrical technology are combined. EMAS suffer from lower power to weight ratio than systems utilizing hydraulics. For example, the aerospace field has been using EHAs to increase the amount of electrical technology, but to retain the good characteristics of hydraulics. (Altare & Vacca 2015, p. 8 – 11.) Development directions of actuators are presented in figure 3.

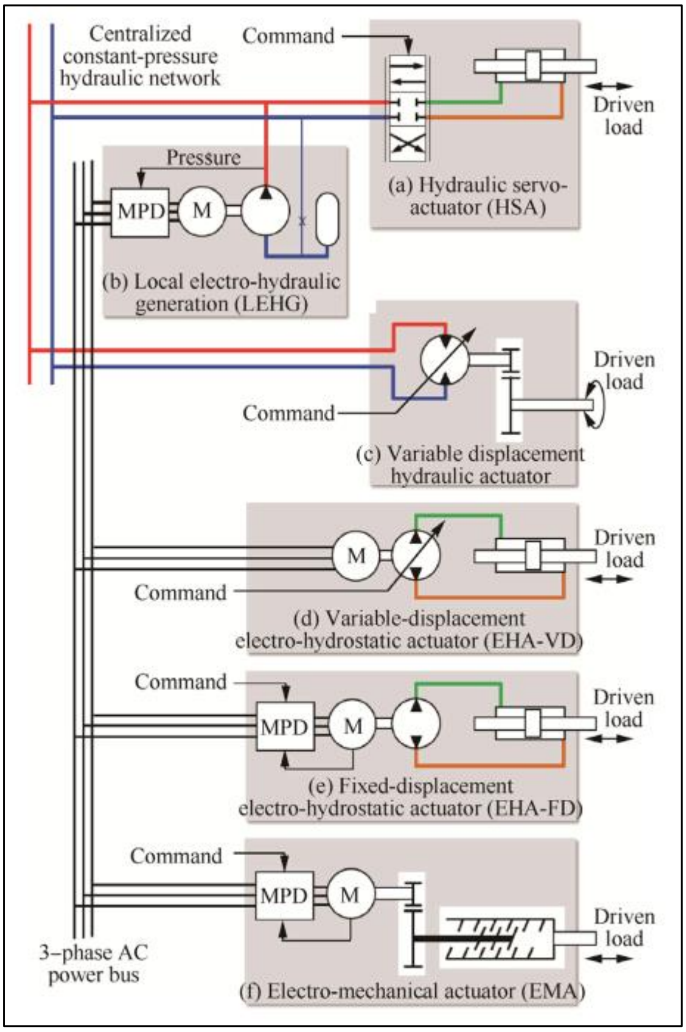


Figure 3. Directions of development of actuators (Maré & Fu 2017, p. 862).

In figure 3, development directions of actuators are divided to six parts (a – f). At the top of the figure, actuators include only hydraulic technology. The amount of hydraulics declines to zero on the way to the bottom of the figure. Part (a) of the figure presents hydraulic servo-actuator (HSA), that utilizes centralized hydraulic power unit. As this system requires metering valves, losses are significant. Part (b) of the figure upgrades the power density of the power distribution system by locally producing the power to the actuator. This concept is called local electro-hydraulic generation (LEHG). Throttling losses can be eliminated with concept (c) called variable-displacement hydraulic actuator. In this concept, a variable-displacement pump is used to control the flow instead of valves. Concept (d) is a variable-displacement electro-hydrostatic actuator (EHA-VD). These actuators do not require centralized hydraulic network because each actuator has an own motor and pump. According to the authors, EHA-VDs have some advantages from electric systems, but they still suffer from low efficiency. Better alternative for them are fixed-displacement electro-hydraulic actuators (EHA-FD) presented in part (e) of the figure. In an EHA-FD, rotating speed of the motor is altered to adjust the volume flow. EHA-FDs can have high efficiency and advantages of both hydraulic and electric systems are combined. Part (f) of the figure presents electro-mechanic actuator (EMA) that does not have any hydraulics. In conclusion, figure 3 presents the general concepts for actuators. The different actuating concepts can be combined to create a decentralized power distribution system. (Maré & Fu 2017, p. 862 – 866.)

2.2.3 Examples of pump-controlled hydraulic circuits

There have been many different approaches to replace a centralized hydraulic system with a pump-controlled closed circuit. Closed circuits have been more interesting in many studies because of their possibility to reduce energy consumption further compared to open circuits. First, a throttle-less hydraulic circuit for a single-rod hydraulic cylinder is presented. Secondly, a direct-driven hydraulic circuit with load compensating is presented.

Challenge with throttle-less hydraulic circuits for a single-rod cylinder is to connect the other chamber to the hydraulic circuit's low pressure side. Jalayeri et al. (2015) wanted to overcome the issues related to single-rod cylinders and created a trouble-free throttle-less hydraulic circuit. (Jalayeri et al. 2015, p. 2.) The circuit is presented in figure 4.

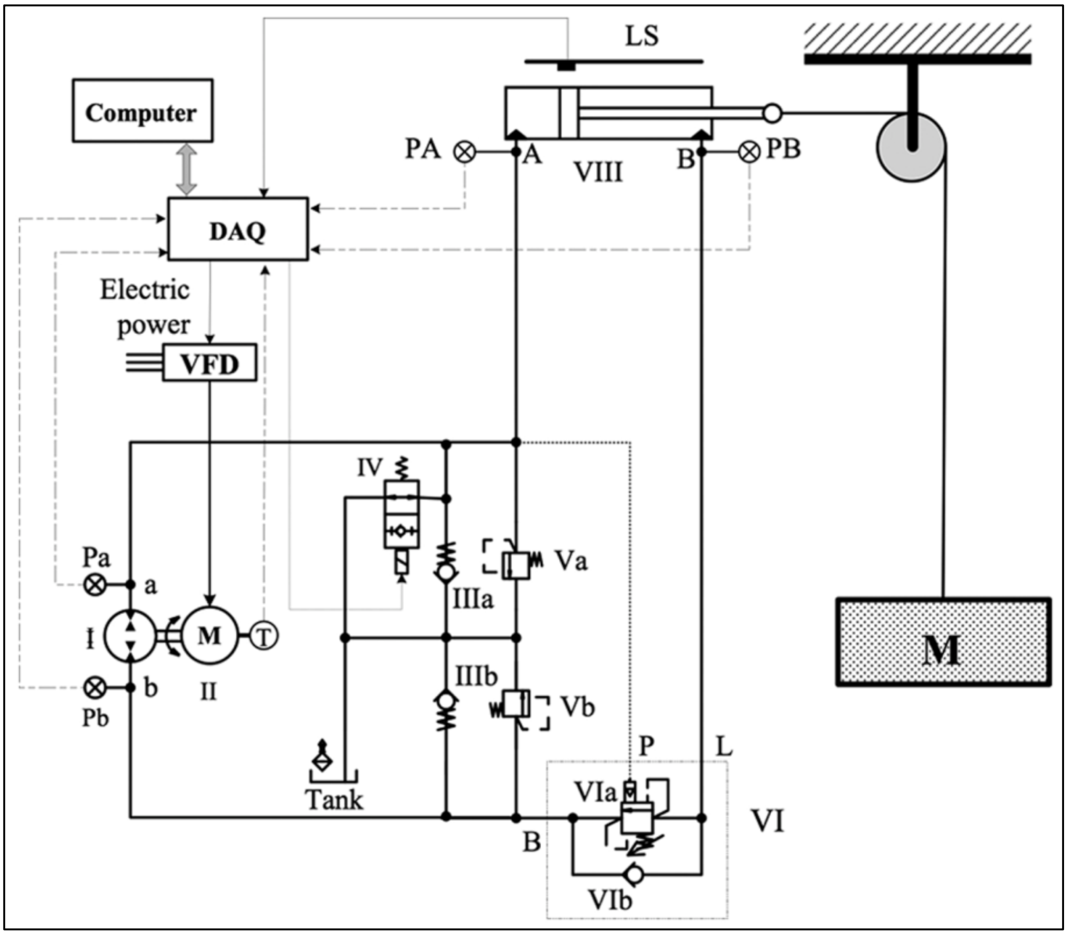


Figure 4. A throttle-less hydraulic circuit for a single-rod hydraulic cylinder (Jalayeri et al. 2015, p. 5).

In figure 4, I is a fixed displacement gear pump that is driven by an AC induction motor (II) with a VFD. IIIa and IIIb are check valves whereas IV is an on/off valve. On/off valve and check valves control the uneven flow rates from the chambers of the single-rod hydraulic cylinder (VIII). Va and Vb are pressure relief valves which are intended to protect the system from overpressure. VI is a counterbalance valve consisting of an adjustable relief valve (VIa) and a check valve (VIb). The counterbalance valve is important for the system as it prevents the direct connection from cylinder’s B port to the tank. It also prevents the load from falling. (Jalayeri et al. 2015, p. 4 – 8.) Controlling of the throttle-less hydraulic circuit is presented in figure 5.

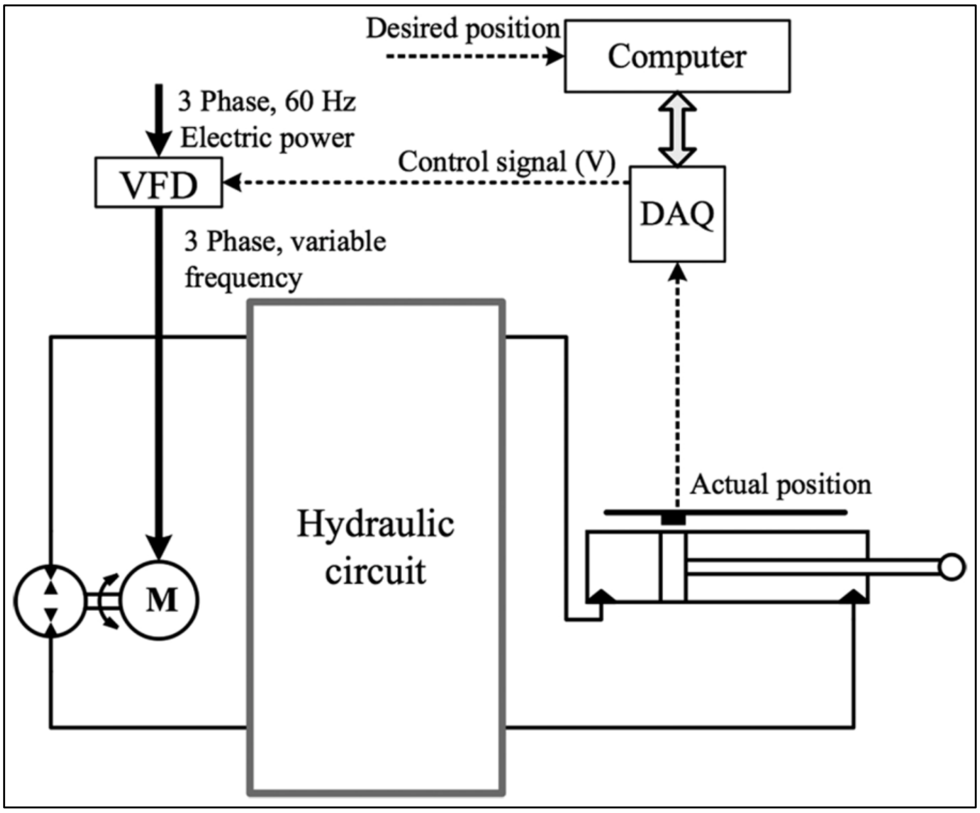


Figure 5. Controlling of the hydraulic circuit (Jalayeri et al. 2015, p. 7).

As presented in figure 5, control structure of the single-rod hydraulic cylinder is not complicated. Displacement of the cylinder is the only variable that required measuring. Jalayeri et al. reported an accuracy of ±0,034 mm for the position response. According to the researchers, the response was also consistent and repetitive. Compared to a similar valve-controlled hydraulic circuit, the energy consumption was low as it was 21 % of the energy consumed by the valve-controlled circuit. Efficiency of the throttle-less circuit was 60 % as efficiency of a similar valve-controlled circuit is less than 31 %. To find the importance of the counterbalance valve, tests were performed without it. The system was not as controllable without the counterbalance valve and the power consumption was higher than when the counterbalance valve was included to the system. (Jalayeri et al. 2015, p. 7 – 13.)

Koitto et al. (2018) studied a DDH (directly driven hydraulic) system that includes a load compensating circuit. The purpose of the research was to replace a traditional valve-controlled circuit with a more energy efficient system. The system has a hydraulic cylinder that is meant to move a mass between two points in vertical direction. The old system had a variable-displacement pump that loaded a hydraulic accumulator. The pump was producing volume flow only when the accumulator was filled, because the volume flow for the cylinder came from the accumulator. The old system was replaced with a DDH system that is presented in figure 6.

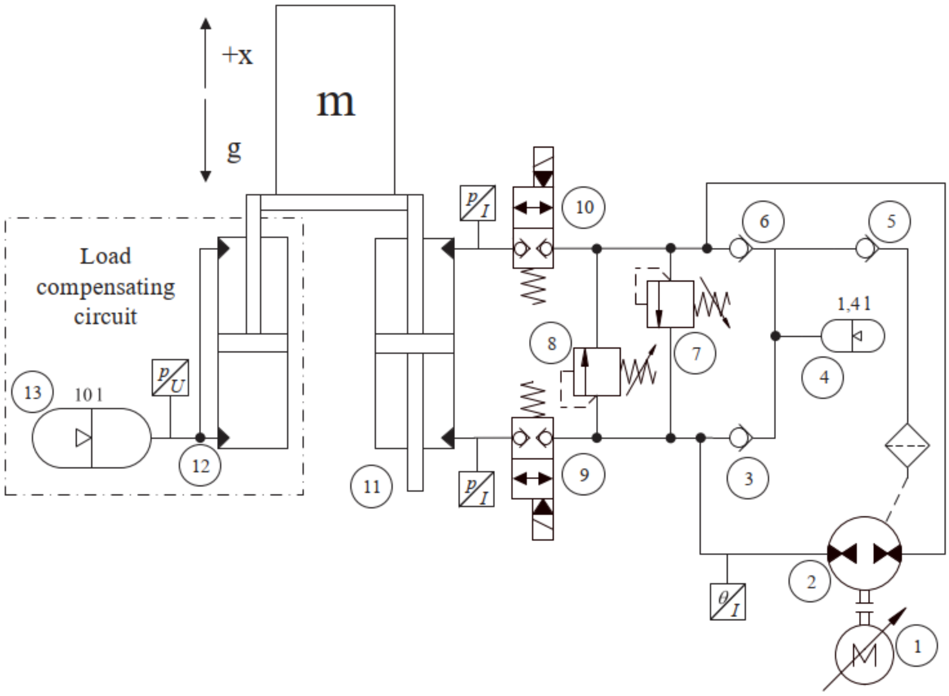


Figure 6. DDH system with load compensating circuit (Koitto et al. 2018, p. 351).

The system presented in figure 6 includes a double-rod hydraulic cylinder (11) as a main cylinder and an additional differential cylinder (12) in the load compensating circuit. The purpose of the load compensating circuit is to create a vertical force with the differential cylinder by converting the pre-charge pressure of a accumulator (13). The force is opposite to the gravitational force caused by the mass (m). The direction and speed of the main cylinder is controlled with the pump (2) by adjusting the rotating direction and rotating speed of the motor (1). There is also an accumulator (4) that is meant to be a buffer if there is a minor external leak in the system. The DDH system was experimentally tested and compared with the traditional system that was simulated. The reduction of energy input was 49 % compared to the old system. The reduction was 74 % when full load compensating was used. The results showed that there is a significant possibility to decrease energy consumption of industrial hydraulic systems. (Koitto et al. 2018, p. 348 – 359.)

3 HYDRAULIC SYSTEM OF OUTOTEC LAROX PF60 FILTER

This chapter discusses the hydraulic system of Outotec Larox PF60 filter. First the current system and the challenges are explained and the new decentralized hydraulic concept is developed. Hydraulics of plate pack’s opening and closing is covered on a detail level, because a hydraulic test unit is manufactured for that function.

Outotec Larox PF series filters are vertical filter presses that are used to separate solids and liquids on various industrial fields, such as mining, chemical and pharmaceutical industries. Filtration takes place in the plate pack that consists of filter plates and filter cloth. The slurry is fed into the plate pack and the cloth separates liquids and solids. A membrane is used for pressing the slurry. Membrane is pressed with water or air depending on filter model. The solids form a cake on top of the cloth. The cloth acts as a conveyor for the cake when the cake is discharged from the plate pack. (Kaipainen 2018.)

Hydraulics are in charge of the mandatory motions of the PF filter. However, hydraulics are not taking part in the actual filtration process. The amount of hydraulic operations varies depending on the filter model. The product portfolio of Outotec includes four different PF models: PF1.6, PF12, PF15 and PF60. In addition, the total filtration area can be chosen individually for each model by choosing the desired amount of filter plates. A PF60 type filter is chosen for analysis in this thesis because it is the largest model of PF type filters and it has the greatest number of hydraulic operations. Therefore, it also has the highest energy consumption of the PF type filters. (Kaipainen 2018.)

3.1 The current hydraulic system of PF60

The hydraulic operations of Outotec Larox PF60 filter are presented in table 1.

Table 1. Hydraulic operations of PF60 filter.

Hydraulic operations in PF60 filter (standard model)	
operation	**actuators**
opening and closing the plate pack	four double-acting double-rod cylinders
sealing the plate pack for filtration	eight double-acting single-rod cylinders
driving of the filter cloth	four to eight hydraulic motors
centering the filter cloth (cloth tracking)	one double-acting single-rod cylinder
securing the plate pack when open or closed	two double-acting single-rod cylinders
tensioning the filter cloth	one hydraulic motor
actuators for process valves (pinch valves)	seven double-acting single-rod cylinders

As presented in table 1, PF60 filter has seven different hydraulic operations. Hydraulic actuators of the filter include cylinders and motors. All the cylinders are double-acting and most of the cylinders are single-rodded. The only double-rod cylinders are used to open and close the plate pack.

Outotec Larox PF60 filter has a centralized hydraulic system. A hydraulic power unit (HPU) delivers power to all hydraulic actuators of the filter. The HPU has a tank for hydraulic oil. Volume of oil in the tank is 800 liters (does not include the amount of oil in the actuators and pipelines). The HPU has a 90 or 110 kW electric motor to drive the variable-displacement pump. This hydraulic pump is responsible to deliver the volume flow for pressure line P1 that is the main pressure line in the hydraulic system. The HPU also has a second hydraulic pump to deliver volume flow for pressure line P2. Line P2 has a hydraulic accumulator to store hydraulic energy for a case of a power failure. For safety, it is necessary to close all process valves during a power failure. Therefore a accumulator must be included in the line P2, where all process valves are connected.

The separate HPU is connected to the PF60 filter with steel pipes or hydraulic hoses, depending on the filter’s installation site. The hydraulic lines (P1, P2 and T) from the HPU are connected to directional valve block (DV block), that has directional valves to control the directions of flow for different hydraulic operations. A scheme of the DV block is presented in figure 7.

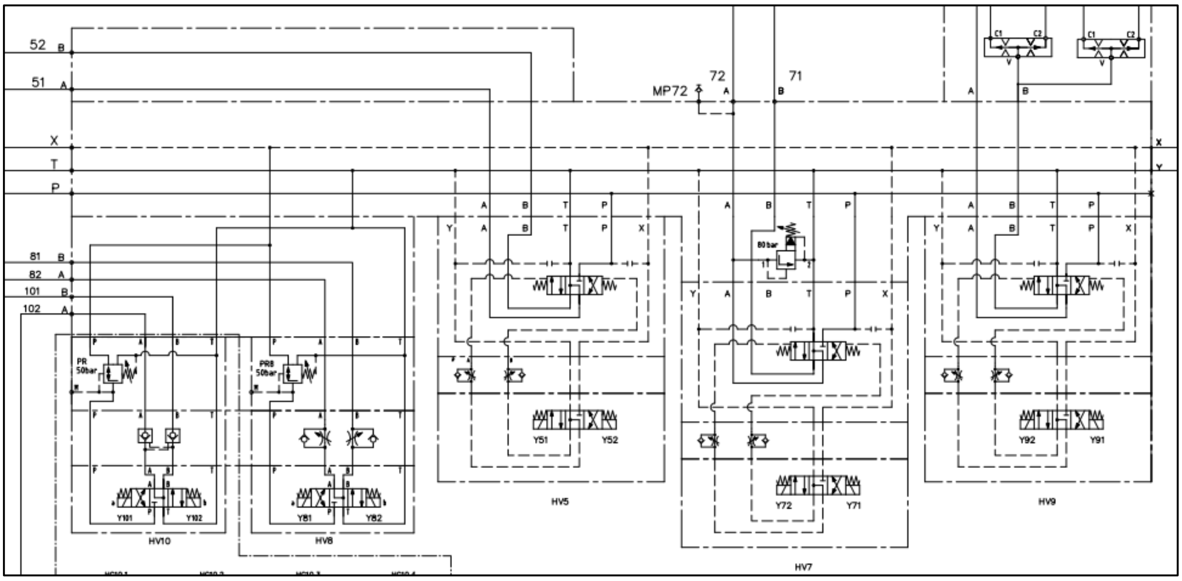


Figure 7. DV block of PF60 filter.

DV block is presented in figure 7. DV block has five directional valve units. HV9 is for the cloth driving motors, HV7 for sealing cylinders, HV5 for plate pack's quick action cylinders and HV8 for cloth tensioning device. An additional directional valve HV10 is included if the filter includes optional device called cake chute flap.

3.1.1 Filter cloth drive

The filter cloth is driven at the end of the filtration cycle when the cake is discharged and the cloth is washed. Directional valve HV9 is used to control the flow for the cloth driving motor circuit. Cloth driving motors are situated in the plate pack and in the cloth driving unit of the filter. A scheme of the filter cloth drive is presented in figure 8.

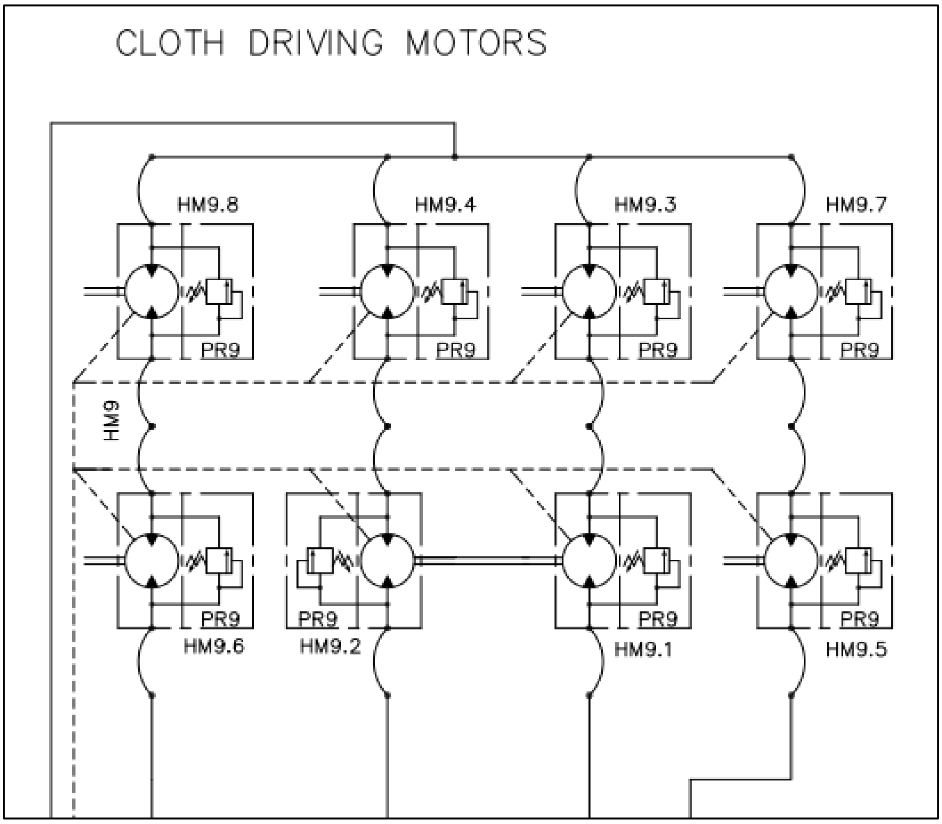


Figure 8. The filter cloth drive motors.

The motors HM9.1 to HM9.8 presented in figure 8 are hydraulic motors that rotate rollers of the cloth. The maximum number of filter cloth drive motors is eight. The amount of driving motors is determined based on process conditions and the amount of filter plates in the filter. As presented in figure 8, the motors HM9.1 and HM9.2 are driving the same roller called the main drive roller. The motors are located in both ends of the main drive roller and rotate with the same speed. Therefore, the flow is equal between both flow divider valves FD91.2 and FD91.1. The main drive roller is located in the cloth drive unit whereas the other hydraulic motors are located in the plate pack. The motors HM9.3 to HM9.8 each drive one roller. These rollers are called auxiliary drive rollers and they are evenly located in the plate pack. As the flow is evenly distributed between HM9.6, HM9.2, HM9.1 and HM9.5, all motors are synchronized because the rest of the motors are series-connected to them. Synchronized hydraulic motors are required when a new filter cloth is fed to the filter or when there is low friction between the filter cloth and the roller because of the process. Bypassing the synchronization is also required, because the peripherical velocity of the rollers is not equal due to slight differences in the diameters of the rollers. If friction between the filter cloth and the roller is good, use of synchronization of the hydraulic motors will result in unnecessarily high friction loads. Directional valves HV91.1 and HV91.2 are included in the system to bypass the synchronization.

3.1.2 Sealing cylinders

Directional valve HV7 is connected with hydraulic pipes to sealing cylinder block (SC block). Sealing cylinder block divides the flow for eight sealing cylinders that are responsible for pressing the plate pack tightly closed during the filtration process. A scheme of the sealing cylinders is presented in figure 9.

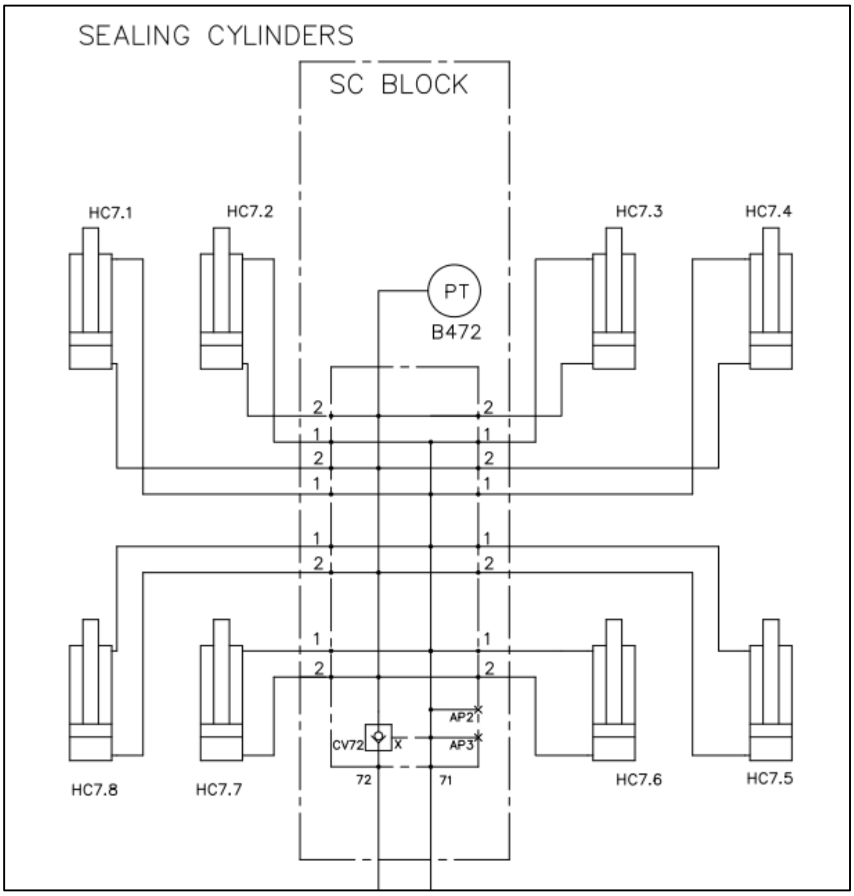


Figure 9. Scheme of the sealing cylinders.

The sealing cylinders presented in figure 9 are double-acting single-rod cylinders specially designed for PF60 filter. The stroke of the cylinders is 100 mm. As it is important to retain the specified sealing pressure, a pressure transmitter is included to the system. The sealing cylinders are located under the plate pack. The sealing cylinders must retain the pressure during filtration because pressure inside the plate pack can be as much as 16 bar. Pressure in the sealing cylinders can rise up to 300 bar during the process. The pressure inside plate pack is caused by water or air depending on the filter type. When filtration is done, sealing cylinders are positioned to the end position (lowest position).

3.1.3 Filter cloth tensioning

Directional valve HV8 in DV block is used to control the direction of filter cloth tensioning motor. A scheme of the cloth tensioning motor is presented in figure 10.

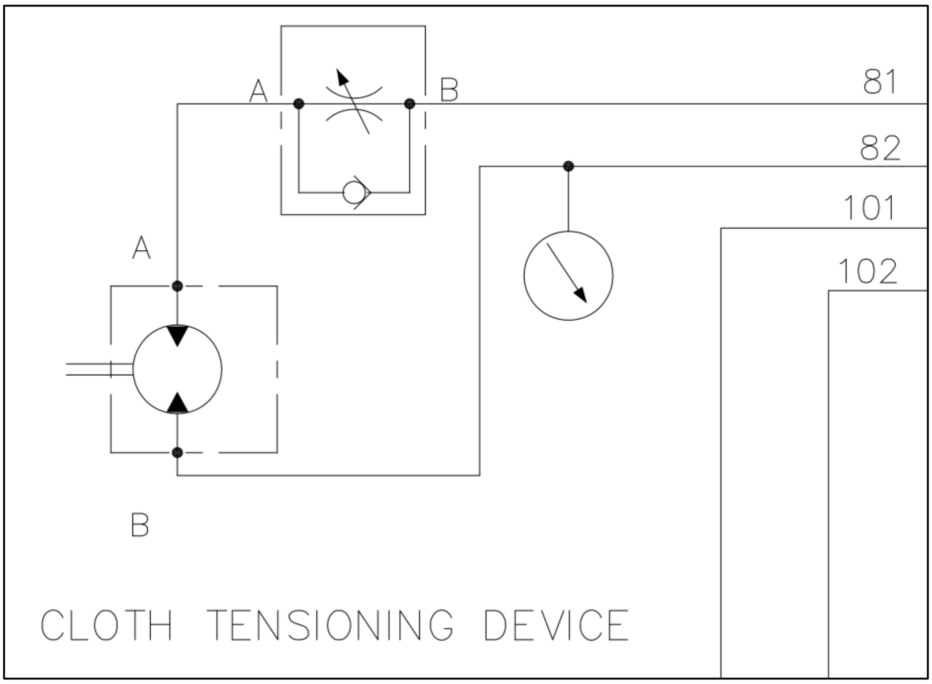


Figure 10. A scheme of the cloth tensioning motor.

Cloth tensioning motor is presented in figure 10. As the name suggests, cloth tensioning motor's function is to keep the filter cloth at constant tension during plate pack movements. Cloth tensioning motor is attached to the cloth tensioning device that has roller chain drive system to adjust the tension on cloth. A roller for the filter cloth is attached to the chain system and by controlling the vertical position of the roller, the tension on the cloth is adjusted. There are throttle check valves included in HV8 of the DV block as well as one throttle check valve near the hydraulic motor. This is because the initial flow adjustment is done from the DV block and those flows should not be altered later. The throttle check valve located near the motor is for maintenance purposes, when there is a need to hold the cloth tensioning roller on its upper locations. The flow can be throttled to zero and no adjustments are required for the DV block's throttle check valves.

3.1.4 Locking pins and cloth tracking

The last valve block in PF60 filter is locking cylinder block (LC block). LC block is connected to pressure line P2. A scheme of the LC block is presented in figure 11.

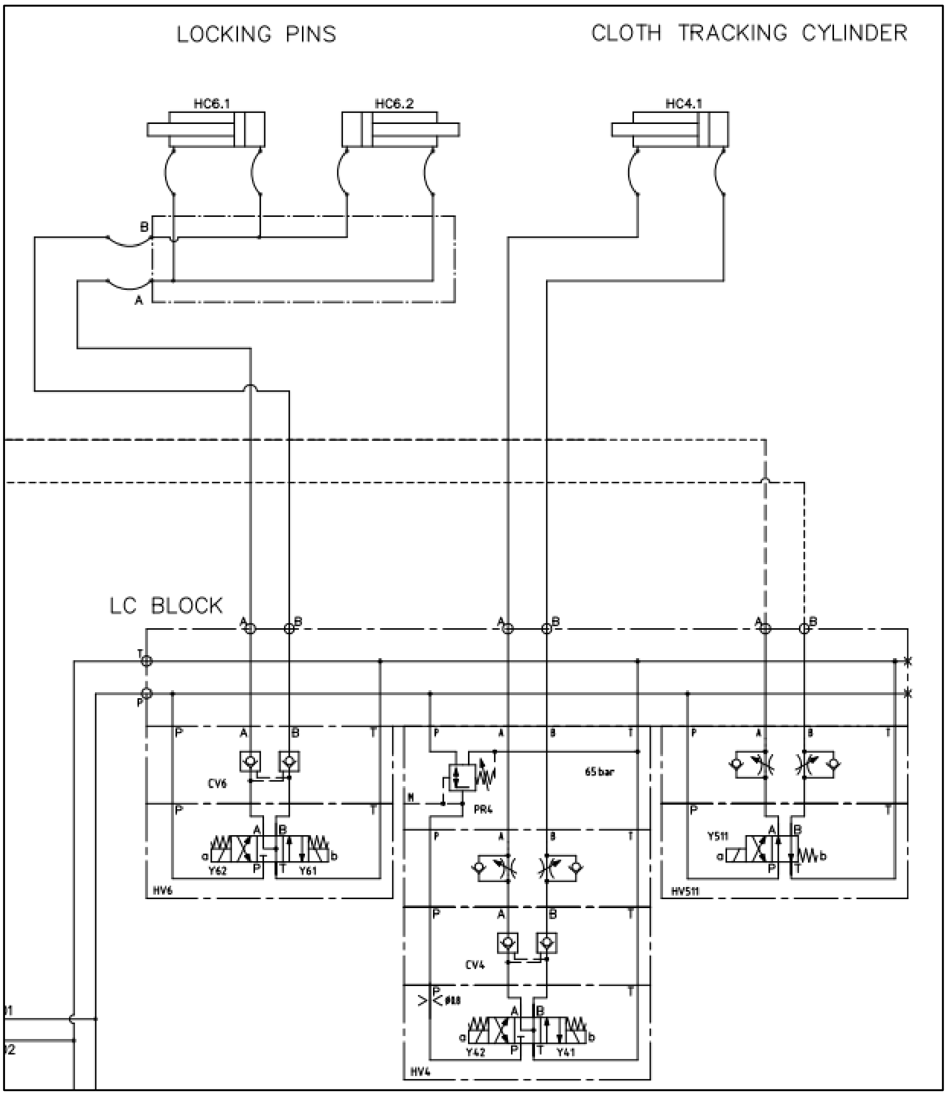


Figure 11. Locking cylinder block (LC block).

LC block is presented in figure 11. LC block is used to control two locking cylinders that are double-acting single-rod cylinders. Locking cylinders' function is to secure the top pressing plate of the plate pack on place before filtration and during maintenance operations. Both of the cylinders are connected to locking pins that are the actual securing element of the plate pack. The four columns of the filter have holes for the locking pins. Directional valve HV6 controls the direction of locking pins. The upper pressing plate has a transmitter to monitor its level to find the correct position of holes for the locking pins. Despite the name LC block, this valve block has other functions as well. The second function is to control the direction of cloth tracking cylinder with a directional valve HV4. Cloth tracking cylinder is also a double-acting single-rod cylinder and its function is to keep the filter cloth centered on the rollers. The cylinder is attached to end of a centering roller to alter the angle of the roller and move the filter cloth to the required direction. Cloth tracking is required when the cloth is driven. Position of the cloth edge is detected with analog position transmitter and protective mechanical limit switches stop the cloth movement in case the cloth is incidentally driven out from the rollers. A third directional valve in LC block is HV511 for quick action cylinders and that topic is covered later in this thesis.

3.1.5 Process valves

Process valves are connected to the line P2 as mentioned before, because they must be able to close during a power failure. Therefore, an accumulator is included to the line P2 in the hydraulic unit. A scheme of the process valves is presented in figure 12.

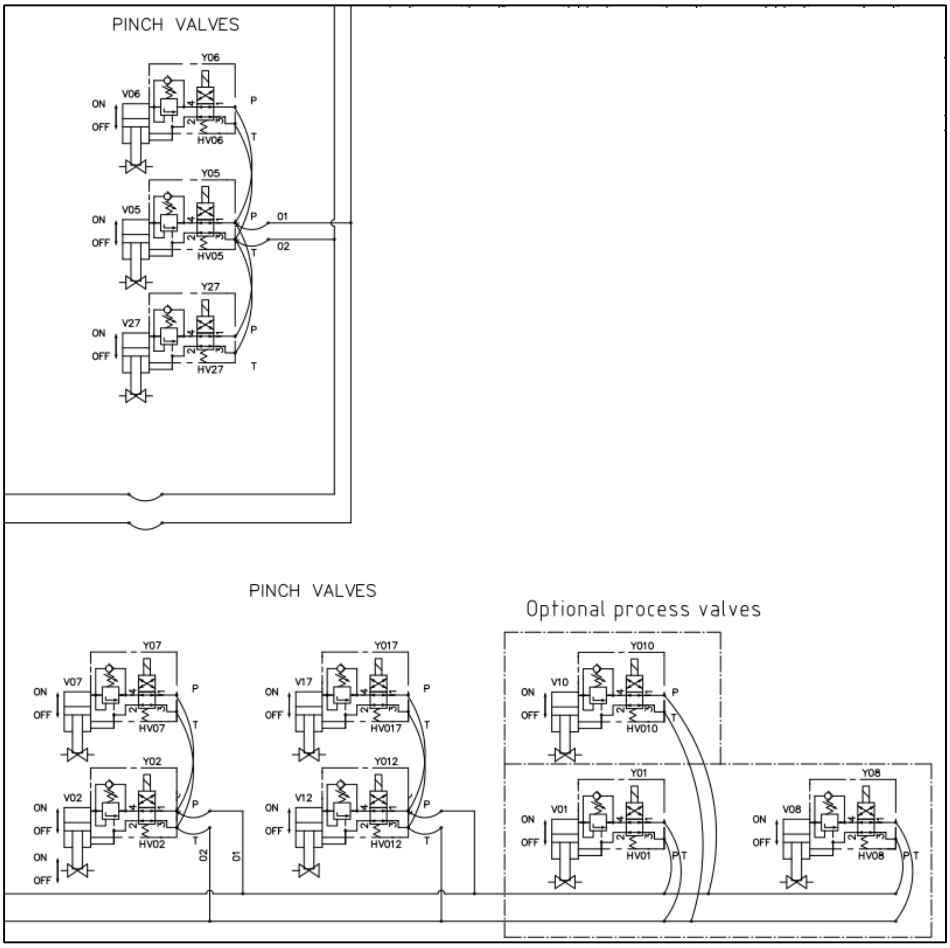


Figure 12. Hydraulically actuated process valves of PF60.

The process valves and their actuators are presented in figure 12. All hydraulically actuated process valves are pinch valves with hydraulic cylinders. The direction of cylinders is controlled with directional valves included to each pinch valve. In addition to directional valve, each pinch valve includes a counterbalance valve to retain the valve in closed position.

3.1.6 Hydraulic circuit of opening and closing the plate pack

Four double-acting double-rod hydraulic cylinders (also called quick action cylinders of PF60) are responsible of opening and closing the plate pack of the filter. The flow is distributed to quick action cylinder circuit (QAC circuit) from line P1 and through directional valve HV5 in the DV block. HV5 is connected to QAC block that distributes the flow to four cylinders. QAC circuit is presented in figure 13.

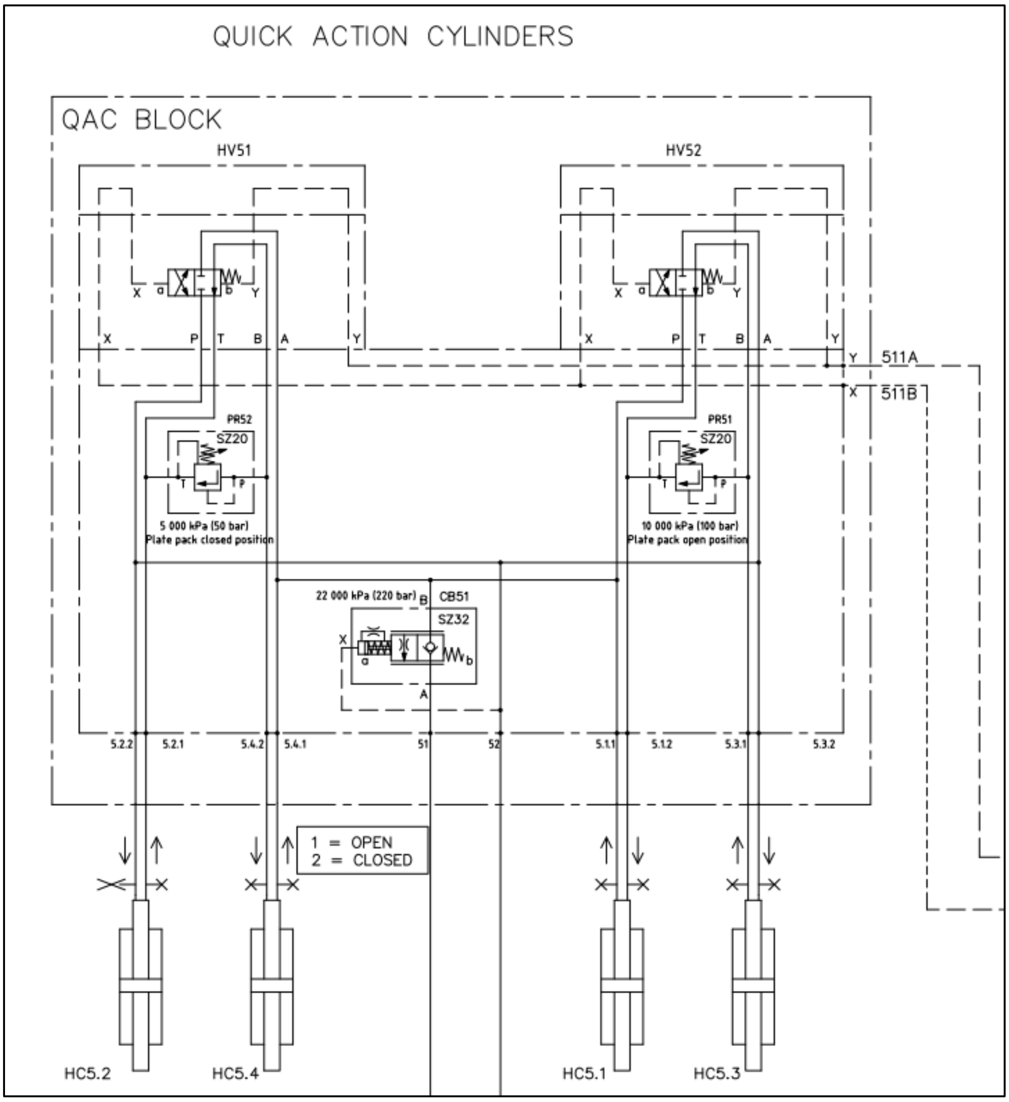


Figure 13. QAC circuit.

QAC circuit presented in figure 13 includes directional valve HV5 in DV block, QAC valve block, pipelines, valve HV511 in LC block and four hydraulic cylinders. Principle of opening and closing the plate pack is that the plate pack should be opened and closed fast to maintain a good filtration cycle time. The objective is to use a speed of 80 mm per second. (Kaipainen 2018.)

It is important to keep the plate pack straight. Therefore, all four cylinders must be synchronized. The cylinders of opening and closing the plate pack are called quick action cylinders (QACs) because fast movement of the plate pack is done by taking advantage of two connecting possibilities of the cylinders. During fast plate pack movement (quick action) two of the cylinders are in series connection and the series connected cylinders are parallel connected. This is possible because the cylinders are double-rodded and therefore the volumes are equal on both sides of the cylinder. This connection enables that volume flow is required for two cylinders and the required speed is reached. The cylinders that are in series connection are located on the diagonal line of the upper pressure plate in order to move the plate pack as steadily as possible. However, synchronization in this connection is not perfect and there can be differences in the position between the four cylinders. Before reaching the end positions all differences between the cylinders must be compensated. To do this, all cylinders are switched to parallel connection. As the cylinders are connected to line P1, the control pressure for switching to parallel connection must come from different pressure line to maintain the required pressure level. Parallel connection of cylinders in PF60 filter is called the balance mode. When balance mode is on the speed of the cylinders is reduced because the total volume flow is divided to four cylinders instead of two cylinders.

3.2 Challenges of the current hydraulic system

The current hydraulic system of PF60 filter is generally functioning well, but there are some challenges and development possibilities. The new hydraulic system should have an upgrade to the challenges for creating value. A challenge concerning the QAC circuit (opening and closing the plate pack) is the synchronization of the cylinders. The synchronization is prone to failures, for example sticking of a valve stem. If the cylinders are not properly synchronized during plate pack movement, wide-ranging damages can occur in the machine. In addition to machine damages, danger can be caused to people. Unsynchronized cylinders incline the plate pack and that can cause damages to cylinders, hydraulics and plate pack as well as other structures in the filter. Hands-on experience has shown that particularly challenging components are the directional valves that are used to select the desired drive mode (fast movement or balancing mode). If a valve stem is sticking in one directional valve, volume flow can end up in the wrong direction and cause asynchronization of the cylinders. QAC circuit does not have position transducers for the cylinders so the actual position of the cylinders is not known during opening or closing the plate pack. There is an encoder in the center of the upper pressure plate that is based on draw wire measuring principle. The problem is that as there is only one encoder that is located in the center of the plate pack, the system does not detect the inclination of the plate pack. (Kaipainen 2018.)

In a general level, there are other challenges in the hydraulic system. The current system has a lot of hydraulic pipelines, hoses and valves that cause losses in the system and reduce the energy efficiency. Another challenge is concerning the safety aspect of hydraulic pipelines and especially hydraulic hoses. The current system has various hydraulic hoses for moving actuators. The hoses can be a safety hazard and they often need protective equipment to protect against bursts, such as sleeves. As the system has a large centralized hydraulic unit, it requires an own space somewhere close to the pressure filter. This space requirement can be challenging if the plant has limited space available for pressure filter and its equipment. The centralized system can be difficult during troubleshooting. It is difficult to locate a faulty component when there are many possible locations for the fault. Availability of components is another challenge, because the system includes special components such as the variable-displacement pump. Delivery times for these components can be as high as six months. Because of the number of components and pipelines, assembling the hydraulics takes a lot of time during manufacturing of PF60 filter. This leads to high costs and increased delivery times. (Kaipainen 2018.)

The volume of hydraulic fluid in the tank in the current hydraulic system is 800 liters. This volume does not include the oil volume in pipelines and actuators. General advice for customers is to change the oil in the tank once a year but it is recommended to analyze oil quality to find the optimum changing cycle. The oil in the pipelines is not changed during a planned oil change, only the oil in the tank of hydraulic unit is replaced. Therefore, at least 24 000 liters of hydraulic oil is consumed to planned oil changes during the lifetime of the pressure filter (estimated 30 years). The oil should be recycled properly but pressure filter's manufacturer cannot be sure what the pressure filter's user does to the used oil. (Kaipainen 2018.) The environmental properties of mineral oil based hydraulic fluids are poor not only because of the mineral oil but because of the additives. A vegetable oil based hydraulic fluid would be a better option in terms of biodegradability. (Ekman & Börjesson 2011, p. 297 – 304.)

3.3 Decentralized hydraulic concept for PF60

The aim is to create a new concept that utilizes speed-controlled hydraulic pumps for the hydraulic system of PF60 filter. As a part of this thesis a new circuit for opening and closing the plate pack is developed to overcome the challenges described earlier. Therefore, opening and closing the plate pack is covered in detail and the rest of the hydraulic system on a concept level. The hydraulic operations of PF60 filter are divided across the machine. Figure 14 presents the locations of the current hydraulic operations.

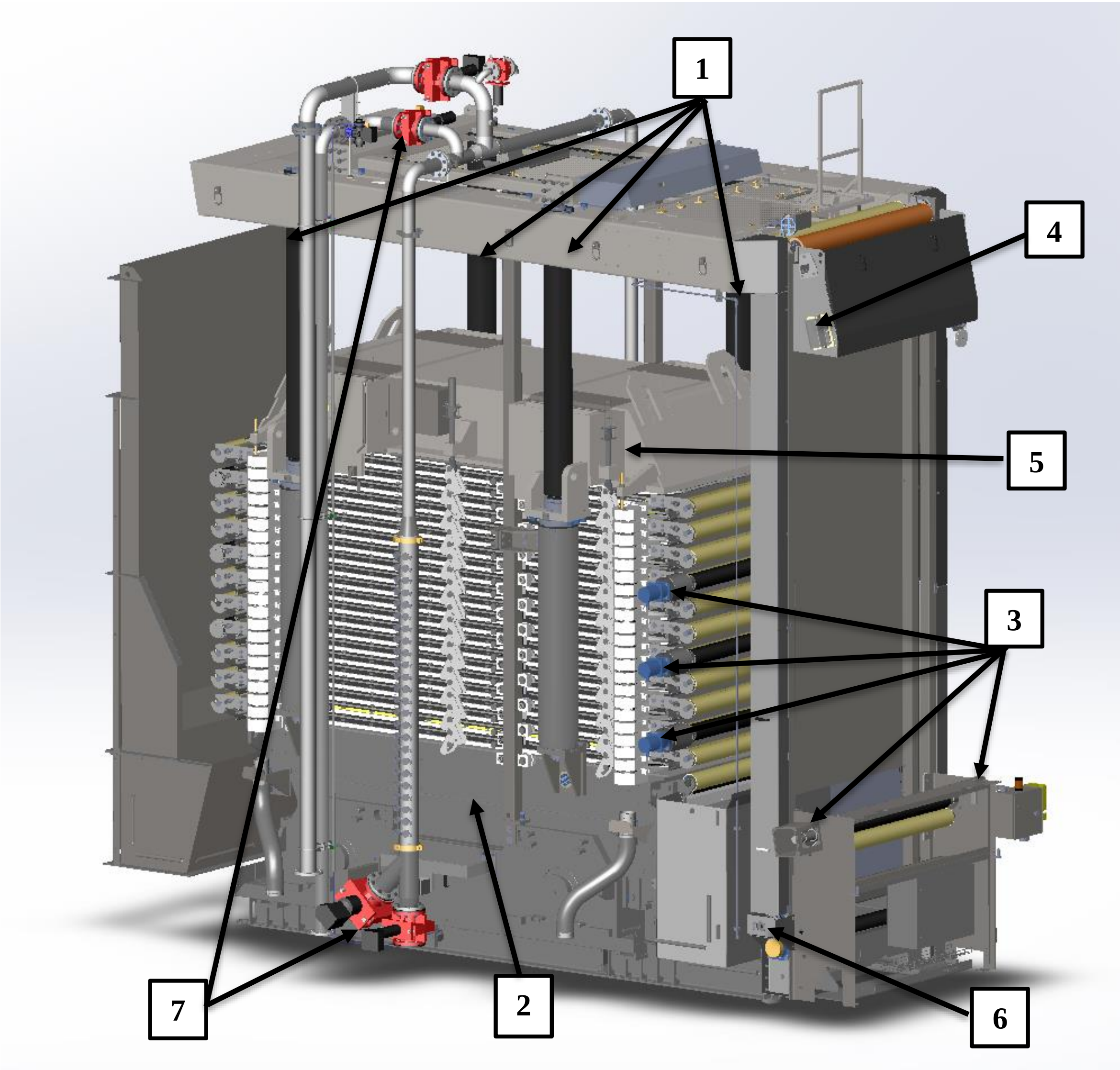


Figure 14. Locations of the current hydraulic operations.

Locations of the current hydraulic operations are presented in figure 14. Number 1 in the figure is opening and closing the plate pack (QAC circuit). Sealing the plate pack for filtration is number 2 in the figure. The sealing cylinders are located under the plate pack. Filter cloth drive has hydraulic motors in cloth drive unit and in the plate pack (number 3 in the figure). Number 4 in the figure presents centering of the filter cloth (cloth tracking). The plate pack is secured in closed-position before filtration with two cylinders that are located in the top pressure plate (number 5 in the figure). Number 6 presents the filter cloth tensioning motor. Finally, number 7 presents the hydraulic-operated process valves of PF60 filter.

As presented earlier, the hydraulic operations are physically divided to different locations across the PF60 filter which creates a good basis for the decentralized system. As a baseline for the new hydraulic concept, opening and closing movement of the plate pack is separated from rest of the system. The rest of the currently hydraulic operations can be hydraulic, electric, electro-hydraulic, electro-mechanical or something else in the future system. From a safety point of view, hydraulic hoses can cause danger if they are damaged. Therefore, usage of hydraulic hoses should be avoided in the new hydraulic concept. One part where usage of hydraulic hoses cannot be avoided is the process valves of the filter. The process valves are manufactured by an external manufacturer that uses floating hydraulic cylinders where both piston rod and cylinder barrel move to operate the valves. That is why the process valves will have some hydraulic hoses in the future as well.

Currently, there are hydraulic hoses in filter cloth drive, securing the plate pack in position (locking pins) and centering the filter cloth. Most of the cloth drives and all locking pins are located in the moving part of the filter, the plate pack. That is why there should not be hydraulic operations in the plate pack area in the new concept or power-by-wire concept should be used. Driving of the filter cloth requires a certain amount of auxiliary drive motors that are located across the rollers of the filter cloth. These motors could be servomotors or gearmotors, for instance, to avoid the usage of hydraulic hoses. Same considers one hydraulic motor that is used for filter cloth tensioning.

The locking pins are located in the moving part of the PF60 filter, the top pressure plate. At the moment, there are two floating hydraulic cylinders that are both connected to their own pair of locking pins. Another function where a hydraulic cylinder and hydraulic hoses are used is centering of the filter cloth. Both of these currently hydraulic operations could be replaced with electromechanical or electro-hydraulic actuators.

The sealing cylinders that are located under the plate pack are of special design for PF60. The pressure inside the cylinders can rise up to 300 bar during the process, but that pressure is not caused by the mass of the plate pack, it is caused by the process pressure inside the plate pack. The pressure during sealing is monitored to keep the plate pack tightly closed even when the seals between the filter plates are slightly worn. Hydraulic operation of plate pack sealing has proven to be functional and sealing cylinders could have their own motor-pump unit in the future system.

The process valves have integrated directional valves and hydraulic cylinders, as described earlier. The functioning of process valves must be assured during a power failure, so a hydraulic accumulator must be included in the system. The process valves could have their own motor-pump unit in the new concept. The downside is that hydraulic pipelines and hoses are still required in the new system due to the locations of the process valves. Some of the valves are located on top of the filter and some near the floor level.

Suggestion for concept of the currently hydraulic operations is the following:

- Separate motor-pump units for the hydraulic cylinders that open and close the plate pack
- separate motor-pump unit for sealing the plate pack
- separate motor-pump unit for the process valves, including hydraulic accumulator
- electric motors for filter cloth drive and tensioning
- electro-mechanical or electro-hydraulic actuators for filter cloth centering and locking pins.

The concept is further developed regarding opening and closing of the plate pack. The results from the tests are useful when the rest of the system is further developed in the future.

4 NEW SYSTEM FOR OPENING AND CLOSING THE PLATE PACK

The new design for opening and closing the plate pack is presented in this chapter. The basis for the new system is that the same hydraulic cylinders for plate pack are used. As the new circuit is tested with a current standard model filter, minimum modifications can be done to the pressure filter's mechanical structure. The aim is to gain experience from the new technology and its benefits. When the new technology is implemented to the standard pressure filter model range, more modifications can be done to the hydraulic system and mechanical structure of the PF type filters. The intention is that the new system is for test purposes only and the new system will be further developed in the future.

4.1 Requirements for the new circuit

The new circuit for opening and closing the plate pack should solve the challenges the current circuit and system has. Therefore, the challenges set the requirements among other properties of the current QAC circuit. The new circuit for opening and closing the plate pack should reduce energy losses compared to the current system. In addition, synchronization of the cylinders should be improved. Of course the new circuit should be able to perform all the same tasks as the current system. (Kaipainen 2018.)

One requirement concerning the energy consumption is reducing the amount of hydraulic pipelines. As the basic mechanical construction of the PF60 is kept the same, hydraulic power unit or units must be brought closer to the hydraulic cylinders. The new system should be simple in terms of installation, because the new system is tested on a current standard model PF60 filter. Only minor modifications can be done to the filter. After the practical tests the current hydraulic system must be reclaimed to the filter. This is because the filter is a production model filter that is to be delivered to a customer. (Kaipainen 2018.)

4.2 Concept for the circuit of opening and closing the plate pack

The aim is to create the required volume flow as close to the actuator (cylinder) as possible. Currently both ports of the hydraulic cylinders are located on top of the filter. As the length of hydraulic pipelines or hoses must be kept in a minimum, the hydraulic unit or units must be located on top of the pressure filter.

The basis of the new test system is that rotating speed-controlled pumps are used. The volume flow can be adjusted by controlling the rotating speed of the pump. The current system has a large variable-displacement pump that has a long delivery time. By using a standard fixed-displacement pump, lower costs and shorter delivery times can be achieved.

There are several manufacturers for rotating speed-controlled hydraulic systems. Hydac Finland had previously presented their variable speed-controlled systems to Outotec and Hydac was interested to participate in the development of hydraulics of PF type filters. That is why Hydac was chosen to be a supplier of the test system. At first it was decided that the test system will have individual motor-pump units for each cylinder. The decision is based on literature study and the requirement for synchronization of the cylinders. In this study, the synchronization of the cylinders is of high importance and a good way to avoid the challenges of the current system is to have individual pumps for each cylinder.

When each cylinder has a similar motor-pump unit, the rotating speed of each motor must be equal to maintain the synchronization of the four cylinders. However, this only applies to a system when all components are in good condition. For example, wear of the cylinder rod seals can cause leakages and a higher volume flow is required for this cylinder to maintain the speed. The current system has no sensor to monitor the inclination of the plate pack. With an inclinometer it is possible to detect a cylinder that is not reaching the level of synchronization. The data from inclinometer could also be used for condition monitoring of the hydraulic cylinders. The best place for an inclinometer is on top of the upper pressure plate that is the topmost component of the plate pack. Feedback control from the inclinometer is not required for the test system, because the tests are done with a new PF filter that has fully functioning cylinders. For a production model hydraulic system, feedback control from the inclinometer is mandatory to maintain the synchronization in all circumstances.

The system is constructed so that four similar motor-pump units are mounted on top of the same oil tank. Each motor has a frequency converter that controls the rotating speed. The motor-pump units could each have an own oil tank but expenses would increase with that decision. For test purposes the expenses should be kept at a moderate level. This is why all components are as standard components as possible, including the oil tank. Hydac will choose the components according to the requirements from their portfolio. The direction of flow will be controlled with directional valves to move the plate pack in both directions. As the total mass of the plate pack is up to 84 000 kg, it must be ensured that the movement downwards is in control. This is chosen to be done with a load-holding counterbalance valve. As each cylinder has own motor-pump units, each cylinder must also have a counterbalance valve. The test system will not require heating or cooling systems as the tests are done in normal environmental conditions inside a production assembly hall.

It is decided that Hydac will manufacture the hydraulic test unit. Outotec's responsibility will be to provide electrical power supply for the unit. In addition, designing the hydraulic connections between the cylinders and test unit are also on Outotec's responsibility. Outotec will also provide and design other equipment required for the tests, such as additional support structures. Outotec will arrange enough time for the tests and will have PF60 filter available for that time. The author of this thesis will be responsible of Outotec's tasks in this project, but automation department will do the designing of current supply and selecting the required electrical components. In addition, tasks for the author of the thesis include planning of the tests and the required preparations before the tests can take place, for example modifications required to the hydraulic pipelines of the current system. Hydac will assist with the testing and provide a data recorder and the required transmitters during the tests.

4.3 Calculating the initial values

Initial values for the circuit of opening and closing the plate pack are calculated based on the requirements of the system. The components for the test system are dimensioned according to the calculations. Required volume flow for a hydraulic cylinder can be calculated from equation 1.

$$q = \frac{A \cdot v}{\eta_v} \qquad (1)$$

, where q is volume flow, A is area of the piston, v is velocity of piston movement and η_v is volumetric efficiency (Kauranne et al. 2013, p. 200).

A pressure that is required to move the cylinder under load can be defined with equation 2.

$$p = \frac{F}{A_{in} \cdot \eta_{hm}} + p_{out} \cdot \frac{A_{out}}{A_{in}} \qquad (2)$$

, where F is the load, A_{in} is area of the piston on the working side, η_{hm} is hydromechanical efficiency, p_{out} is pressure on the chamber opposite to working chamber and A_{out} is area of the piston opposite to working chamber (Kauranne et al. 2013, p. 202).

The total hydraulic power required can be calculated from equation 3.

$$P = \frac{F \cdot v}{\eta_t} \qquad (3)$$

, where η_t is total efficiency of the hydraulic cylinder (Kauranne et al. 2013, p. 203).

The equations 1, 2 and 3 include cylinder losses that are taken into consideration with volumetric efficiency, hydromechanical efficiency and total efficiency. Volumetric efficiency $\eta_v \approx 1$ if cylinder has seals that are in contact with barrel or piston rod. Hydromechanical efficiency varies between 0,80 to 0,96 and is dependent on piston seal types, pressures and surface qualities. The efficiency improves when the pressure increases as frictional forces are significantly lower than cylinder forces. The total efficiency of hydraulic cylinders is mostly defined by hydromechanical efficiency, when volumetric efficiency $\eta_v \approx 1$. (Kauranne et al. 2013, p. 200 – 203.)

When the plate pack of PF60 is opened from closed position, the load on the cylinders increases gradually. At first, the only load is the top pressing plate. Connecting links are used to connect the filter plates to each other and to the top pressing plate. The maximum load is present when the plate pack is fully opened. Maximum load includes the plate pack, top pressing plate and the filtered cake. For the largest PF60 type pressure filter, the load on the cylinders can be up to 84 000 kg. The load varies between filter sizes because of filtration area and density of the slurry. The maximum load without slurry is 56 000 kg. The load is uniformly distributed between the four cylinders.

The current hydraulic cylinders of the plate pack are double-acting double-rod cylinders. Diameter of the piston is 250 mm and diameter of the piston rod is 140 mm. The maximum stroke length of the cylinder is 1430 mm for the largest filtration area. Movements of the plate pack must be fast enough, so the desired velocity of the cylinder stroke is 80 mm per second. The calculations for initial values for the circuit of opening and closing the plate pack are presented in appendix I. Based on the calculations, the required volume flow for each cylinder is 163,4 liters per minute. On the other hand, the maximum pressure is less than 68 bar. The current cylinders are originally dimensioned for different operating principle as described earlier, so the values for this study are less optimal. The total plate pack opening and closing time is around 18 seconds. Maximum hydraulic power is 18,4 kW. The test system will not be assembled on a largest PF60 filter and that is why the load is not as high as with a full-sized filter. The calculations for initial values with the actual test filter are presented in appendix II. The tests are conducted in a production assembly hall and because of that the filter will not have the actual filtration process ongoing. The load is caused for the cylinders from the mass of the plate pack, which is 33 000 kg in the tests.

4.4 Test system for opening and closing the plate pack

Based on the initial values, Hydac manufactures a hydraulic test unit. Hydraulic diagram of the test unit is presented in appendix III. The unit has four 22 kW motor-pump units mounted on a single oil tank. The rotating speed of each motor is controlled with a VFD. The units have identical valve blocks for controlling the direction of movement and load-holding. Hydraulic diagram of one motor-pump unit is presented in figure 15.

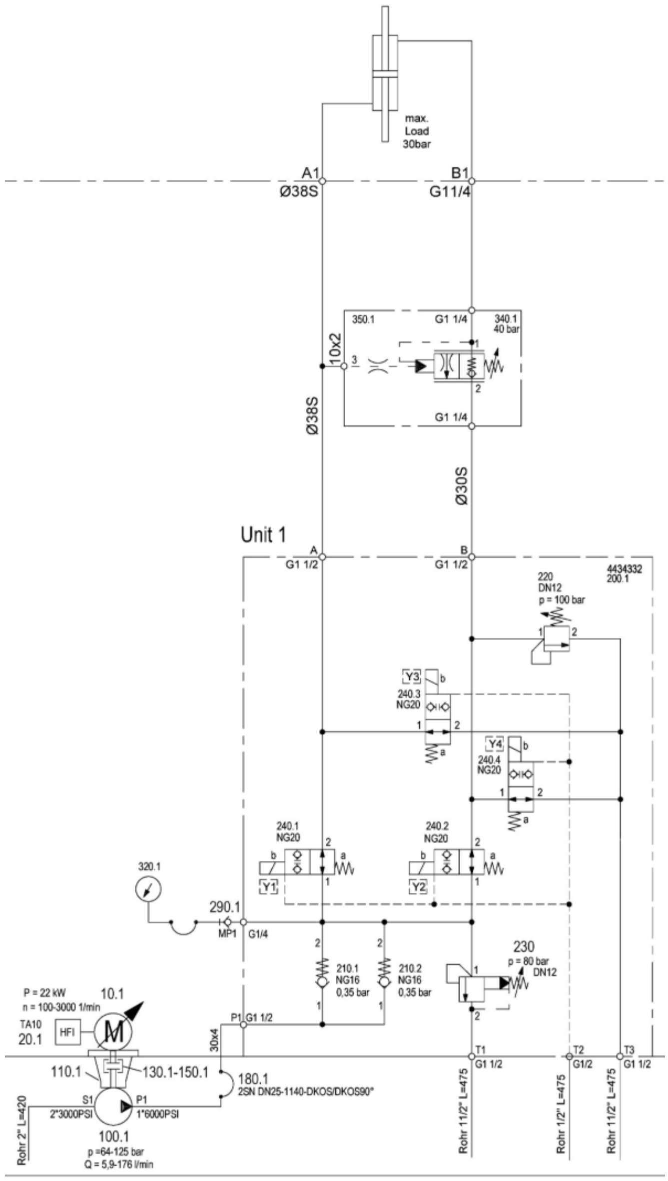


Figure 15. Hydraulic diagram of one motor-pump unit (Hydac 2019a).

The hydraulic test unit consists of four identical motor-pump units (hydraulic diagram in figure 15). Opening of the plate pack occurs when the direction of flow is from port B1 to port A1. There are two poppet type check valves directly after the pump. The direction of flow is controlled with pilot operated 2/2 solenoid directional valves. There are four directional valves in each motor-pump unit. During opening of the plate pack, the pressure relief valve sets the maximum pressure at 80 bar. When the direction of flow is changed from port A1 to port B1, the plate pack closes. To control the closing movement, a counterbalance valve is included in the system. The counterbalance valve has pilot ratio of 3:1. If the plate pack is left at open position or stopped at any position between open and closed, the counterbalance valve will retain the position of the plate pack. The plate pack will move downwards only if the pressure increases to the set level on the side of port A1.

The motors are 22 kW squirrel cage motors. Each motor has a frequency inverter to control the rotating speed. Rotating speed of the motor determines the volume flow that the pump produces. The pumps are internal gear pumps. The maximum volume flow of the pump in this system is 176 liters per minute and the maximum pressure is 125 bar. Figure 16 illustrates the pressures and volume flows of the internal gear pump.

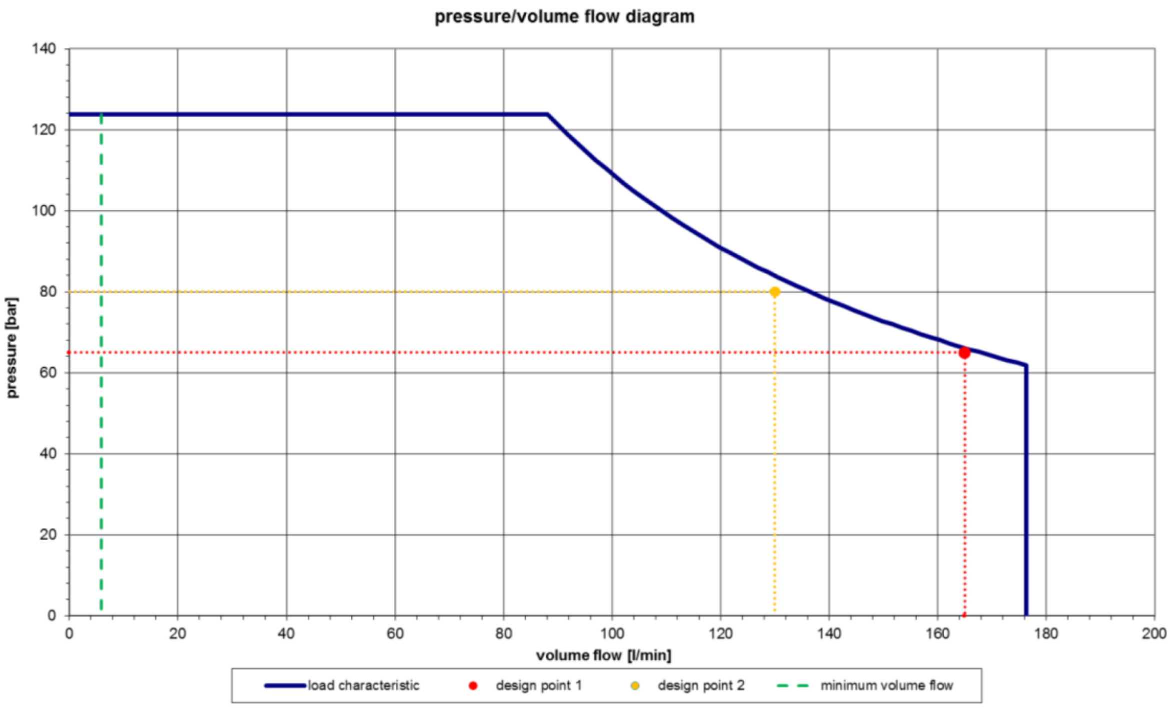


Figure 16. Pressure and volume flow for the test unit (Hydac 2019b).

Figure 16 presents the accessible pressures and volume flows of each motor-pump unit. Field weakening above the rated speed (1470 rpm) of the motor causes the reduction of pressure above volume flow of 88 liters per minute. It is possible to reach a pressure of 64 bar with the maximum volume flow (176 l/min).

The test unit has a software that enables the motors to rotate with equal speed. The unit has a Profinet BUS communication interface. The test unit is not controlled from the PLC of the PF60 filter because modifications cannot be done to the software of a production model filter. The hydraulic test unit has simple push-buttons and a potentiometer on the cover of its electrical cabinet. These can be used to control the movement of the plate pack during the tests. However, the hydraulic test unit will be located on top of the PF60 filter during the tests and there is no easy access to the electrical cabinet. That is why a remote control cabinet is required to operate the hydraulic unit from ground level. Figure 17 illustrates the remote control cabinet of the hydraulic test unit.

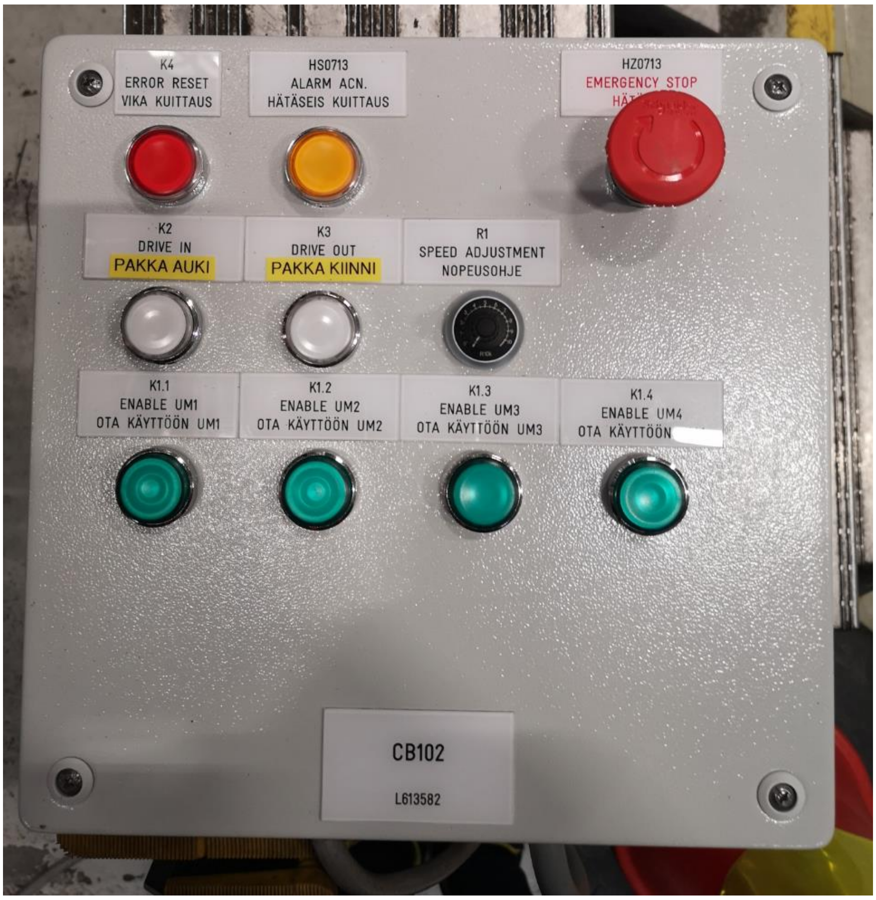


Figure 17. Remote control cabinet for the tests.

The remote control cabinet presented in figure 17 has push-buttons for alarms, errors and direction of flow. There is also a possibility to choose which motor-pump units are active. This feature is mandatory if the plate pack will get inclined and there is a need to drive only some of the motor-pump units. Rotating speeds of the motors are controlled with a potentiometer.

The hydraulic test unit is rather large as the motor-pump units are mounted on a 630 liter oil tank. Mass of the unit is approximately 2400 kg with oil. Dimensions of the unit are approximately 2340 mm x 1350 mm x 1780 mm. A photograph of the test unit is presented in figure 18.

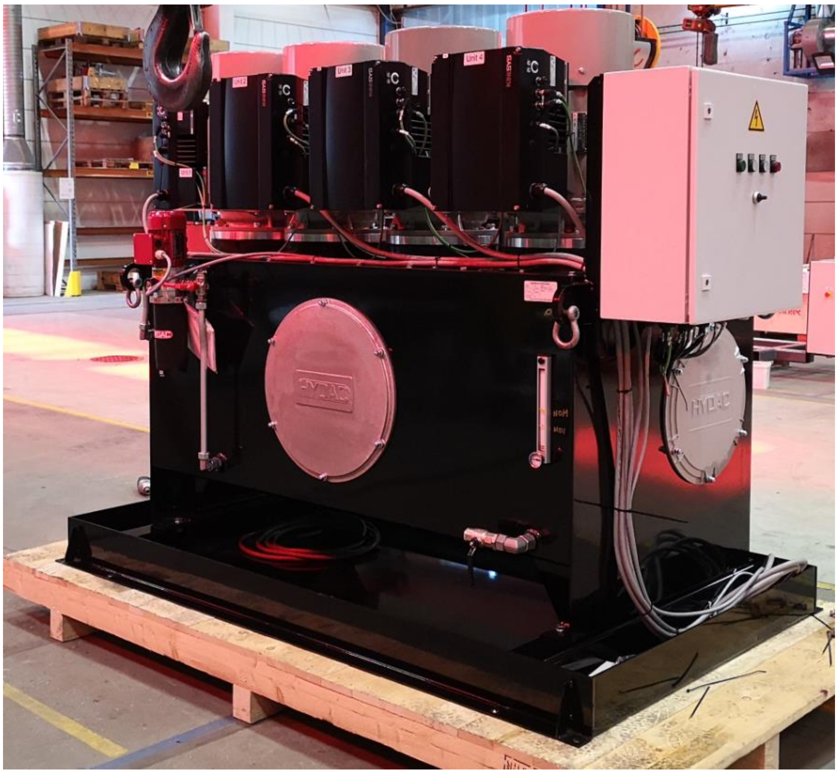


Figure 18. Hydraulic test unit for PF60.

In addition to the test unit (presented in figure 18), other components are required for the tests. These components include a support structure, electrical power supply cabinet, remote control unit, hydraulic oil, hydraulic fittings and hydraulic hoses. In addition to these parts, there are many smaller parts such as screws and nuts. All of the parts are listed in appendices IV and V. A support structure is required to install the hydraulic test unit as close to the hydraulic connection ports on cylinders as possible. As the mass of the test unit is approximately 2400 kg with oil, it requires a separate support on top of the PF60 filter. The topmost part of the filter (the upper body) is a sheet metal structure that is not designed to withstand as heavy loads as the hydraulic test unit. On the other hand, the upper body is installed on top of the columns of the filter. The columns are the load-carrying structures of the filter and they can withstand the extra load caused by the hydraulic test unit. Therefore, the only option is to support the test unit on the columns. The test unit should be supported on two columns because of the physical location of the columns.

As the hydraulic test unit is only for testing purposes, the support structure should be as simple as possible. It must be easy and fast to install on top of the filter, because of the limited time available for the tests. Due to the geometry of the filter's columns, there are four possible fixing points for the support structure. Maintenance platforms are located on top of the filter and the possible fixing points for the support structure. That is why the maintenance platform must be disassembled before the tests can take place. In addition, the surface of the upper body has parts that are higher compared to the planar part. Therefore, the support structure must have some sort of platform under it. The height of the support structure should be kept at minimum because of the limited height of the assembly hall. When the hydraulic test unit is placed on top of the PF60 filter, the total height of the assembly is around 7300 mm.

As described before, the hydraulic test unit requires an electrical power supply cabinet. The most convenient location for power supply cabinet is close to the test unit, because the amount of cabling work should be kept at minimum. The size of the power supply cabinet is 760 mm x 760 mm x 300 mm and it must be properly supported. It is best that the amount of manufactured parts is as low as possible, considering that the test arrangements are temporary. The support structure of the test unit is a manufactured assembly that can include a support for the electrical power supply cabinet.

The support structure does not have specific corrosion resistance requirements as the tests are conducted in an assembly hall. Requirements for the material are good weldability, low cost and easy availability. The fastest way to manufacture the support structure is to use standard structural steel profiles. Due to the limited space available, I-profile was chosen for the platform parts. The platform parts are connected to the support structure with screws to allow the installation of platform parts on the columns before lifting the hydraulic test unit on top of the PF60 filter. The support structure could be made of I-profile as well, but welding an I-beam to the web of another I-beam requires machining and increases costs of the structure. Therefore, a rectangular square section was chosen as a profile. The ends of the beams can be left open as the structure is only for testing purposes. A support made of flat bars and angle bars is welded to the other end of the support structure. The support structure is illustrated in figure 19.

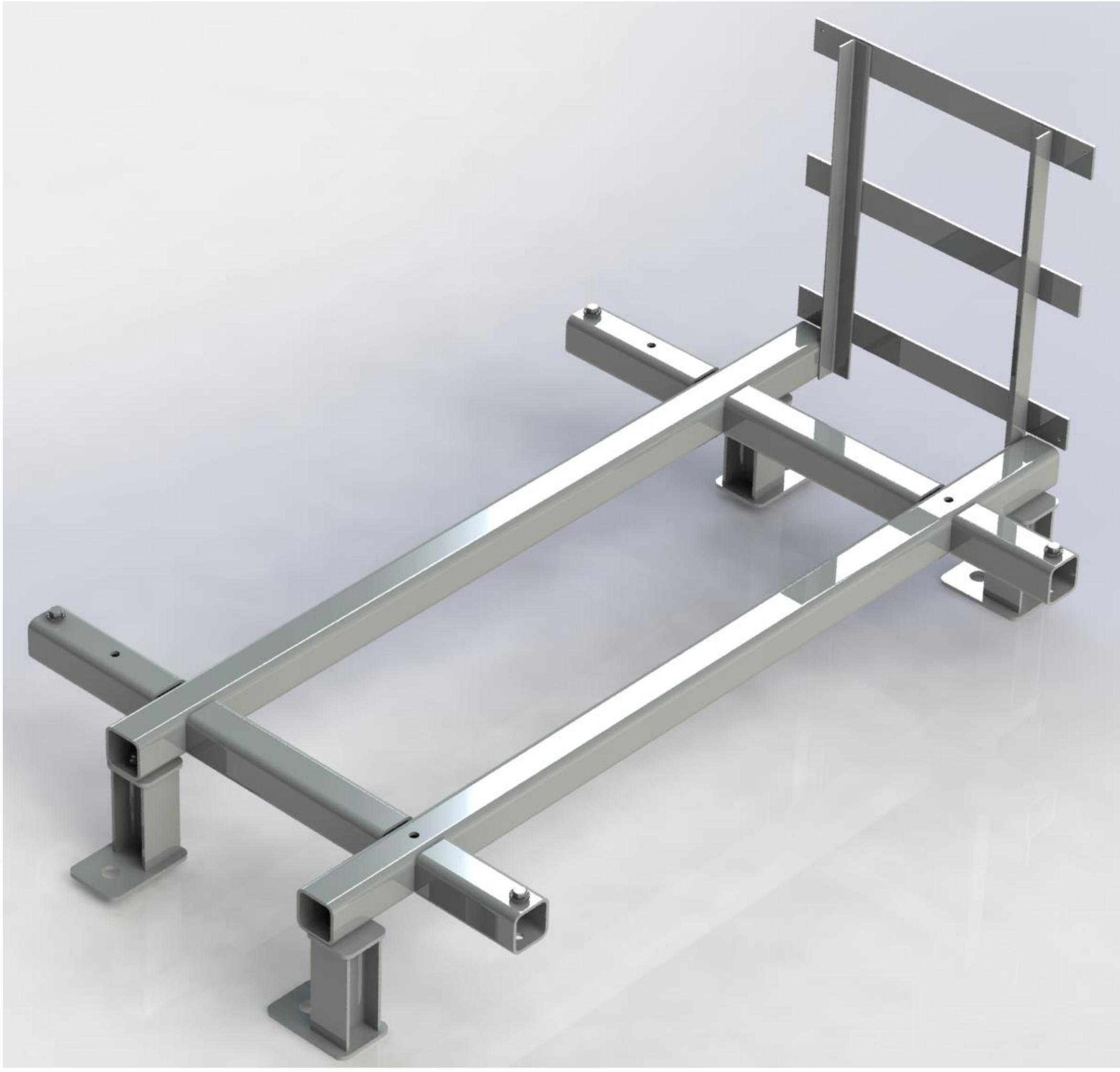


Figure 19. Support structure for the hydraulic unit.

The assembly drawing of support structure for the hydraulic test unit (presented in figure 19) is included in appendix IV. The assembly includes four welded platform parts, welded support structure, screws, nuts and washers. The support structure is made of structural steel S355J2G3 and mass of the structure is 190 kg.

Hydraulic connections for the tests are done with hydraulic hoses because of their bendability. Hydraulic connectors that are required for the tests are presented in appendix V among the other components for hydraulic connections. The motor-pump units are connected to hydraulic cylinders with 1 ¼ inch hydraulic hoses.

4.5 Measuring the test system

The tests are done with different connections, later in this thesis they are called test assemblies. In test assembly one, each cylinder is driven by own motor-pump unit. Connections between cylinders and hydraulic test unit are done with hydraulic hoses. The plate pack is opened and closed with various volume flows. The intended speed is 80 mm / s. Comparison is done between test assembly one and the current hydraulic system. The current hydraulic system is also driven in “balance mode” (parallel connected cylinders) to achieve comparative values. The same speeds of the plate pack are compared with both systems. The current hydraulic system is measured after the tests have been done with hydraulic test unit.

In test assembly two, two motor-pump units are connected to the ports 51 and 52 in QAC block of the current hydraulic system. Hydraulic pipelines between cylinders and QAC block are the same as in the current hydraulic system of PF60. Comparison is done between test assembly two and the current hydraulic system. Measurements to the current system are done after the tests with assemblies one and two are done. Various volume flows are used but comparison is done between the same speeds.

Measurement data is recorded with portable data recorder provided by Hydac. As consumption of electricity is part of the main research question of the study, active power is measured. An energy measurement device is used to measure active power. Pressure in both cylinder chambers of each cylinder is measured with pressure transmitters. The position and speed of the plate pack are measured with the own automation system of PF60 and data is transferred to the data recorder. Inclinometer is installed on top of the top pressure plate of PF60 to monitor the angle of the plate pack. The measurement plan and devices are covered in detail later.

4.5.1 Schedule for the tests

The tests are conducted in July 2019 in Lappeenranta. The tests are estimated to take about one week with the preparations. A schedule for the tests is presented in table 2.

Table 2. Schedule for conducting the tests.

Wednesday 3.7.	Hydraulic test unit arrives in Lappeenranta, preparations
Thursday 4.7.	Cabling work, filling the tank with oil, lifting the test unit
Friday 5.7.	Installing Hydac HMG and transmitters
Monday 8.7.	Tests with test assembly 1, changing connections to assembly 2
Tuesday 9.7.	Tests with test assembly 2, changing back to the current system
Wednesday 10.7.	Comparison tests with the current system

4.5.2 Preparations before the tests

The regular test run of PF60 filter must be done with the current hydraulic system before testing the new hydraulic test unit. The regular test run is important because mechanical operations of the pressure filter must be in working order. In addition, the hydraulic cylinders are purged of air during the regular test runs.

The filter cloth is removed from the pressure filter before testing the new hydraulic concept. This way the possible oil spills do not destroy the filter cloth. Oil spills are possible when changing the hydraulic connections.

The working platform on top of the pressure filter must be removed before the hydraulic test unit can be lifted on top of the pressure filter. Also the two supports of the platform must be removed. The parts to be removed are illustrated in figure 20.

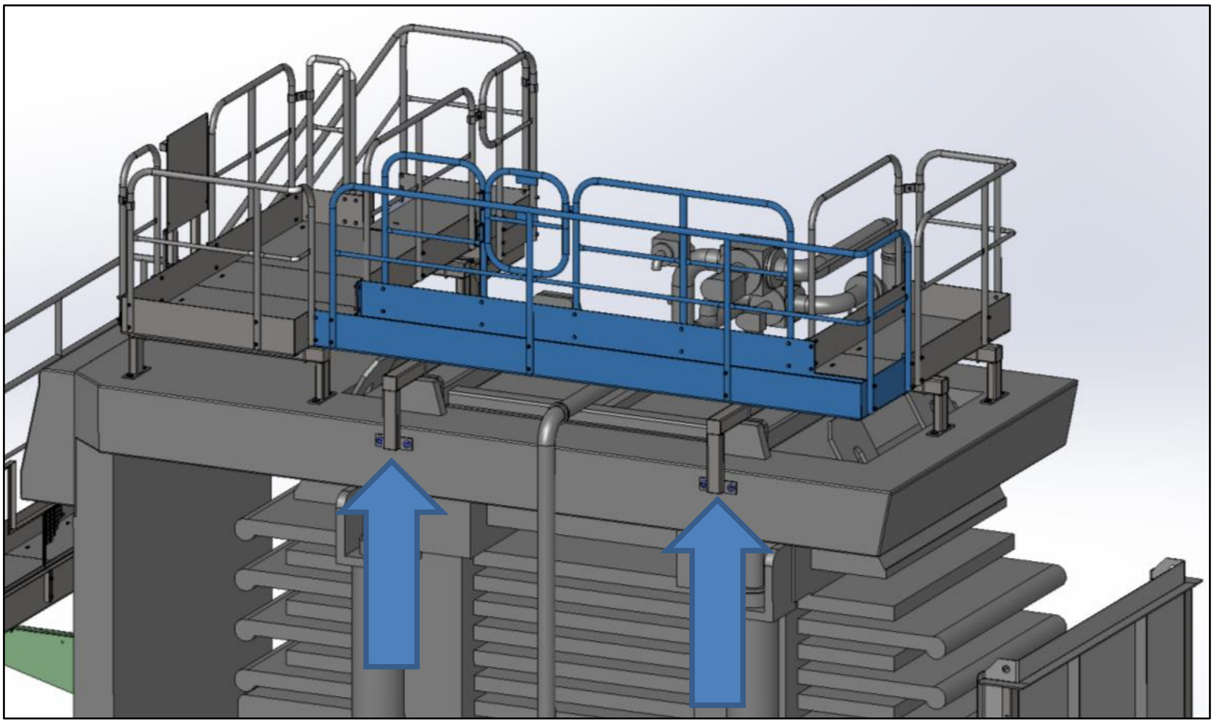


Figure 20. Maintenance platform that is disassembled before the tests.

A part of the maintenance platform that is disassembled is presented in figure 20. Disassembling the platform creates a space for the test unit. Before lifting the test unit on top of the PF60, the electrical power supply cabinet for hydraulic test unit is installed to support structure of the hydraulic test unit. Installation can be done on ground level. The electrical power supply cabinet should be ready and installed before the test hydraulic unit arrives. The location of electrical power supply cabinet is illustrated in figure 21.

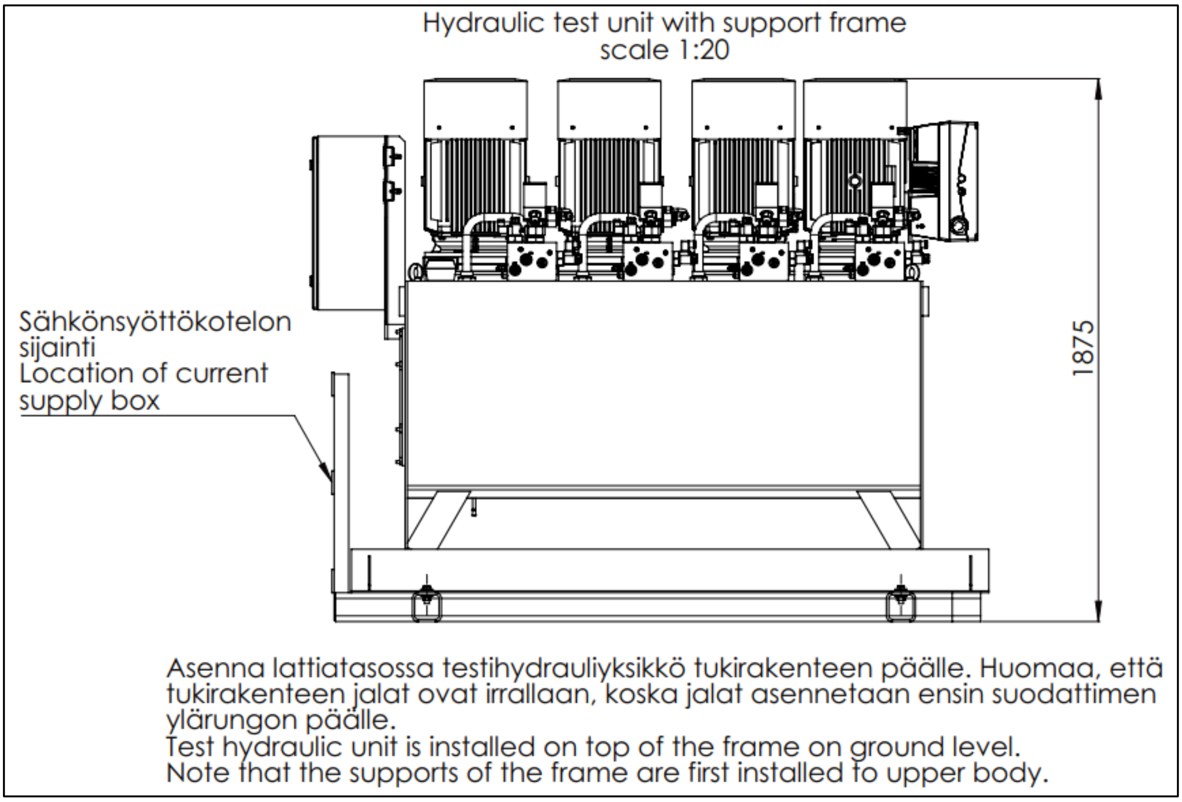


Figure 21. Location of the power supply cabinet.

Location of the electrical power supply cabinet is presented in figure 21. The same figure also illustrates the hydraulic test unit installed on top of the support frame. Frame of the hydraulic test unit has separate supports that are installed on top of PF60 before lifting the hydraulic test unit with frame on top of the supports. The supports are secured to the columns of the filter by replacing the existing bolts between upper frame and columns with longer bolts.

For test assembly one, the existing hydraulic pipelines from cylinders are disassembled. The pipes to be removed are parts 5 (4 pcs) and 8 (4 pcs) in figure 22. Plugs are installed to the ends of elbow connectors (part number 30 in figure 22). The pipes and elbow connectors are indicated in the figure 22 below. Plugs are from item structure presented in appendix V.

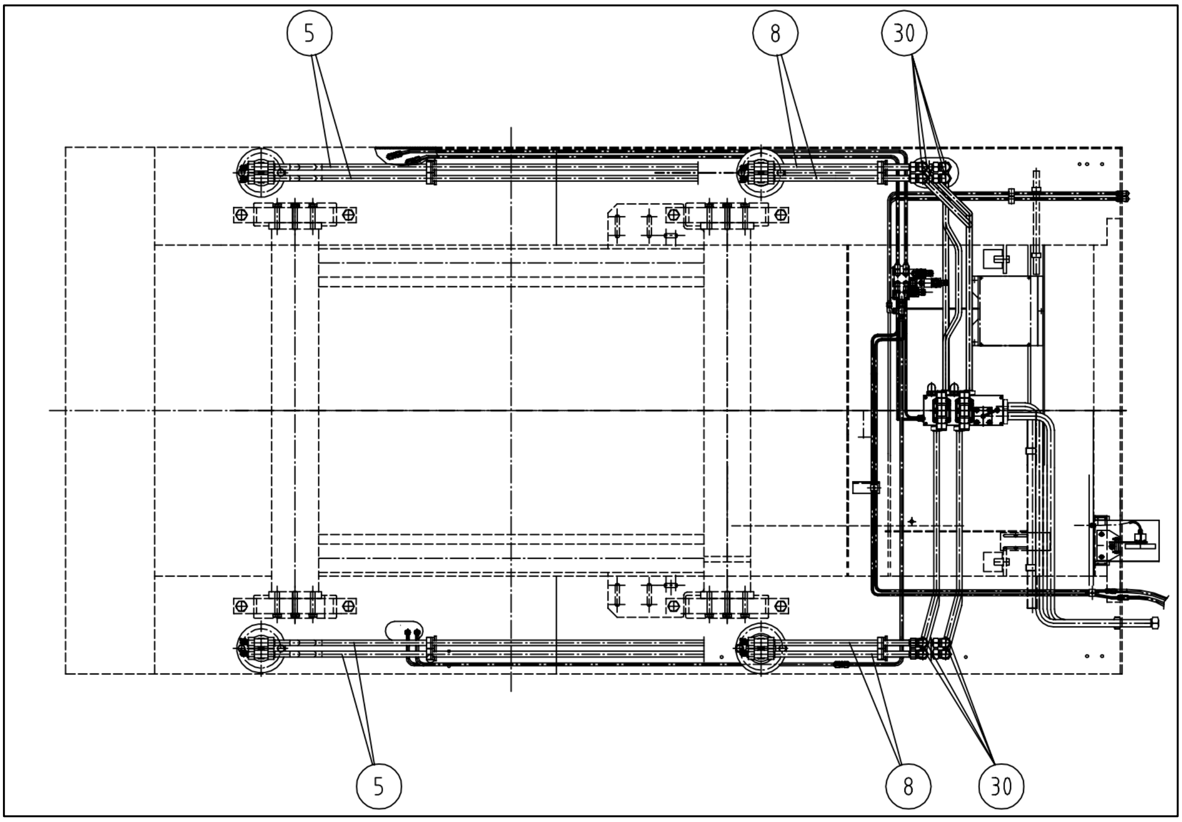


Figure 22. Hydraulic pipelines for cylinders.

The existing pipelines for hydraulic cylinders are presented in figure 22. These pipelines are disassembled because hydraulic hose connections are used for the test purposes. When the hydraulic test unit arrives, the tank of the unit is filled with hydraulic oil. The amount of oil to be filled is approximately 650 liters. The nominal volume of the tank is 600 liters, but some of the oil is filling the hydraulic hoses. The remaining electrical connections are also finished before lifting the hydraulic test unit on top of PF60. When everything is done, the test unit is lifted on top of the supports located on upper body. The unit is secured with screws. After this the hydraulic hoses can be connected between the cylinders and the hydraulic test unit. Refer to appendix V for details on connections. A photograph of the installed hydraulic test unit with hydraulic hoses is presented in figure 23.

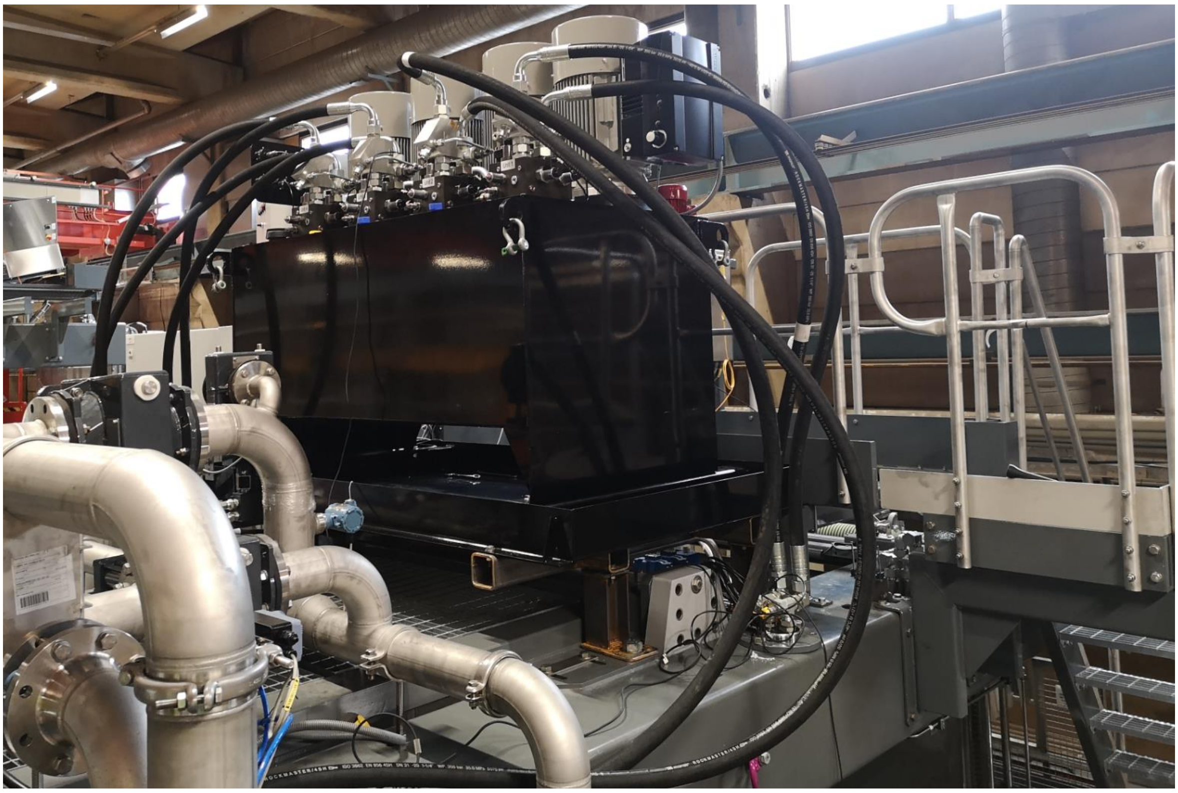


Figure 23. Hydraulic test unit installed on top of PF60.

Hydraulic hoses that are installed on cylinders and hydraulic unit are illustrated in figure 23. Pressure sensors are installed to the cylinders and connected to Hydac HMG 4000. Eight pressure transmitters are used because a transmitter is fitted to both ports of each cylinder. Four of the pressure transmitters are analog and four of them are CAN because of the limited amount of analog ports in HMG 4000. A HIT 1500 inclinometer is installed to a cover of the top pressure plate and connected to HMG. A cover plate with holes for HIT 1500 is in item structure presented in appendix IV.

The preparations also include changing the connections for the second test assembly when the tests are done with the first test assembly. Hydraulic diagram of test assembly 2 is presented in appendix VI. All hydraulic hoses between cylinders and hydraulic test unit are disconnected from the cylinders. All hoses are also disconnected from the test unit except the hoses connected to ports A2 and B2. The hydraulic pipelines that were earlier disassembled are assembled back to the cylinders. Pipelines number 51 and 52 from the QAC block are disassembled and plugs are installed to the ends of lines 51 and 52. Plugs are in item structure that is presented in appendix V. Pipes with T-connectors are installed to ports 51 and 52 of QAC block. Location of the pipes is illustrated in figure 24.

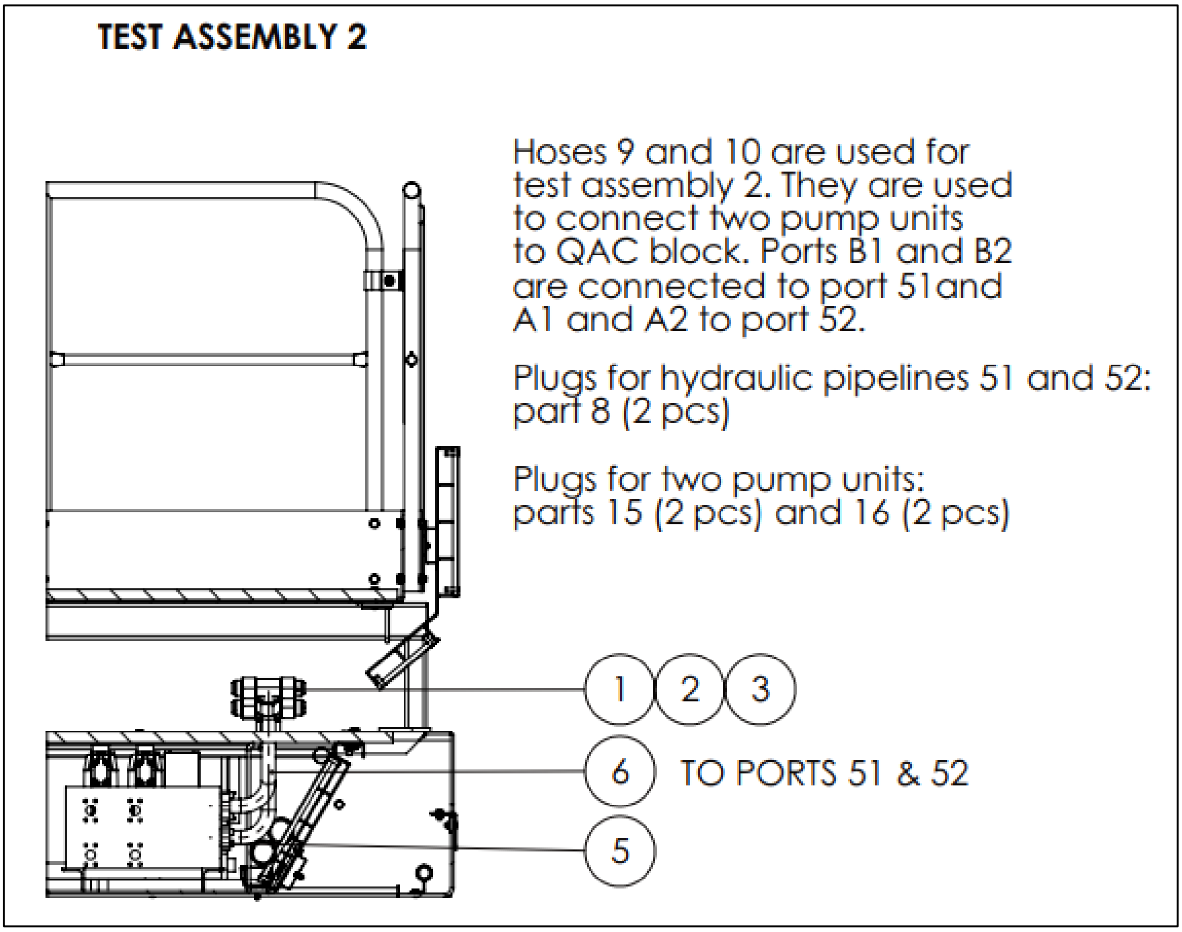


Figure 24. Location of the hydraulic pipes for the second test connection.

Hoses from the hydraulic test unit ports B1 and B2 are connected to port 51. Ports A1 and A2 are connected to port 52. Refer to appendices V and VI for details. The two motor-pump units (3 & 4) that are not used in this test are plugged. The load-control valves of hydraulic test unit are replaced by two plugs because the QAC valve block includes a load-control valve. The second test assembly requires control pressure for the valves from the current system. Therefore the current hydraulic system must be operating during the tests with test assembly 2.

4.5.3 Measurement devices

The test equipment includes Outotec Larox PF72/96 M60 type filter. Filtration area of the filter is 72 m^2 and the total mass of the plate pack is 33 000 kg. Hydac Kinesys hydraulic test unit is used to test the plate pack movements of the filter. The test unit has four motor-pump units with a power of 22 kW each. Tank of the test unit is filled with ISO VG 46 hydraulic oil (Shell Tellus S2 VX). Measuring device is Hydac HMG4000 portable data recorder. The following sensors are connected to the data recorder:

- five analog pressure transmitters for cylinders 1 and 2 and motor-pump unit 2, type Hydac HDA 4748-H-0250-000
- four CAN pressure transmitters for cylinders 3 and 4, type Hydac HDA 4748-F11-0250-000
- one inclinometer, type Hydac HIT 1508-F11-X-180-Y-90-1-000
- one energy measurement device, type Janitza UMG96-PA
- one rope length transmitter, type FSG SL3002-PK1023-MU/GS80/G/F.

Locations and connections of the sensors are illustrated in figure 25.

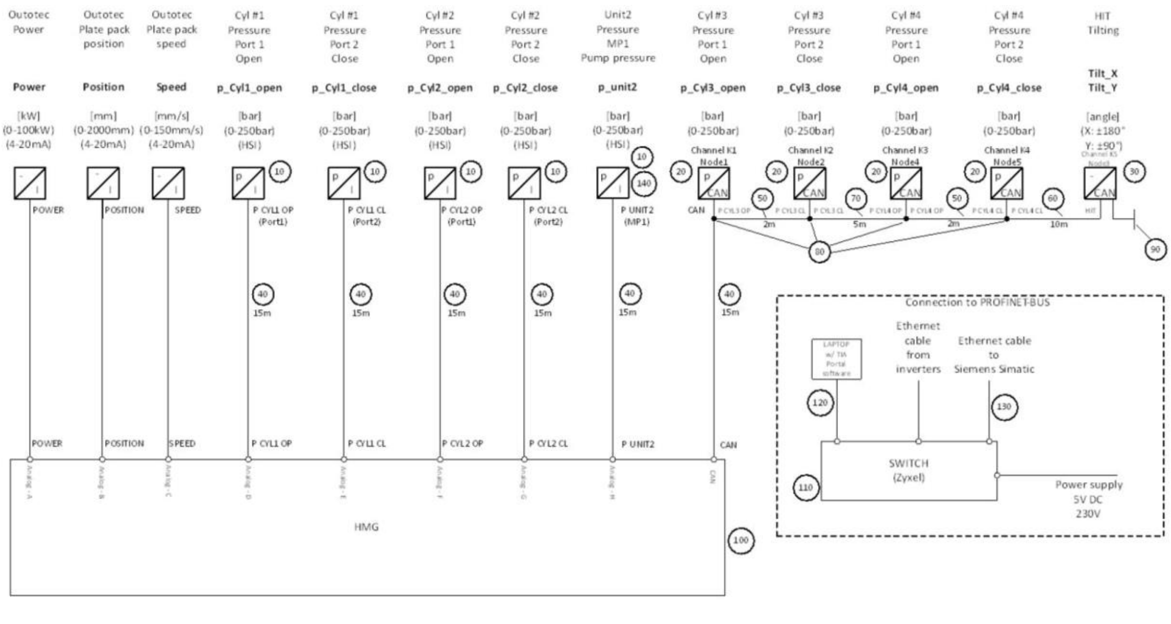


Figure 25. Measuring equipment and connections (Hydac 2019c).

All sensors that are used in the tests are presented in figure 25. Pressure transmitters are installed to both ports of each cylinder. Figure 26 illustrates the pressure transmitters on cylinder four.

Figure 26. Pressure transmitters fitted in the ports of cylinder four.

The pressure transmitters illustrated in figure 26 are CAN transmitters. All of the CAN transmitters are connected in series. Both of the cylinder ports have tee fittings to branch the ports for hose connection and pressure transmitter. The tee fittings have a measuring connection for easy installation of pressure transmitters. A DN2 measuring hose is used between the measuring connection and pressure transmitter. The hose enables that the pressure transmitters can be located more freely compared to installation directly into the measuring connection. It would be difficult to install the pressure transmitters if there were no hoses. This considers especially cylinder number two.

A pressure transmitter is installed after one pump of the test unit to monitor the pressure. Only one pump has a pressure transmitter in this test, because the amount of analog ports in the data recorder is limited. The pressure transmitter of the pump is illustrated in figure 27.

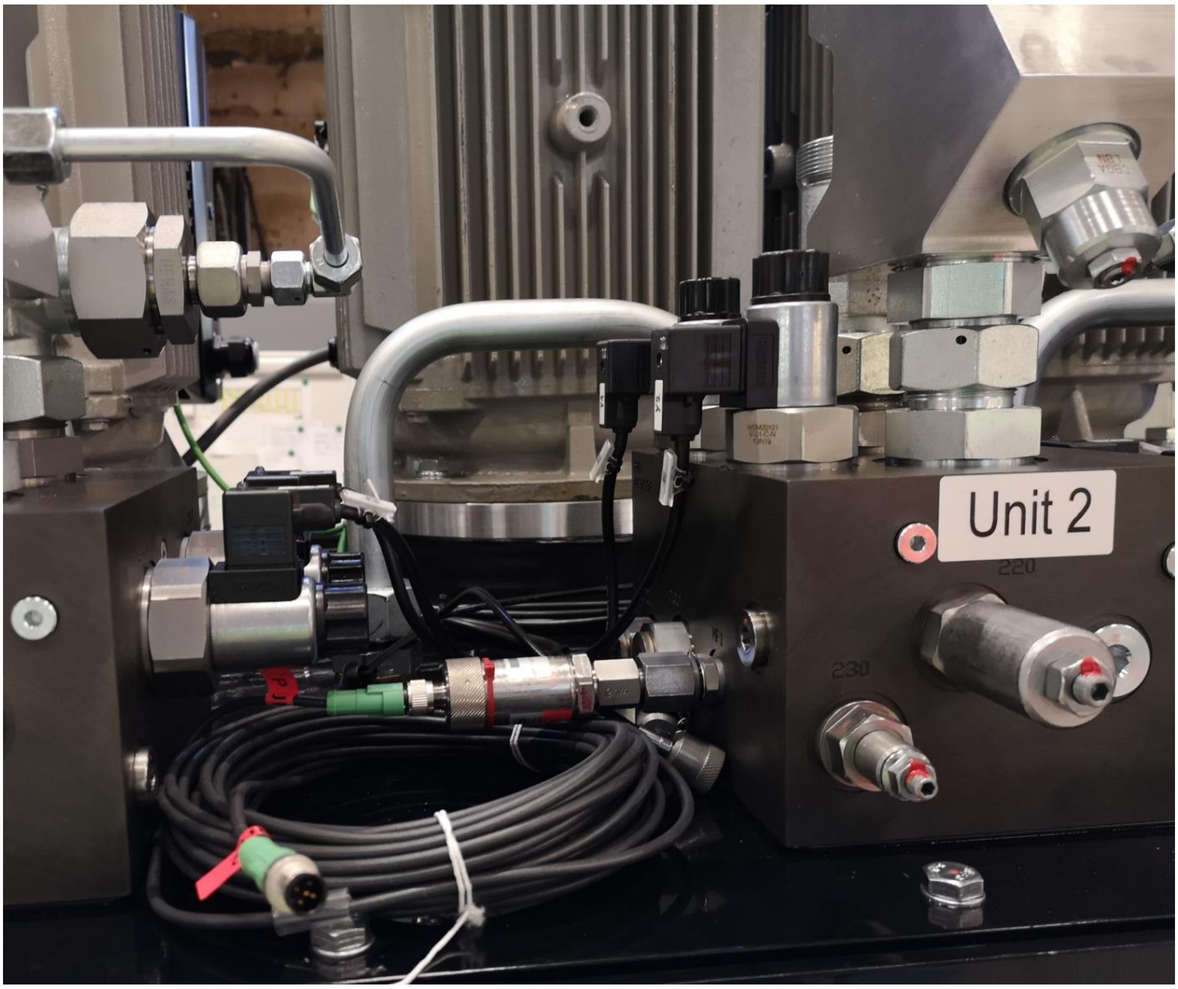


Figure 27. Pressure transmitter is fitted to pump unit 2.

As illustrated in figure 27, pump unit 2 has a pressure transmitter. Unit 2 is chosen to be the location for a pressure transmitter because it is at use on both of the test assemblies.

An inclinometer can be used to record the synchronization of the hydraulic cylinders. If the hydraulic cylinders are not synchronized, the upper pressure plate of PF60 will incline. Location of the inclinometer is illustrated in figure 28.

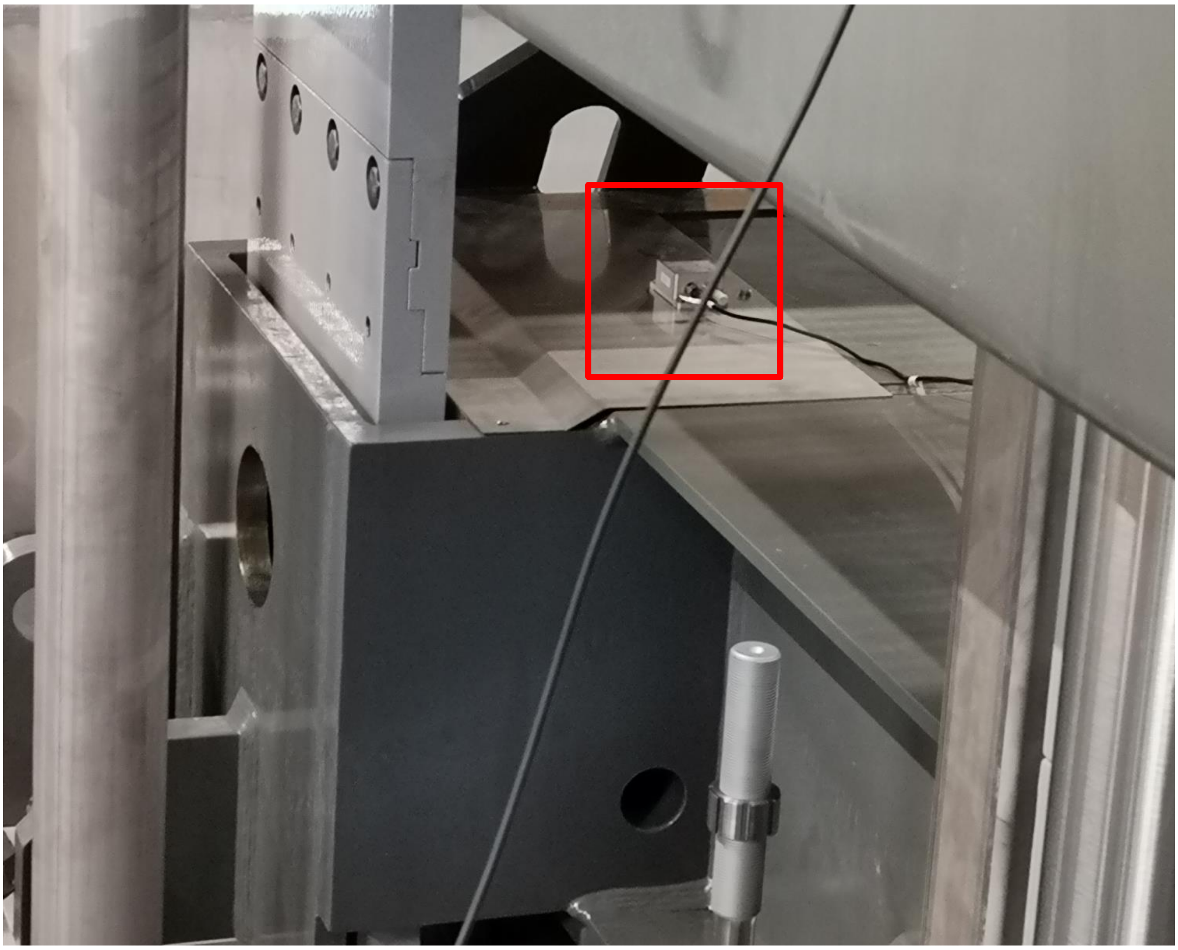


Figure 28. Location of the inclinometer.

The inclinometer is installed on top of the upper pressure plate, as illustrated in figure 28. The inclinometer is configured to detect the inclination of upper pressure plate between every 0,1 degrees. The data provided by the inclinometer is useful during the tests as inclination angle should be observed because too much inclination can damage the PF60 filter. When analyzing the results of the tests, angle of inclination is taken into account to compare the systems.

An energy measurement device is required to record the active power. The device is located in the electrical power supply cabinet, as illustrated in figure 29.

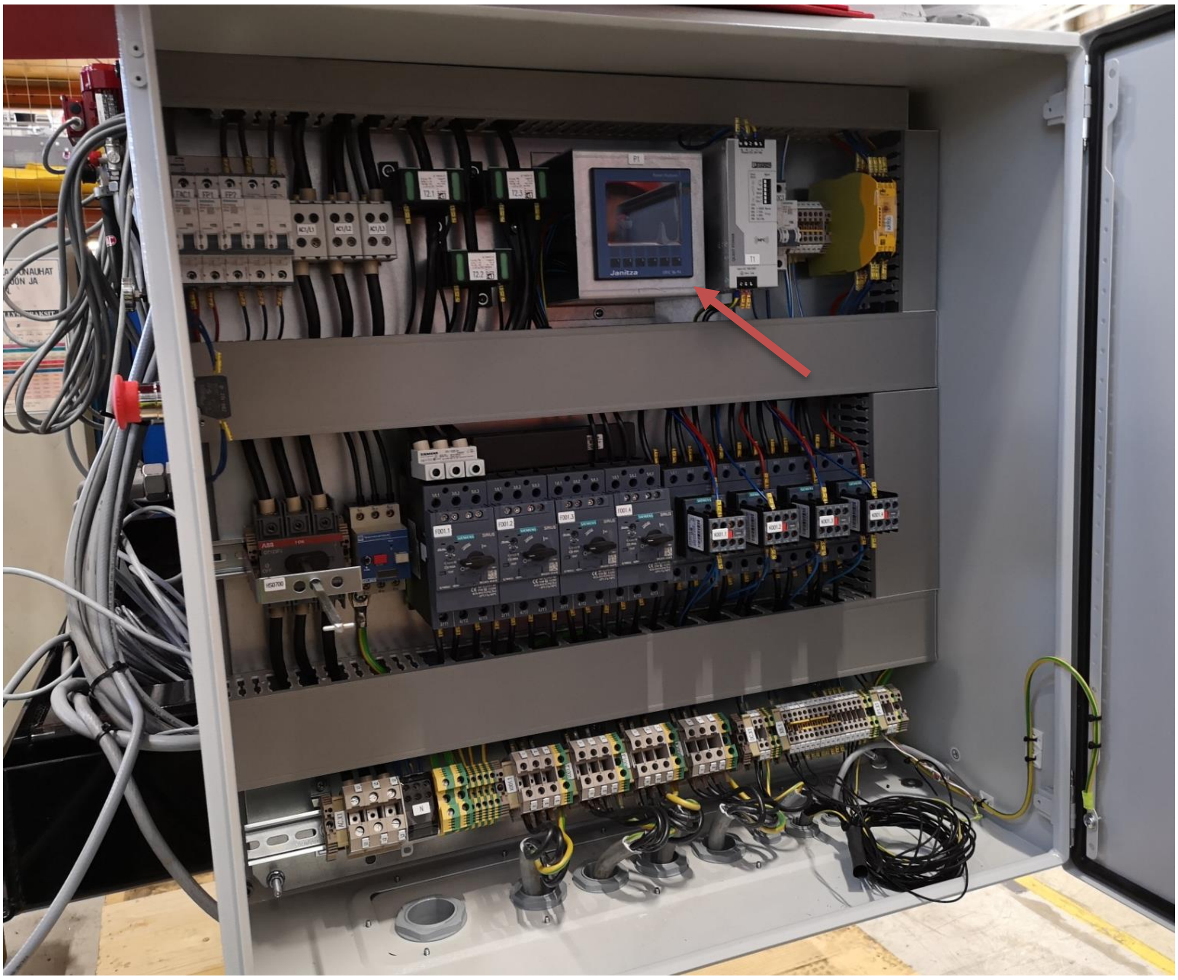


Figure 29. Location of the energy measurement device (shown with an arrow) in the power supply cabinet.

The energy measurement device (illustrated in figure 29) is configured to measure the active power. The output is analog and the range is between 4 to 20 mA. The starting value is configured at 0 kW and the end value at 100 kW.

Rope length transmitter is a standard component of PF60 filter. It is used to monitor the position of the upper pressing plate and to calculate the speed of the plate pack during opening and closing movement. Information of the plate pack position and speed is required during the tests, so analog signal between 4 to 20 mA is delivered to the data recorder from the own logic of PF60. The starting value for speed is 0 mm/s and for position 0 mm whereas the end values are 100 mm/s for speed and 2000 mm for position.

4.5.4 Measuring principles for the tests

Measurement devices are presented in the previous chapter. Hydac HMG portable data recorder is configured to have measuring frequency of 20 ms. Each measurement will consist of opening the plate pack, short waiting time in open position and closing the plate pack. Various speeds are measured with each test assembly. The same speed is measured four times in a row. Temperature of the oil is measured before and after each speed. If temperature of the oil is low before the measurement, few practicing drives will be done to heat the oil to about 30°C.

The speed of the cylinders is adjusted manually with a potentiometer when measuring test assemblies one and two (with the hydraulic test unit). This can cause variation to the actual speed as it is difficult to reach exactly the same speed every time. There is also a need to ramp down the speed when approaching end positions of the cylinders. In the second test assembly, two motor-pump units are connected to the current valve block of PF60, the QAC block. In this case, about 100 mm before the cylinder end positions, the connection is changed to parallel connection. This is a standard principle when operating with the current system and in normal operation it is done automatically. When measuring test assembly 2, the connection is changed by pushing a button on the handheld control unit of PF60. It will ensure that all cylinders will reach end positions at the same time. When measuring the current hydraulic system, the speed of the plate pack is adjusted by volume flow request. The requested volume flow is first entered to the system and the opening and closing movement is controlled with handheld control unit. The system will automatically switch the series connection to parallel connection about 100 mm before the end positions.

Rotating speed of the motors is written down when testing the first and second test assembly. The actual volume flow can then be calculated based on the rotating speed. When measuring the current system, volume flow request is written down. Before starting the measurements, it must be ensured that there is no air in the system. Therefore, the system must be purged of air. In case of the first test assembly, air is purged from cylinders one by one. The measuring connections on the ports of the cylinders can be used to purge air. The pressure transmitters are simply removed from the end of the measuring hose and while slowly driving the cylinder, air and oil are bled to a bucket from the active side of the cylinder chamber. The second test assembly has QAC block connection, and QAC block has valves

PR51 and PR52, which are helping to purge the air out of the system. That is why there is no need to purge the air out of the cylinders one by one.

The motor-pump units are connected to hydraulic cylinders with 1 ¼ inch hydraulic hoses. Lengths of the hoses are the following: 4,8 meters for cylinders one and four, 3,1 meters for cylinder two and 2,4 meters for cylinder three. 4,8 meters long hoses are used for the second test assembly, where two motor-pump units are connected to the current QAC block. The connection from QAC block to the cylinders is the same as in the current hydraulic system.

4.5.5 The conducted tests

The tests were conducted during week 28 in July 2019. In addition to that week, the previous week was also spent with practical preparations for the tests. The tests were conducted in Outotec’s manufacturing facilities in Lappeenranta, Finland. The preparations were done according to the instructions formulated in this thesis. The preliminary schedule was a bit too tight due to challenges related to active power measuring, but there was enough time to conduct all the tests, after all.

In total, 52 actual tests were conducted. In addition to the actual test drives, 16 test drives were recorded with Hydac HMG. Besides the recorded test drives, there were many unrecorded test drives to check that everything functions correctly and to get familiar with the system. The sequence of the tests was as planned so the first assembly to be tested was test assembly one. When the tests with test assembly one were ready, the connections were changed to test assembly two and the tests were conducted with that assembly. Finally, the hydraulic connections were restored for the current hydraulic system and comparison tests were done.

There were 16 actual tests recorded with test assembly one. The tests are presented in table form in table 3.

Table 3. Measurements of the first test assembly.

Test assembly one				
number of the measurement	speed (mm/s)	frequency (Hz)	rotating speed (rpm)	temperature of oil (°C)
1	40	41	1230	33
2	40	41	1230	
3	40	41	1230	
4	40	41	1230	
5	60	60	1800	34
6	60	60	1800	
7	60	60	1800	
8	60	60	1800	
9	80	80	2400	36
10	80	80	2400	
11	80	80	2400	
12	80	80	2400	
13	85	85	2550	37,5
14	85	85	2550	
15	85	85	2550	
16	85	85	2550	39,5 (end)

As presented in table 3, four different speeds were tested with test assembly one. These include the cylinder speeds of 40 mm/s, 60 mm/s, 80 mm/s and 85 mm/s. Each speed was tested four times. Frequency was checked from Siemens TIA Portal software that was connected to the Profinet Bus of frequency converters. The frequency is then used to calculate the rotating speeds of the motors. Frequency of 100 Hz corresponds to 3000 rpm. Temperature of hydraulic oil in the tank was checked before and after each speed.

The 12 tests recorded with test assembly two are presented in table 4.

Table 4. The second test assembly.

Test assembly two				
number of the measurement	speed (mm/s)	frequency (Hz)	rotating speed (rpm)	temperature of oil (°C)
1	40	41	1230	32
2	40	41	1230	
3	40	41	1230	
4	40	41	1230	
5	60	60	1800	32
6	60	60	1800	
7	60	60	1800	
8	60	60	1800	
9	70	70	2100	33
10	70	70	2100	
11	70	70	2100	
12	70	70	2100	34 (end)

As presented in table 4, the cylinder speeds that were tested included 40 mm/s, 60 mm/s and 70 mm/s. There was no possibility to test higher speeds than 70 mm/s because the power of only two motors (44 kW) was not enough. The frequencies and rotating speeds are included in the table as with test assembly one. In addition, the temperature of hydraulic oil in the tank was checked similarly as with test assembly one.

Total of 24 comparison tests were recorded with the current hydraulic system. Tests of the current hydraulic system are presented in table 5.

Table 5. The current hydraulic system.

The current hydraulic system			
number of the measurement	speed (mm/s)	volume flow request (l/min)	temperature of oil (°C)
Parallel connected cylinders			
1	40	310	31
2	40	310	
3	40	310	
4	40	310	36
Series connected cylinders			
1	40	165	36
2	40	165	
3	40	165	
4	40	165	
5	60	240	37
6	60	240	
7	60	240	
8	60	240	
9	70	275	39
10	70	275	
11	70	275	
12	70	275	
13	80	305	43
14	80	305	
15	80	305	
16	80	305	
17	85	330	44
18	85	330	
19	85	330	
20	85	330	47 (end)

As presented in table 5, the tests of the current hydraulic system are divided into two parts: parallel connected cylinders and series connected cylinders. Parallel connection is included to the tests to achieve more comparative recordings with test assembly one. The parallel connection includes only one cylinder speed, 40 mm/s. The series connected cylinders connection was tested with 20 tests. All of the speeds that were recorded with test assemblies one and two were tested with the current system. These include the speeds of 40 mm/s, 60 mm/s, 70 mm/s, 80 mm/s and 85 mm/s. Volume flow request of each corresponding speed is listed in the table. Temperatures of hydraulic oil in the tank were checked similarly as with test assemblies one and two.

During the tests it was noticed that the speed recording was about five seconds behind the actual measuring. This appeared when the cylinders had reached their end position, but according to the speed recording the cylinders were still moving. Figure 30 illustrates lagging of the speed.

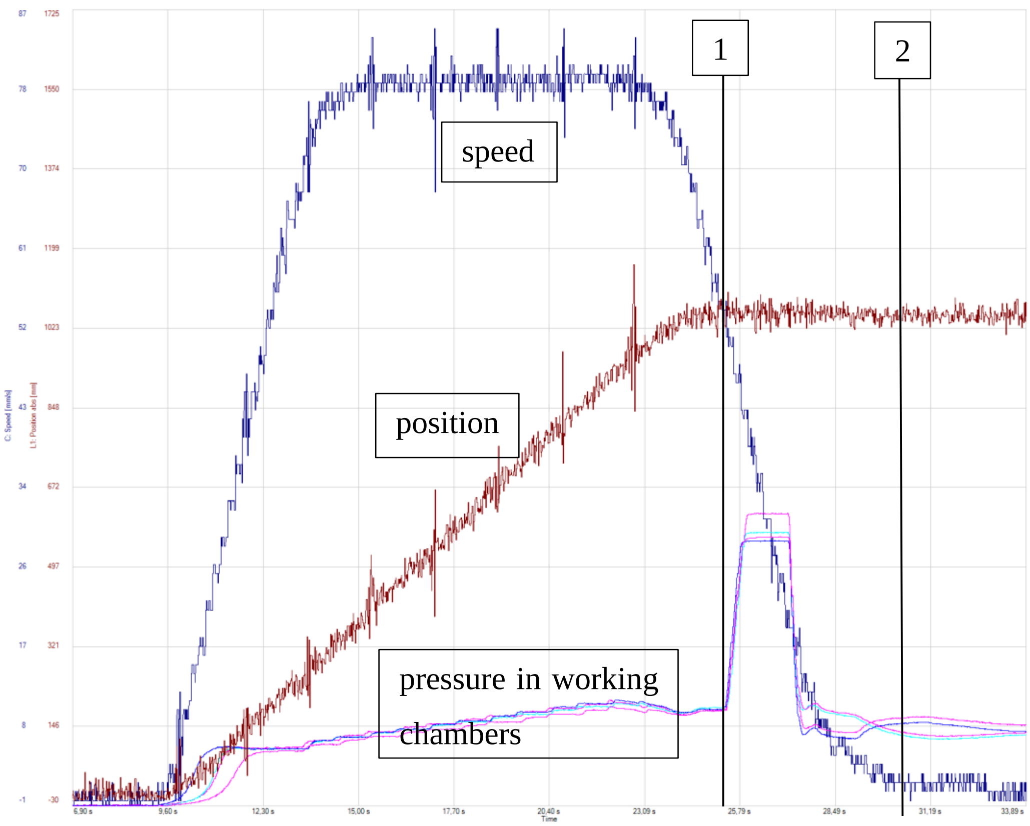


Figure 30. Speed lags about five seconds behind real position.

Speed, position and pressure of the cylinders are presented in figure 30. In point 1, the pressure in cylinders’ chambers rises up to the pressure limit of 80 bar. This means that the cylinders have reached their end positions. Also the position sensor states that the plate pack is not moving at point 1. At the same time, speed still states that the cylinders are moving. According to the speed measurement, the movement stops at point 2. The difference in time between points 1 and 2 is approximately 5 seconds. Therefore, when analyzing the results, it is better to look at the position curve to find the alterations of the speed. As described before, speed does not have own sensor in the tests. The position information is used to calculate the speed of the cylinders by own PLC system of PF60 filter. There was no possibility to alter the method of calculation, so the recorded speed should be used only as a refence of the maximum speed.

5 RESULTS

The results are presented in this chapter. This chapter includes results of test assemblies one and two and results of the current hydraulic system. Finally, the results of the test assemblies and the current system are compared. Tables of the results are presented in appendix VII.

5.1 General observations of the recorded tests

When the analysis of the results was started, it was observed that there is a difference between the recorded speed and the calculated speed of the cylinders. During the tests it was also noticed that the speed lags behind the actual speed of the cylinders. Actual speed of the cylinders is calculated from the results. The speed is calculated from constant speed area during opening of the plate pack and arithmetic mean of all tests is calculated. The speeds are presented in table 6.

Table 6. Calculated speeds during the tests.

target / recorded speed (mm/s)	**Calculated speed (mm/s)**			
all tests	**test assembly 1**	**test assembly 2**	**the current system (cylinders in series)**	**the current system (cylinders in parallel)**
40	39,1	38,7	38,5	38,2
60	57,3	56,1	58,0	-
70	-	65,7	67,0	-
80	75,9	-	76,0	-
85	79,6	-	81,8	-

Comparison of the speeds is presented in table 6. The target speeds are in the left column. Generally, the target speeds are higher than the speeds calculated from the position and time recordings. There is an error in the speed calculation of PF60 based on the calculated actual speeds. There are no significant differences between the actual calculated speeds, so the results can be compared. The recorded speeds in the tests were calculated by own PLC of PF60 filter. It was assumed that the speed is correct, and the volume flows were adjusted based on that speed. That is why the actual speeds are lower compared to the target speeds.

The recorded tests are analyzed with HMGWin program. HMGWin is a program for analyzing data that is recorded with Hydac HMG portable data recorder. There is a need to filter some of the recorded data because there is some interference in power measurement and position measurement. Filtering is done with HMGWin's "filtered channel" function that is an averaging filter.

5.2 Results of test assembly 1

Test assembly 1 was tested with four different target speeds: 40 mm/s, 60 mm/s, 80 mm/s and 85 mm/s. A full test cycle of the target speed of 40 mm/s is presented in figure 31.

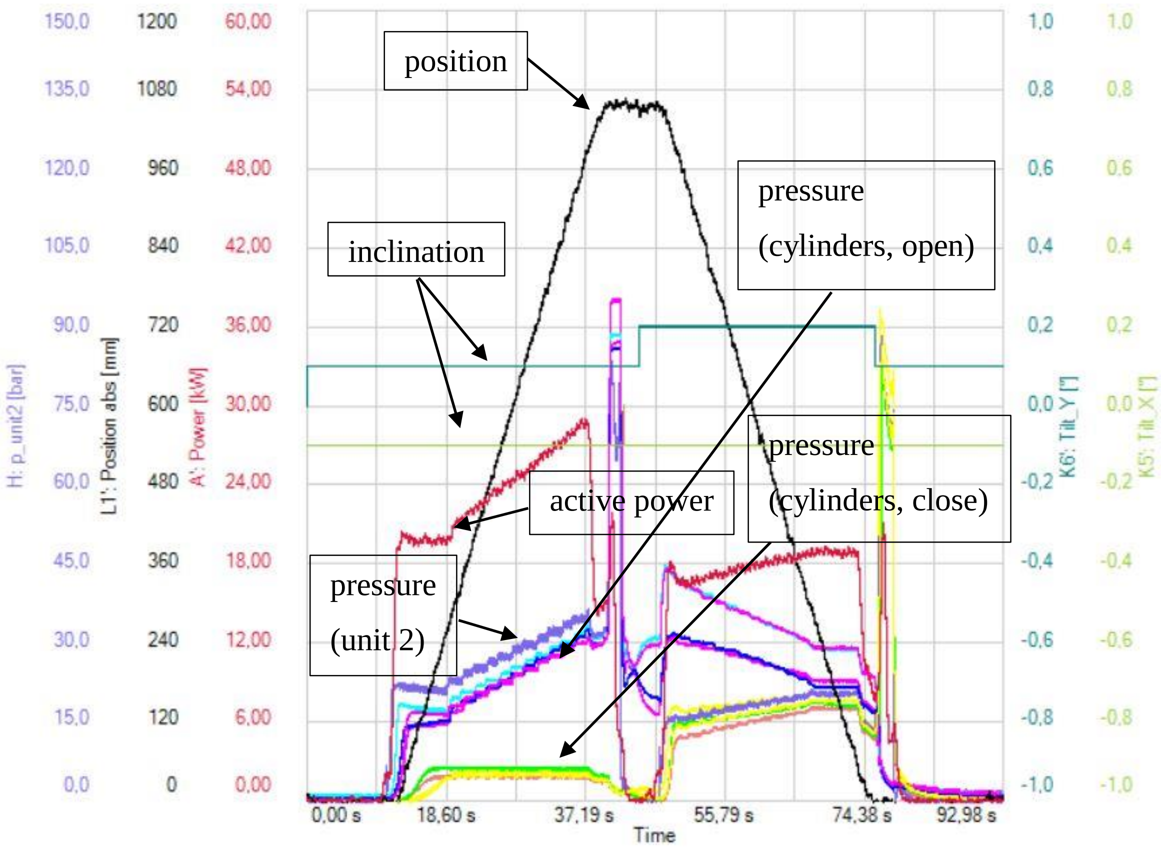


Figure 31. A full test cycle of test assembly 1, test number 2 (40 mm/s).

The test cycle presented in figure 31 is of test number 2 with test assembly 1 and the target speed is 40 mm/s. The different recordings are marked with their own colors in the figure. The pressure scale on the left is the same for all measured pressures. Angle of inclination is presented with two axes, X and Y, on the right side. During opening of the plate pack, the pressure increases in the working chambers of the cylinders. The pressure increases step-by-step as the cylinders lift the filter plates. The working chambers during opening of the plate pack are referred as “cylinders, open” in the figure. On the other hand, “cylinders, close” refers to working chambers of the cylinders when the plate pack closes. Pressure of motor-pump unit 2 is also visible in the figure. Inclination of the plate pack is scaled between -1° and 1° in the figure because the angles of inclination are small. Table 7 summarizes the results of tests 1 – 4.

Table 7. Results of tests 1 – 4 of test assembly 1.

Test assembly 1, tests 1 – 4												
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack					
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
40	39,1	19,6	28,4	21,2	35,1	0,1	16,5	18,7	16,2	20,8	0,1

The results presented in table 7 are calculated based on the tests. The values are from constant speed area. The results are arithmetic means of the measured values. The only exception is the angle of inclination. The angle of inclination in the table is the maximum measured inclination from the initial position during the tests. Based on the rotating speed of 1230 rpm of the motor, the volume flow was approximately 77,9 l/min per cylinder when the target speed was 40 mm/s and volumetric efficiency of the pump is assumed to be 0,97.

A full test cycle of target speed of 60 mm/s is presented in figure 32 and the summarized results are presented in table 8.

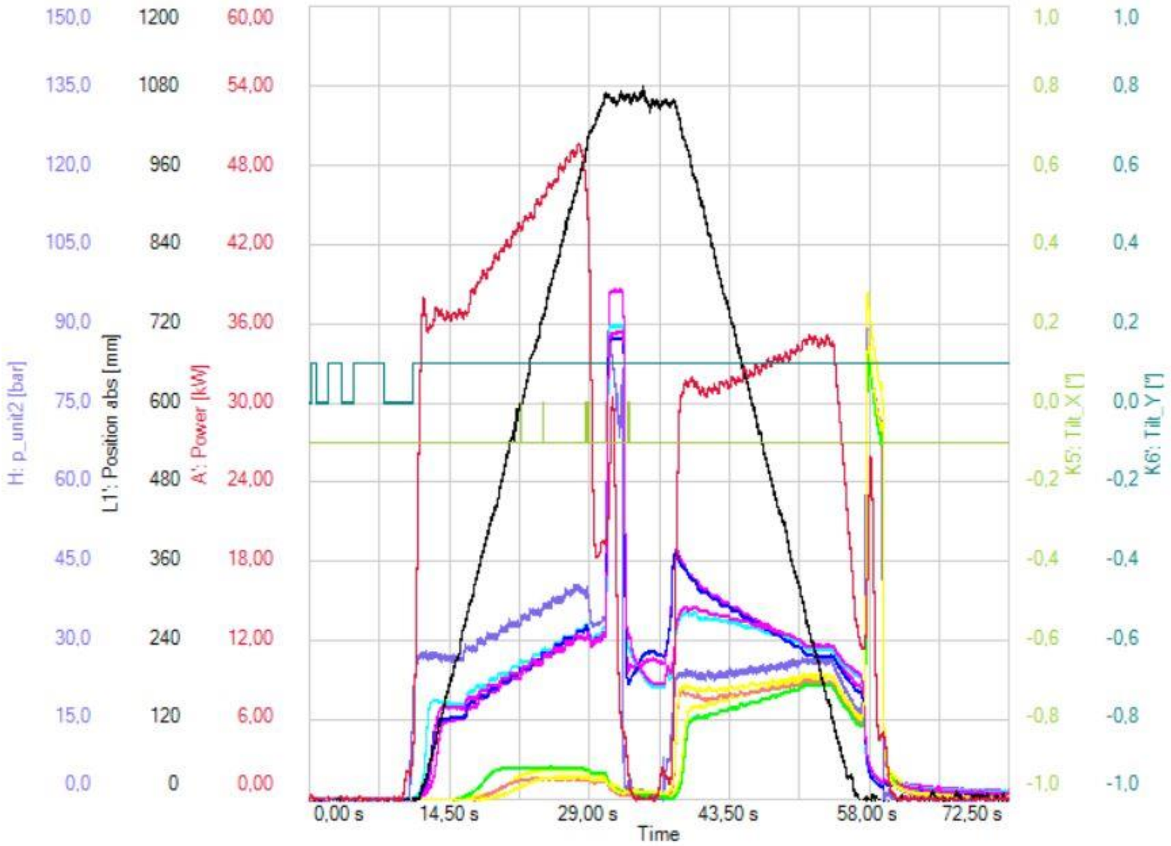


Figure 32. A full test cycle of test assembly 1, test number 5 (60 mm/s).

Table 8. Results of tests 5 – 8 of test assembly 1.

Test assembly 1, tests 5 – 8											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
60	57,3	36,0	48,9	26,6	40,3	0,1	30,5	34,9	23,5	26,3	0,1

A figure and table of the target speed of 60 mm/s are presented in figure 32 and table 8. The colors of the different recordings in the figure are similar as in the previous figure. The results of the table are from the area of constant speed, except for the angle of inclination. Based on the rotating speed of 1800 rpm of the motor, the volume flow was approximately 114 l/min per cylinder when the target speed was 60 mm/s and volumetric efficiency of the pump is assumed to be 0,97.

A full test cycle of target speed of 80 mm/s is presented in figure 33 and the summarized results are presented in table 9.

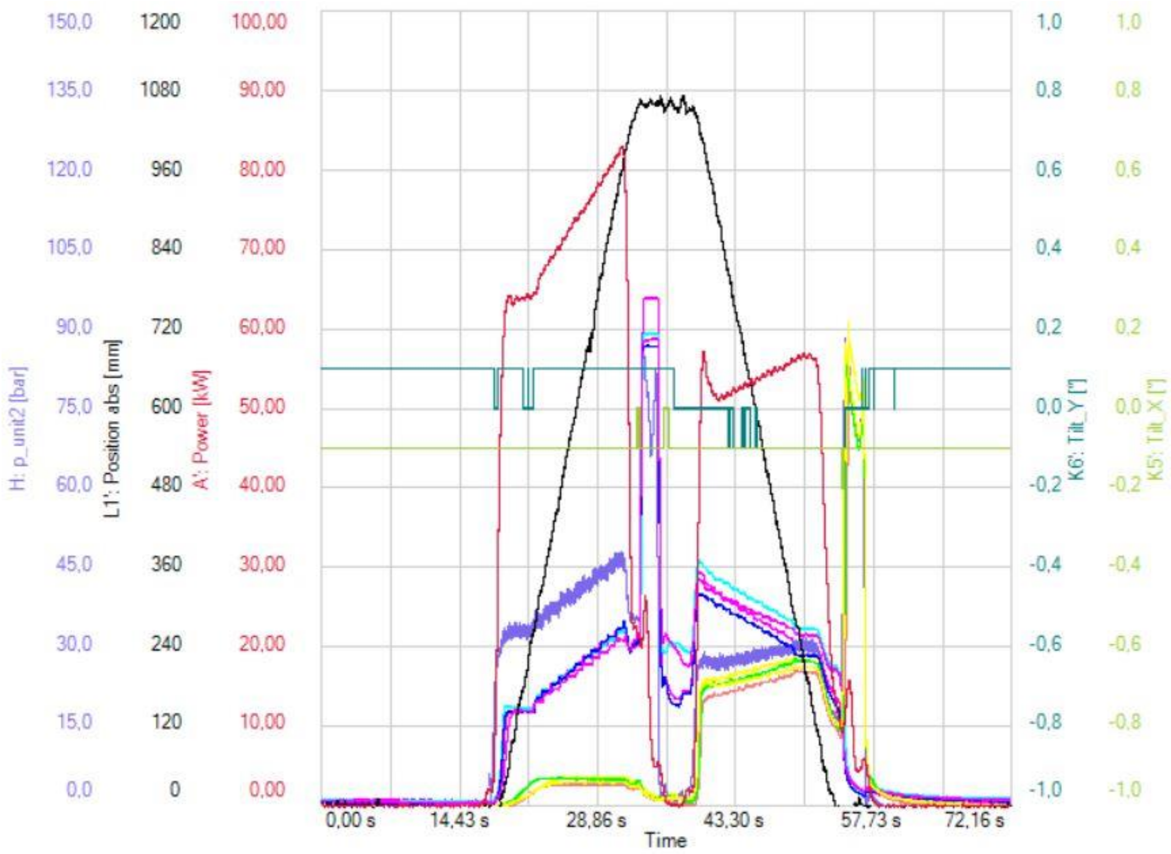


Figure 33. A full test cycle of test assembly 1, test number 12 (80 mm/s).

Table 9. Results of tests 9 – 12 of test assembly 1.

Test assembly 1, tests 9 – 12											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
80	75,9	63,5	81,5	31,7	46,4	0,1	52,4	58,1	27,5	31,8	0,2

A figure and table of the target speed of 80 mm/s are presented in figure 33 and table 9. The colors of the different recordings in the figure are similar as in the previous figures. The results of the table are from the area of constant speed, except for the angle of inclination. Based on the rotating speed of 2400 rpm of the motor, the volume flow was approximately

152 l/min per cylinder when the target speed was 80 mm/s and volumetric efficiency of the pump is assumed to be 0,97.

A full test cycle of target speed of 85 mm/s is presented in figure 34 and the summarized results are presented in table 10.

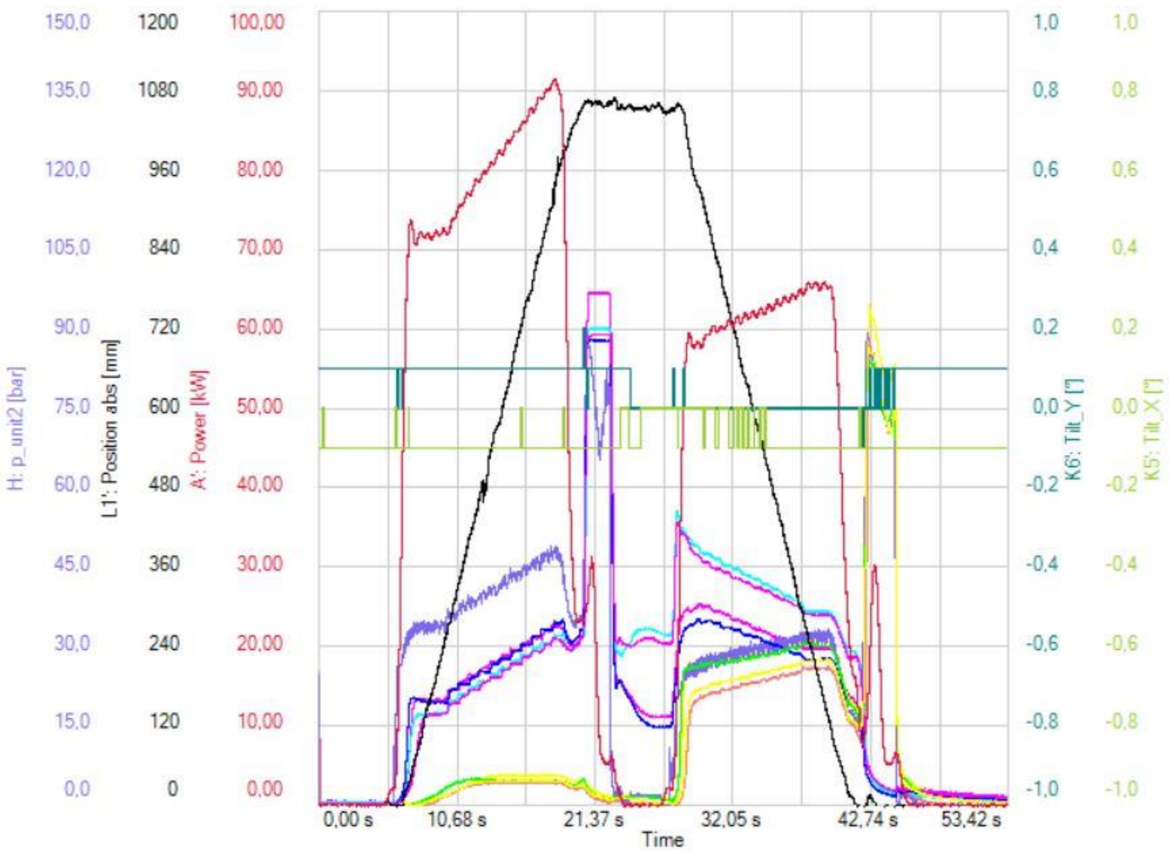


Figure 34. A full test cycle of test assembly 1, test number 15 (85 mm/s).

Table 10. Results of tests 13 – 16 of test assembly 1.

Test assembly 1, tests 13 – 16											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
85	79,6	71,8	90,8	34,1	47,3	0,2	59,4	65,6	27,3	32,8	0,2

A figure and table of the target speed of 85 mm/s are presented in figure 34 and table 10. The colors of the different recordings in the figure are similar as in the previous figures. The results of the table are from the area of constant speed, except for the angle of inclination. Based on the rotating speed of 2550 rpm of the motor, the volume flow was approximately 162 l/min per cylinder when the target speed was 85 mm/s and volumetric efficiency of the pump is assumed to be 0,97.

5.3 Results of test assembly 2

Test assembly 2 was tested with three different target speeds: 40 mm/s, 60 mm/s and 70 mm/s. A full test cycle of the target speed of 40 mm/s is presented in figure 35 and the summarized results are presented in table 11.

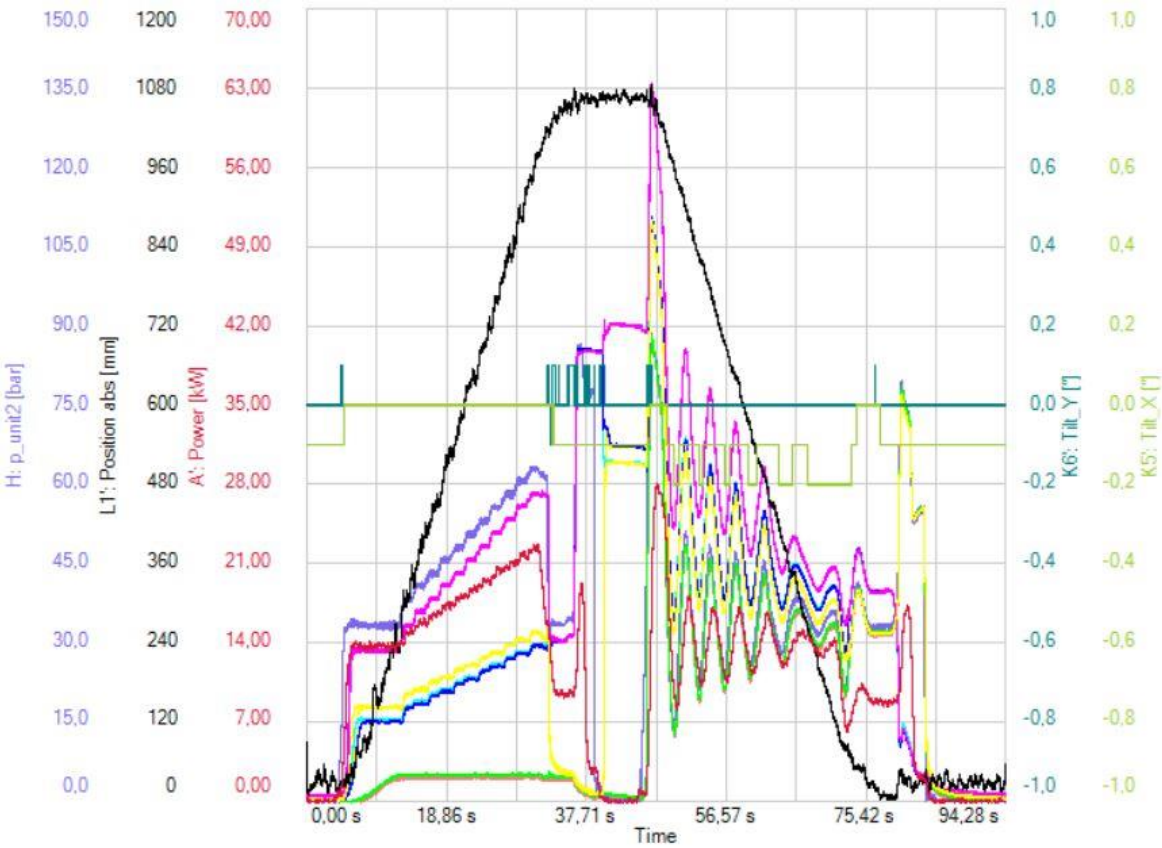


Figure 35. A full test cycle of test assembly 2, test number 3 (40 mm/s).

Table 11. Results of tests 1 – 4 of test assembly 2.

Test assembly 2, tests 1 – 4												
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack					
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
40	38,7	14,0	22,1	33,9	62,1	0,2	12,4	15,9	29,8	38,3	0,2

A figure and table of the target speed of 40 mm/s are presented in figure 35 and table 11. The colors of the different recordings in the figure are similar as in the previous figures. The results of the table are from the area of constant speed, except for the angle of inclination. Based on the rotating speed of 1230 rpm of the motor, the volume flow was approximately 77,9 l/min per pump unit when the target speed was 40 mm/s and volumetric efficiency of the pump is assumed to be 0,97. There were two motor-pump units at use during the tests with test assembly two, leading to a volume flow of 156 l/min. During closing of the plate pack, there was a distinct noise caused by the counterbalance valve. The counterbalance valve is optimized for a higher volume flow and it caused pressure fluctuation in the system. The fluctuation of pressure is also visible in figure 35 during closing of the plate pack.

A full test cycle of target speed of 60 mm/s is presented in figure 36 and the summarized results are presented in table 12.

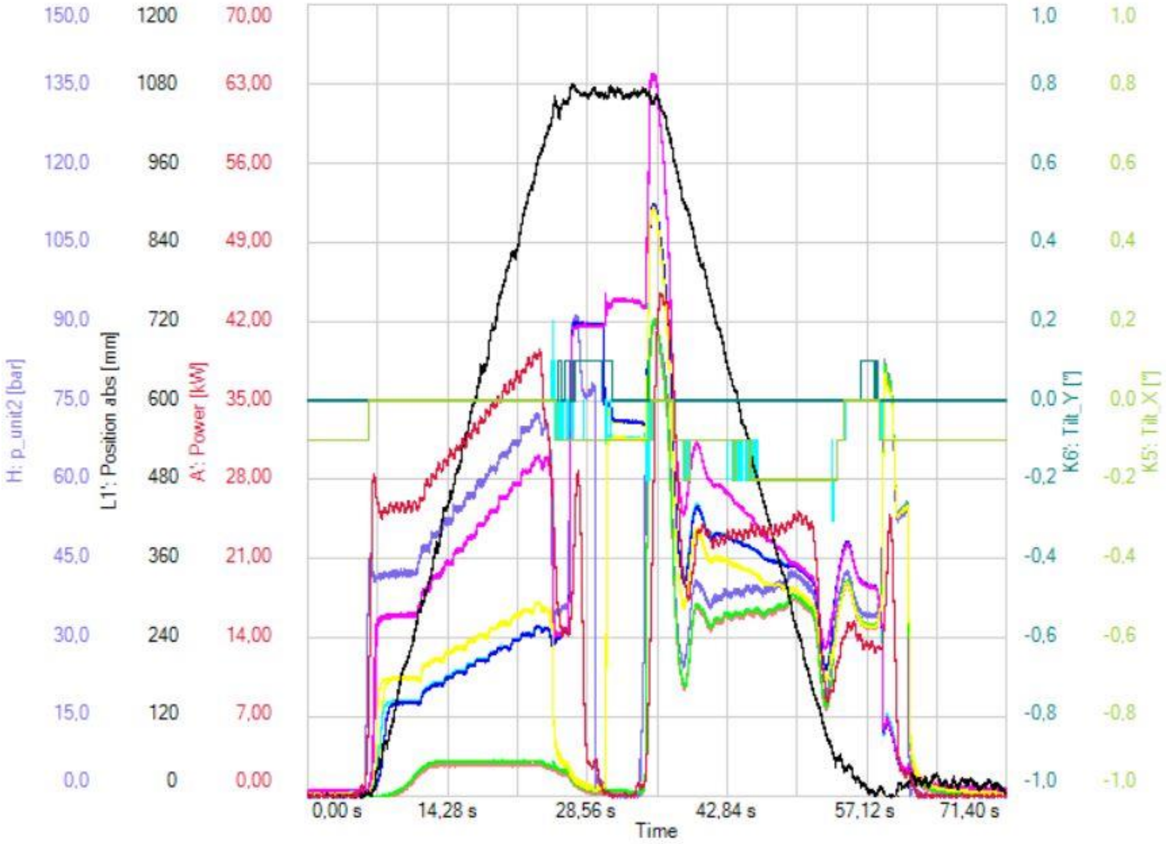


Figure 36. A full test cycle of test assembly 2, test number 7 (60 mm/s).

Table 12. Results of tests 5 – 8 of test assembly 2.

Test assembly 2, tests 5 – 8											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
60	56,1	25,5	38,5	42,4	71,0	0,2	22,6	25,4	36,6	42,5	0,2

A figure and table of the target speed of 60 mm/s are presented in figure 36 and table 12. The colors of the different recordings in the figure are similar as in the previous figures. The results of the table are from the area of constant speed, except for the angle of inclination. Based on the rotating speed of 1800 rpm of the motor, the volume flow was approximately 114 l/min per pump unit when the target speed was 60 mm/s and volumetric efficiency of the pump is assumed to be 0,97. There were two motor-pump units at use during the tests with test assembly two, leading to a volume flow of 228 l/min.

A full test cycle of target speed of 70 mm/s is presented in figure 37 and the summarized results are presented in table 13.

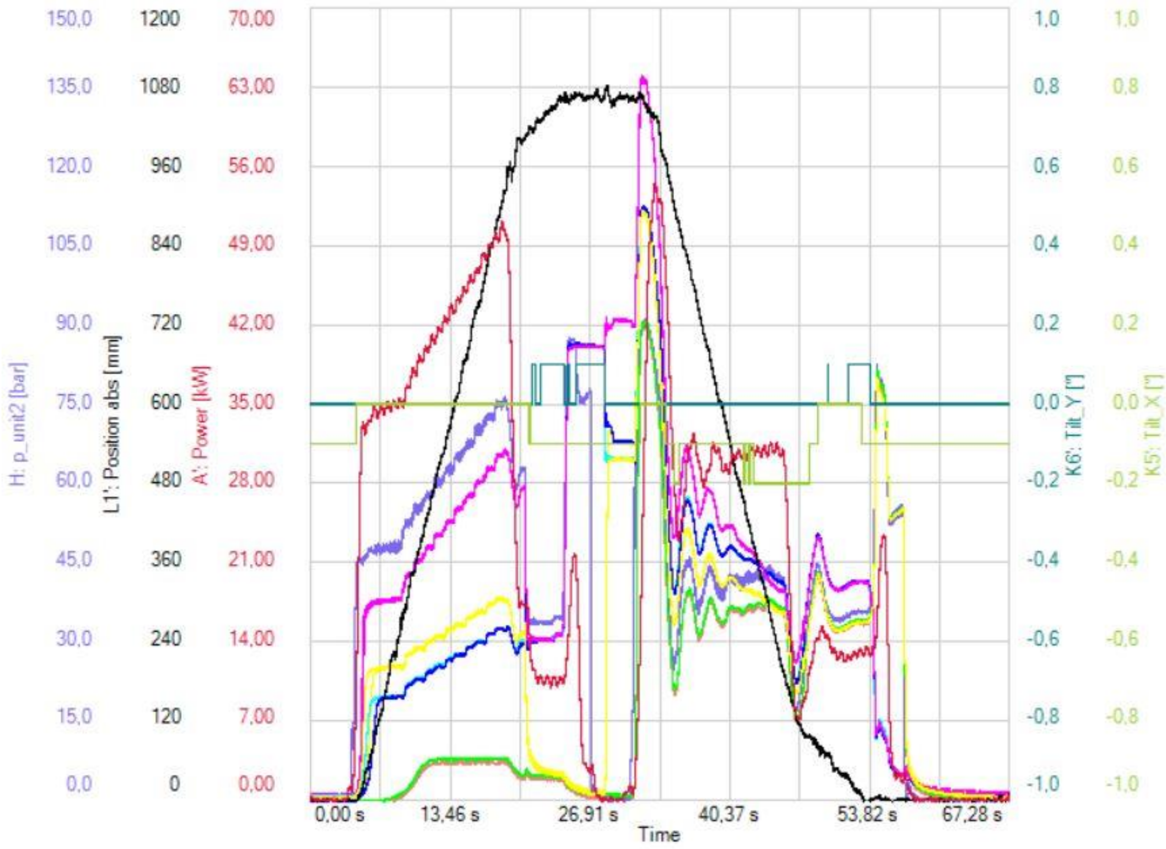


Figure 37. A full test cycle of test assembly 2, test number 10 (70 mm/s).

Table 13. Results of tests 9 – 12 of test assembly 2.

Test assembly 2, tests 9 – 12											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
70	65,7	34,3	49,5	47,3	75,3	0,1	28,8	30,9	38,0	43,9	0,2

A figure and table of the target speed of 70 mm/s are presented in figure 37 and table 13. The colors of the different recordings in the figure are similar as in the previous figures. The results of the table are from the area of constant speed, except for the angle of inclination.

Based on the rotating speed of 2100 rpm of the motor, the volume flow was approximately 133 l/min per pump unit when the target speed was 70 mm/s and volumetric efficiency of the pump is assumed to be 0,97. There were two motor-pump units at use during the tests with test assembly two, leading to a volume flow of 266 l/min.

5.4 Results of the current system

The current system was tested with all of the speeds that were used with the test assemblies. In addition to different speeds, parallel and series connected cylinders were tested. A full test cycle of target speed of 40 mm/s with parallel connected cylinders is presented in figure 38 and the summarized results are presented in table 14.

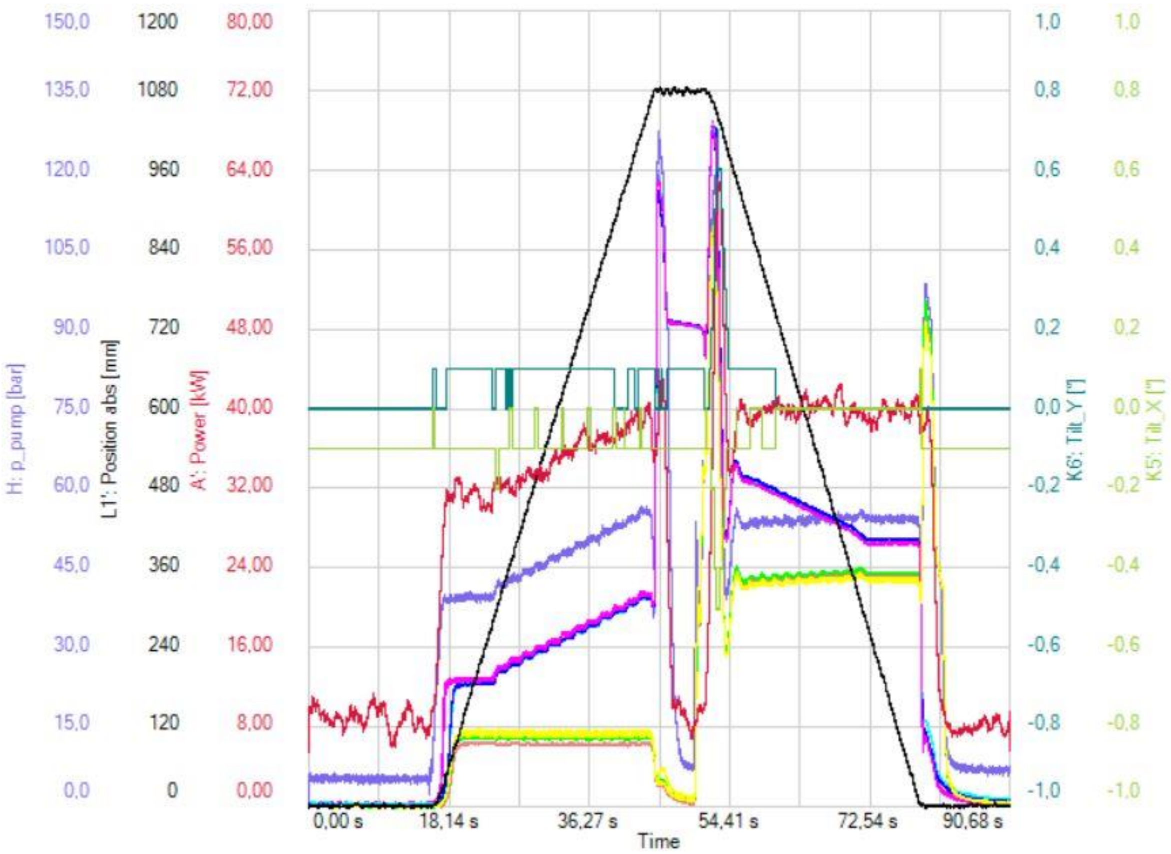


Figure 38. A full test cycle of the current system with parallel connected cylinders, test number 4 (40 mm/s).

Table 14. Results of tests 1 – 4 of the current system with parallel connection.

The current system, parallel connection, tests 1 – 4											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
40	38,2	31,8	40,8	40,0	56,3	0,2	38,8	41,0	53,0	55,0	0,8

A figure and table of the target speed of 40 mm/s are presented in figure 38 and table 14. The colors of the different recordings in the figure are similar as in the previous figures. The results of the table are from the area of constant speed, except for the angle of inclination. The requested volume flow was 310 l/min. Speed of 40 mm/s was the only speed that was tested with parallel connection.

A full test cycle of target speed of 40 mm/s with series connected cylinders is presented in figure 39 and the summarized results are presented in table 15.

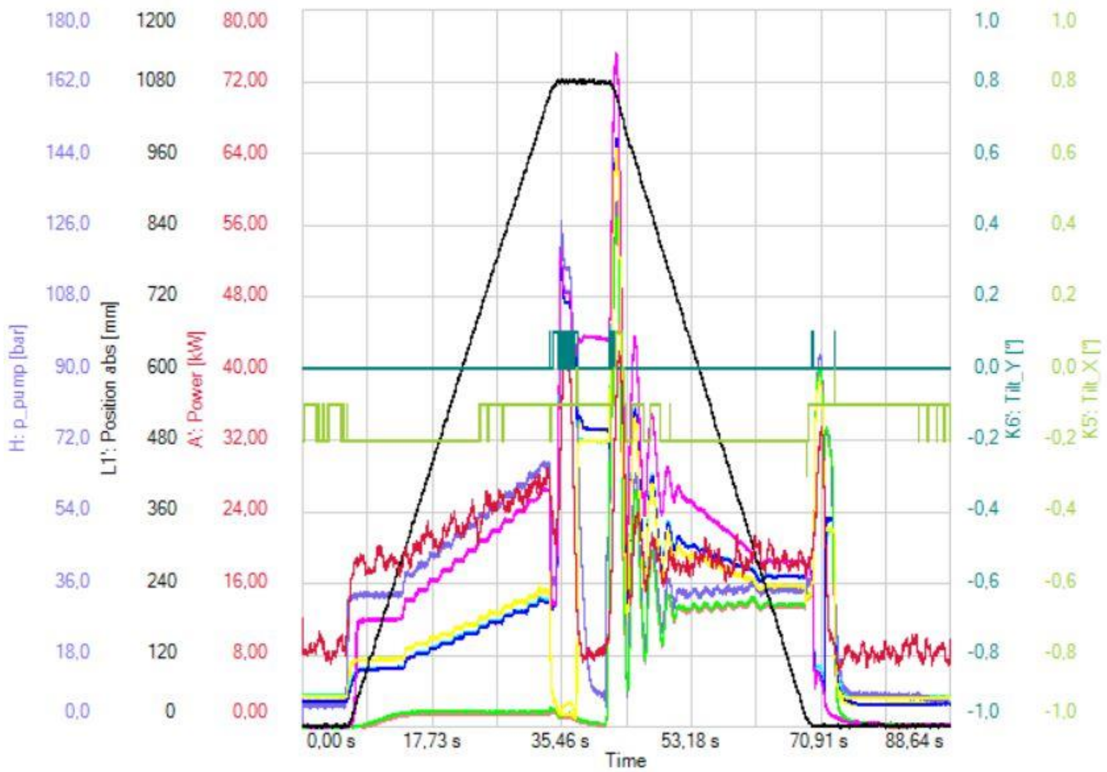


Figure 39. A full test cycle of the current system with series connected cylinders, test number 1 (40 mm/s).

Table 15. Results of tests 1 – 4 of the current system with series connection.

The current system, series connection, tests 1 – 4											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
40	38,5	17,6	27,6	33,0	65,9	0,2	17,3	19,8	31,8	35,8	0,2

A figure and table of the target speed of 40 mm/s are presented in figure 39 and table 15. The colors of the different recordings in the figure are similar as in the previous figures. The results of the table are from the area of constant speed, except for the angle of inclination. The requested volume flow was 165 l/min. With a low volume flow, there are pressure fluctuations visible in the figure, similarly as with test assembly 2.

A full test cycle of target speed of 60 mm/s with series connected cylinders is presented in figure 40 and the summarized results are presented in table 16.

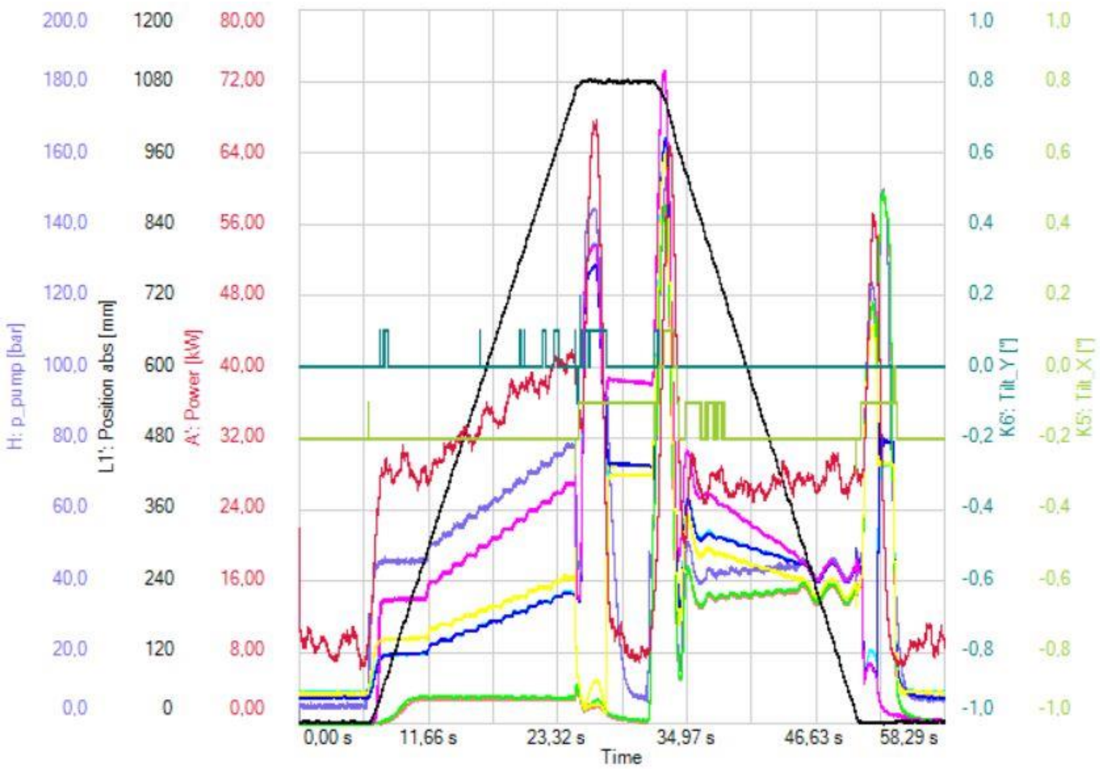


Figure 40. A full test cycle of the current system with series connected cylinders, test number 7 (60 mm/s).

Table 16. Results of tests 5 – 8 of the current system with series connection.

The current system, series connection, tests 5 – 8												
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack					
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
60	58,0	27,5	40,8	45,5	78,0	0,2	25,3	28,8	39,0	45,8	0,3

A figure and table of the target speed of 60 mm/s are presented in figure 40 and table 16. The colors of the different recordings in the figure are similar as in the previous figures. The results of the table are from the area of constant speed, except for the angle of inclination. The requested volume flow was 240 l/min.

A full test cycle of target speed of 70 mm/s with series connected cylinders is presented in figure 41 and the summarized results are presented in table 17.

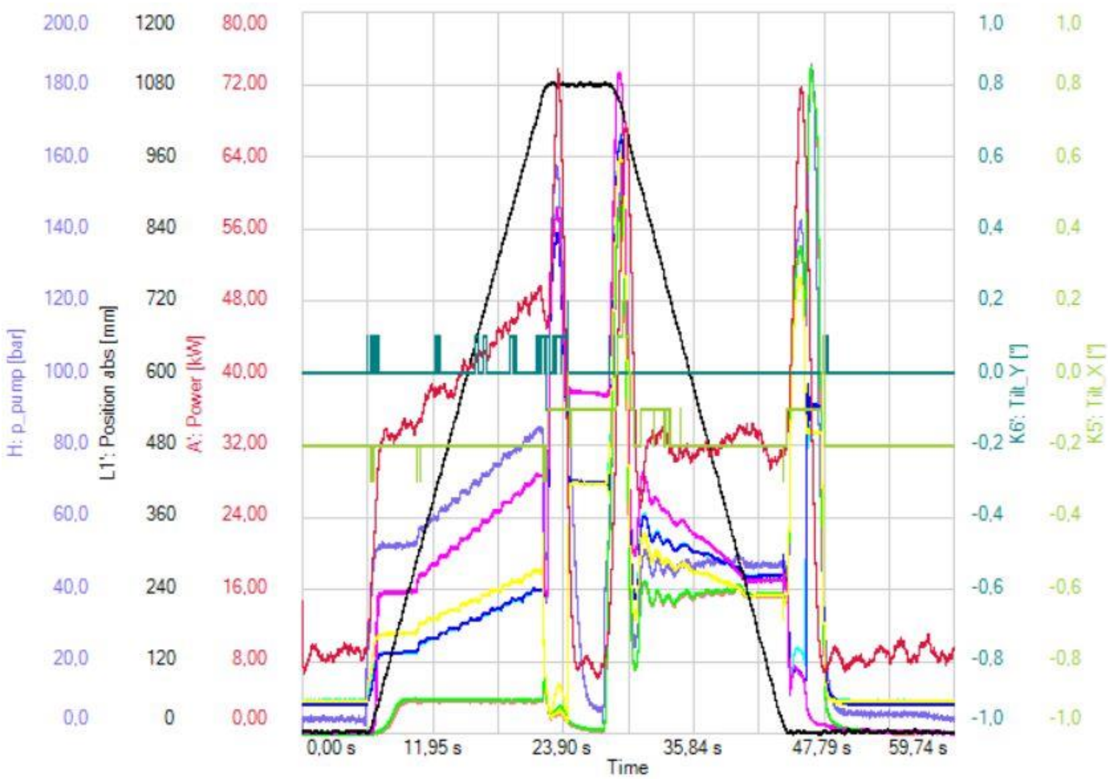


Figure 41. A full test cycle of the current system with series connected cylinders, test number 11 (70 mm/s).

Table 17. Results of tests 9 – 12 of the current system with series connection.

The current system, series connection, tests 9 – 12												
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack					
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
70	67,0	33,0	49,5	52,0	85,0	0,3	31,0	33,8	44,8	49,9	0,4

A figure and table of the target speed of 70 mm/s are presented in figure 41 and table 17. The colors of the different recordings in the figure are similar as in the previous figures. The results of the table are from the area of constant speed, except for the angle of inclination. The requested volume flow was 275 l/min.

A full test cycle of target speed of 80 mm/s with series connected cylinders is presented in figure 42 and the summarized results are presented in table 18.

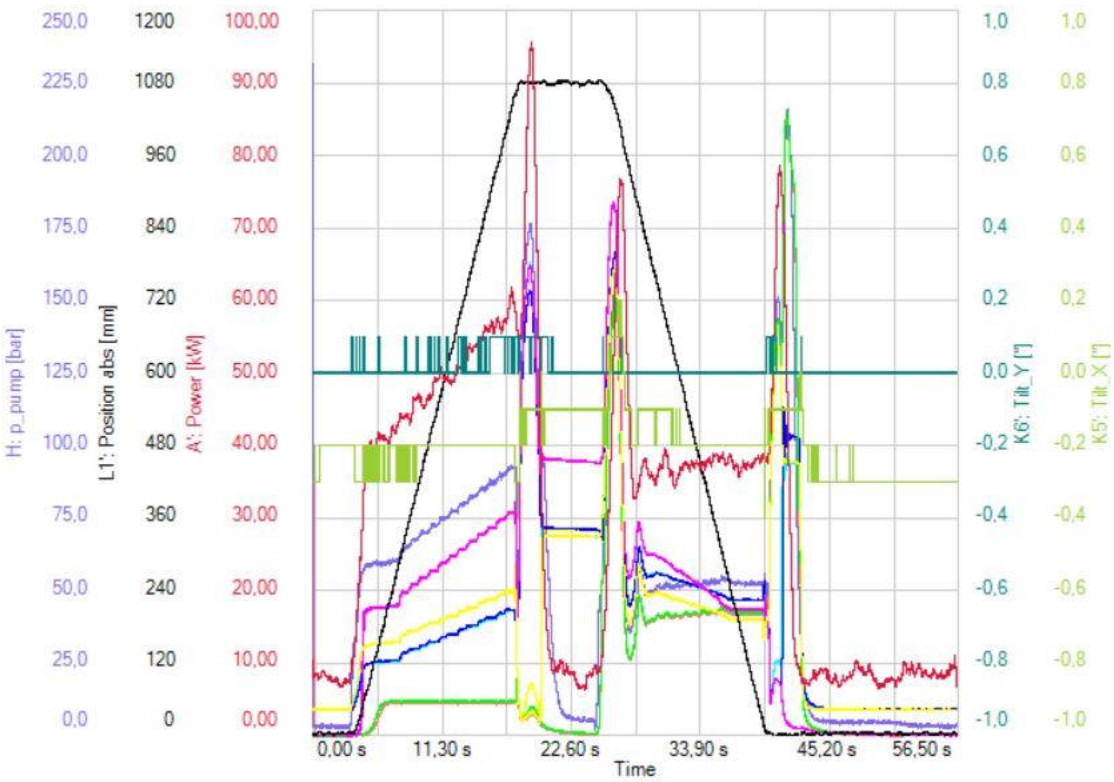


Figure 42. A full test cycle of the current system with series connected cylinders, test number 13 (80 mm/s).

Table 18. Results of tests 13 – 16 of the current system with series connection.

The current system, series connection, tests 13 – 16											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
80	76,0	40,5	59,5	59,0	91,9	0,3	36,0	39,0	49,8	54,1	0,4

A figure and table of the target speed of 80 mm/s are presented in figure 42 and table 18. The colors of the different recordings in the figure are similar as in the previous figures. The results of the table are from the area of constant speed, except for the angle of inclination. The requested volume flow was 305 l/min.

A full test cycle of target speed of 85 mm/s with series connected cylinders is presented in figure 43 and the summarized results are presented in table 19.

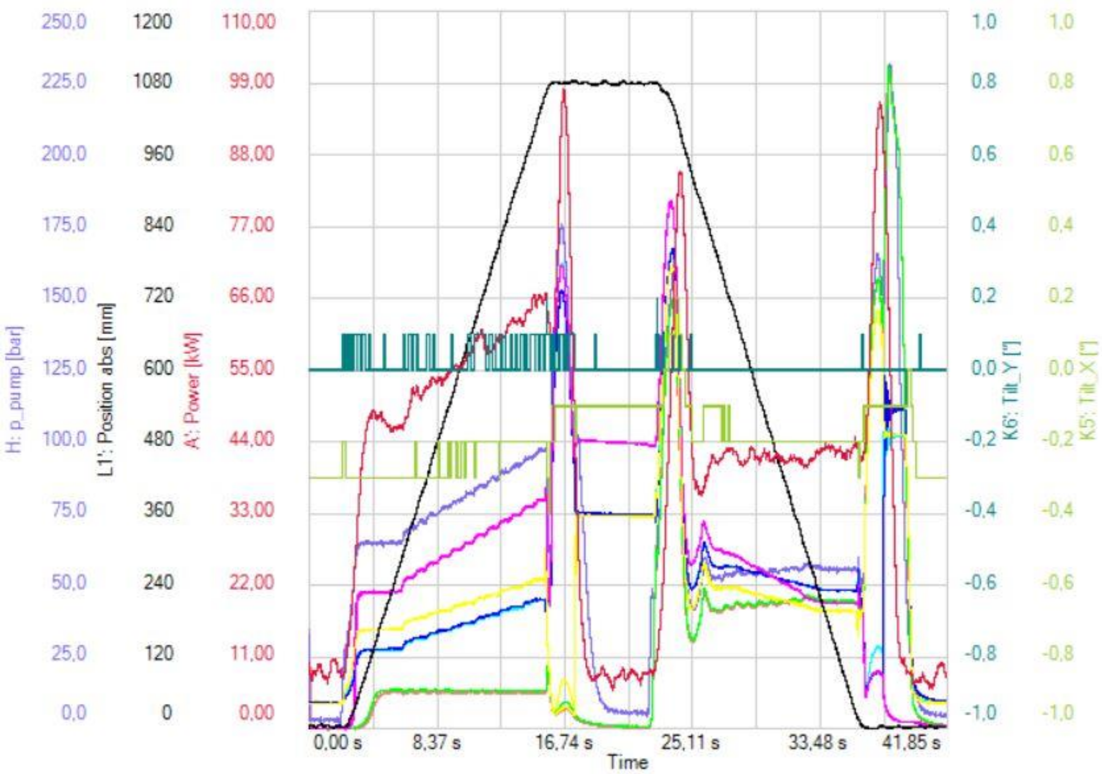


Figure 43. A full test cycle of the current system with series connected cylinders, test number 20 (85 mm/s).

Table 19. Results of tests 17 – 20 of the current system with series connection.

The current system, series connection, tests 17 – 20											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
85	81,8	46,8	65,8	65,3	97,3	0,4	40,8	44,0	53,3	57,3	0,6

A figure and table of the target speed of 85 mm/s are presented in figure 43 and table 19. The colors of the different recordings in the figure are similar as in the previous figures. The results of the table are from the area of constant speed, except for the angle of inclination. The requested volume flow was 330 l/min.

5.5 Comparison between the test assemblies and the current system

Test assembly one is the most comparable with the current system when the current system has cylinders in parallel connection. The pressures are on the same level and volume flow is produced to each cylinder. However, parallel connection is not the standard connection that is used for opening and closing the plate pack and the reachable volume flow is not sufficient for a speed of 80 mm/s. Therefore, there is only one speed to compare. A comparison of test assembly one and the current system with parallel connection is presented in table 20.

Table 20. Comparison between test assembly 1 and the current system with parallel connection.

Test assembly 1 and the current system with parallel connection											
name of the test	calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
test assembly 1	39,1	19,6	28,4	21,2	35,1	0,1	16,5	18,7	16,2	20,8	0,1
current system, parallel cylinders	38,2	31,8	40,8	40,0	56,3	0,2	38,8	41,0	53,0	55,0	0,8

The differences presented in table 20 are significant. When opening the plate pack, reduction of active power with test assembly 1 is between 30 to 38 % compared to the current system. During closing of the plate pack, reduction of active power is between 54 to 57 %. In addition, there is a difference in the angle of inclination during the cycles. The maximum variation of the angle of inclination with test assembly 1 is 0,1° whereas the maximum variation with the current system with parallel connected cylinders is 0,8°.

Test assembly 2 utilizes the QAC valve block of the current system. Because of the similarity with the current system, it can be compared with the series connection of the current system. A comparison between the tests is presented in table 21.

Table 21. Comparison between test assembly 2 and the current system with series connection.

Test assembly 2 and the current system with series connection											
name of the test	calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
test assembly 2	38,7	14,0	22,1	33,9	62,1	0,2	12,4	15,9	29,8	38,3	0,2
current system, series connection	38,5	17,6	27,6	33,0	65,9	0,2	17,3	19,8	31,8	35,8	0,2
test assembly 2	56,1	25,5	38,5	42,4	71,0	0,2	22,6	25,4	36,6	42,5	0,2
current system, series connection	58,0	27,5	40,8	45,5	78,0	0,2	25,3	28,8	39,0	45,8	0,3
test assembly 2	65,7	34,3	49,5	47,3	75,3	0,1	28,8	30,9	38,0	43,9	0,2
current system, series connection	67,0	33,0	49,5	52,0	85,0	0,3	31,0	33,8	44,8	49,9	0,4

A comparison presented in table 21 indicates that there is only a slight difference between the assemblies. The measured active powers are almost the same in both test assembly 2 and the current system with series connection. The pressure levels are a bit lower with test assembly 2 than the current system. When the target speed was 40 mm/s, both of the systems reached almost the same speed. However, with the target speeds of 60 mm/s and 70 mm/s, test assembly 2 had about 2 – 3 % lower speed compared to the current system. That is why there is no remarkable difference between the active powers.

Test assembly 1 was the connection of the test hydraulic system that was designed to replace the current hydraulic system. In general, the current hydraulic system is operated with cylinders in series connection. Therefore, it is required to compare test assembly 1 with the current system in series connection. The comparison of these tests is presented in table 22.

Table 22. Comparison between test assembly 1 and the current system with series connection.

Test assembly 1 and the current system with series connection											
name of the test	calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
test assembly 1	39,1	19,6	28,4	21,2	35,1	0,1	16,5	18,7	16,2	20,8	0,1
current system, series connection	38,5	17,6	27,6	33,0	65,9	0,2	17,3	19,8	31,8	35,8	0,2
test assembly 1	57,3	36,0	48,9	26,6	40,3	0,1	30,5	34,9	23,5	26,3	0,1
current system, series connection	58,0	27,5	40,8	45,5	78,0	0,2	25,3	28,8	39,0	45,8	0,3
test assembly 1	75,9	63,5	81,5	31,7	46,4	0,1	52,4	58,1	27,5	31,8	0,2
current system, series connection	76,0	40,5	59,5	59,0	91,9	0,3	36,0	39,0	49,8	54,1	0,4
test assembly 1	79,6	71,8	90,8	34,1	47,3	0,2	59,4	65,6	27,3	32,8	0,2
current system, series connection	81,8	46,8	65,8	65,3	97,3	0,4	40,8	44,0	53,3	57,3	0,6

As presented in table 22, active power is higher with test assembly 1 in all the tested speeds except the target speed of 40 mm/s, in which the measured active powers are almost the same with both systems. With target speeds of 60, 80 and 85 mm/s active power is up to 38 % higher during opening of the plate pack compared to the current system with series connection. The difference in active power is more significant during closing of the plate pack, because test assembly 1 has up to 49 % higher consumption of active power compared to the current system during closing of the plate pack. However, test assembly 1 seems to be more stable in terms of the angle of inclination of the plate pack compared to the current system. It is noticeable that the difference in angle of inclination increases when the volume flow is higher. The largest variation from the initial level in test assembly 1 is 0,2 degrees, but for the current system it is 0,6 degrees.

6 CONCLUSIONS

The results are further analyzed in this chapter. The factors that affected the performance of the test system are listed and answers are provided for the research questions. Finally, suggestions for future research are made.

6.1 Observations of hydraulic test system

The test hydraulic unit has internal gear pumps that have a rated maximum pressure of 250 bar. However, in these tests, the maximum pressures in test assembly 1 were 47 bar and in test assembly 2 75 bar. Because of the low pressure levels the hydromechanical efficiency of the pumps was not on an optimal level. After the tests it became clear that the pressure and volume flow diagram that was provided by Hydac for the motor-pump units was not accurate for the pressure levels that were used in the tests. The initial diagram was calculated with hydromechanical efficiency of 0,9. An efficiency diagram for a similar pump (PGI103-5-064) that was used in these tests is presented in figure 44.

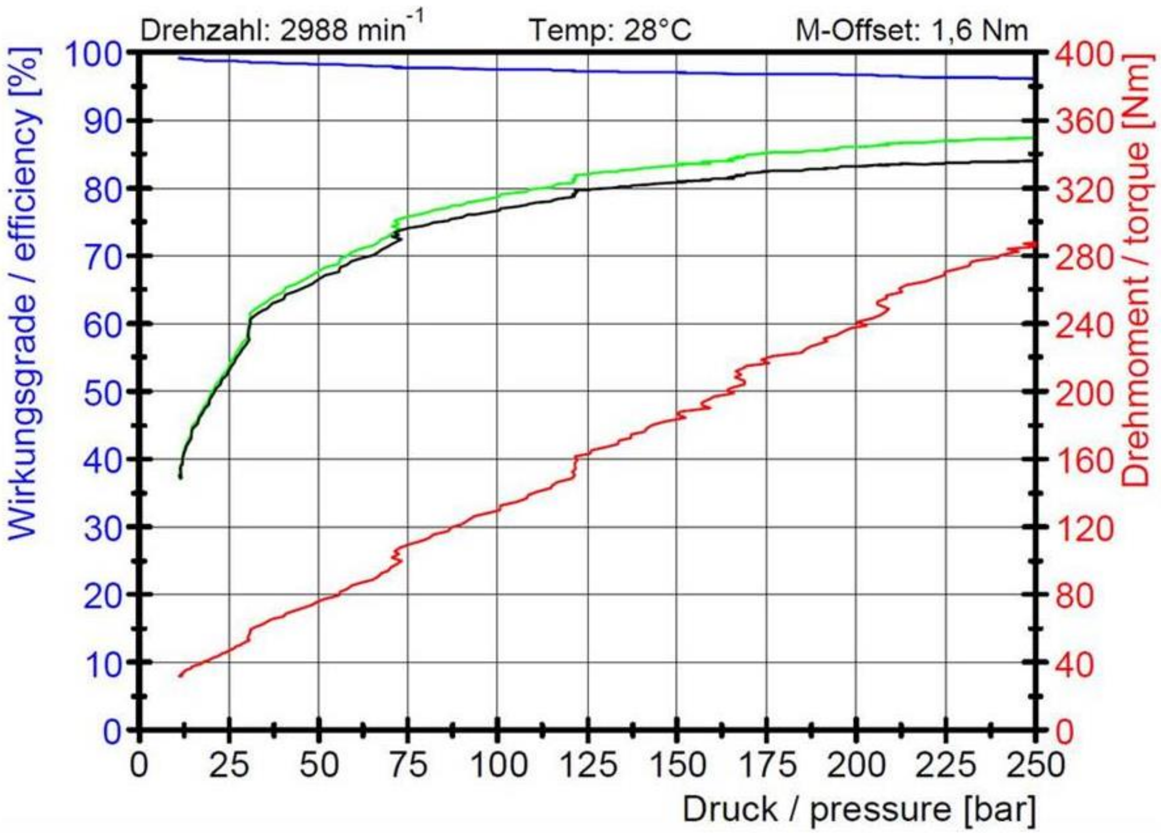


Figure 44. Efficiency diagram for PGI103-5-064 pump (Hydac 2019d).

In efficiency diagram that is presented in figure 44, the blue line indicates volumetric efficiency. The black and green lines present hydromechanical efficiency and the red line is for torque. With a maximum pressure range of 34 – 47 bar (in test assembly 1), volumetric efficiency is about 0,98 and hydromechanical efficiency is about 0,63 – 0,65. For a pressure range of 47 – 75 bar (in test assembly 2) volumetric efficiency is about 0,97 and hydromechanical efficiency is about 0,65 – 0,75. The low hydromechanical efficiencies explain why the measured active power levels were high for the test system.

Hydac provided a new pressure and volume flow diagram for the motor-pump units. The diagram is presented in figure 45.

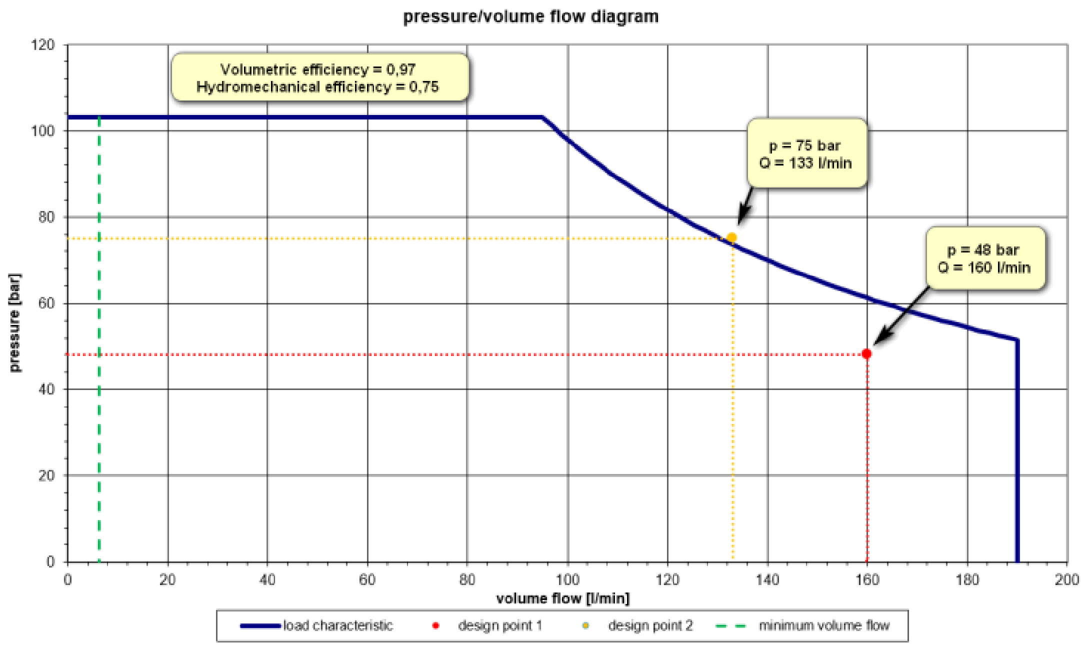


Figure 45. Pressure and volume flow diagram for the motor-pump units (Hydac 2019d).

The pressure and volume flow diagram presented in figure 45 has volumetric efficiency of 0,97 and hydromechanical efficiency of 0,75 (the blue load characteristics curve). The red point presents the maximum pressure and volume flow in tests of test assembly 1. The point is not on the curve because the hydromechanical efficiency is less than 0,75 with that pressure. On the other hand, the yellow point presents the maximum pressure and volume flow in tests of test assembly 2. The point is almost on the curve because hydromechanical efficiency of 0,75 is accurate for that pressure.

High measured active power for the hydraulic test unit is caused by the poor efficiency of the used pumps on a low pressure level. The measured active power values would have been more competitive if the efficiency of the pumps was 0,90. A table with measured values of test assembly 1 transferred to hydromechanical efficiency of 0,90 is presented in table 23. The table also includes the measured values for the current system in series connection.

Table 23. Test assembly 1 with improved hydromechanical efficiency of the pumps compared to the current system in series connection.

Test assembly 1 with hydromechanical efficiency of 0,90 and the current system with series connection					
name of the test	calculated speed (mm/s)	opening the plate pack		closing the plate pack	
		active power (kW)		active power (kW)	
		min	max	min	max
test assembly 1	39,1	13,7	19,9	11,6	13,1
current system, series connection	38,5	17,6	27,6	17,3	19,8
test assembly 1	57,3	26,0	35,3	22,0	25,2
current system, series connection	58,0	27,5	40,8	25,3	28,8
test assembly 1	75,9	45,9	58,9	37,8	42,0
current system, series connection	76,0	40,5	59,5	36,0	39,0
test assembly 1	79,6	51,9	65,6	42,9	47,4
current system, series connection	81,8	46,8	65,8	40,8	44,0

With hydromechanical efficiency of the pumps improved to 0,90 (presented in table 23), test assembly 1 would have quite similar active power measurements as the current system. The values are calculated so that all other parameters are assumed to be the same, only hydromechanical efficiency is improved. In reality, it would be difficult to find a hydraulic pump with hydromechanical efficiency of 0,90 at pressure less than 50 bar. Hydac gear pumps can have hydromechanical efficiency of approximately 0,77 at 50 bar, whereas Hydac vane pumps can reach hydromechanical efficiency of approximately 0,83 at 50 bar (Hydac 2019e; Fixed Displacement Vane Pumps 2014, p. 6 – 10).

An alternative for replacing the pumps of the hydraulic test unit would be to increase the pressure level of the system. By increasing the pressure level, the pumps could operate on an optimal pressure range. This requires reducing the working area of the cylinders. A simple way of increasing the pressure level is to increase the piston rod size to reduce the working area. Another method is to reduce the piston size and find an optimal working area with piston and piston rod for the required pressure range. If the size of the cylinders is decreased, stability of the cylinders must be re-examined. Reducing the size of the cylinders would also provide other benefits, such as less required volume flow to reach the same cylinder speed. Less volume flow would also mean less losses in the system and possibility to reduce hydraulic pipe sizes.

6.2 Answers to research questions

The main research question of this thesis is “Does the decentralized hydraulic system with utilization of speed-controlled hydraulic pumps change PF filters into more energy efficient?”. Based on the results of this study, the new decentralized hydraulic system has potential for energy savings and accuracy of control. However, test assembly 1 consumed more power with the plate pack movement target speed of 80 mm/s than the current hydraulic system in series connection. The higher consumption of electricity is because of the low hydromechanical efficiency of the internal gear pumps in low pressure range. Despite the low efficiency of test assembly 1 in that pressure level, active power of test assembly 1 was 30 – 38 % lower during plate pack opening and 54 – 57 % lower during plate pack closing compared to the current system in parallel connection. Parallel connection, however, is not normally used during the whole stroke of the cylinder.

In addition to the main research question, there are three subsidiary research questions in this study. The first one is “Is it possible to design the hydraulic circuit of opening and closing the plate pack so that the cylinders are always synchronized and operation is accurate?”. The accuracy refers to the angle of inclination of the top pressure plate (and the whole plate pack), in which the test system performed better than the current system during the tests. The cylinders were well-synchronized in each test drive with the hydraulic test system, because the maximum difference in angle of inclination was 0,2 degrees with test assemblies 1 and 2. The current system had slightly more variation as the maximum difference in angle of inclination was 0,8 degrees with parallel connection and 0,6 degrees with series connection. Contrary to the current system, the hydraulic test system has potential to adjust the rotating speed of each pump separately to compensate the possible leakages of the hydraulic cylinders. This feature would eliminate the problems that can cause if the plate pack is inclined. In addition to keeping the plate pack straight, the system could collect data of the inclination and eventually predict maintenance operations required for the cylinders. The possibility to adjust the rotating speed of each pump separately at the same time was not included in this test system, because the system was tested on a new PF60 filter with no problems with the condition of the cylinders.

The second subsidiary research question is “Can the number of different spare parts in hydraulics be reduced?”. The hydraulic test system simplifies the spare part selection as the system has four identical motor-pump units and valve assemblies. For the end-user of the PF60 filter it would mean that they can have components for one motor-pump unit in warehouse and if there is failure in any of the four units, they have the correct components for a rapid repair. In addition, the aim was to use components that have short delivery times and that are less special than some components in the current system, such as the variable-displacement pump.

The third subsidiary research question is “Is troubleshooting during maintenance operations faster compared to the current system?”. The advantage of the hydraulic test system is that it is simple to notice in which of the units the fault is in, because each of the units drive their own cylinders. This is not the case with the current system, as there is only one hydraulic unit and various valve blocks. Troubleshooting can be very time-consuming and sometimes it is difficult to know in which of the valves or cylinders the fault is in. With a decentralized system like the test system, it is more simple to exclude the components during troubleshooting.

6.3 Further development and research

The decentralized system for PF60 filter was partially tested in this thesis. There is still need for further development and research, before the system can be commissioned in a production model filter. Major target for development is increasing the pressure level of the system. By increasing the pressure level, the pumps can operate on an optimal efficiency and consumption of electricity is reduced. Size of the pumps can also be reduced, because less volume flow is required to reach the same speed of the cylinders. When increasing the pressure level by reducing the size of the cylinders, stability of the cylinders must be re-examined.

The size of the hydraulic oil tank in this hydraulic test unit was 630 liters. That size is rather large if the hydraulic unit is placed on top of the pressure filter or integrated to the structures of the filter. Tank size can be reduced by using a technology developed by Hydac called OxiStop. The technology reduces the amount of air in hydraulic oil and size of the tank can reduce by hundreds of liters. The size of the oil tank is determined based on the differential operating volume of the hydraulic system. In case of the test system in this thesis, differential operating volume is almost zero as the cylinders have double-rods. Based on this, the total volume of the tank with OxiStop technology would be 110 liters. (OxiStop OXS, 2016.) However, future research is required to find the possible advantages of OxiStop technology for PF filters.

One interesting development trend is to change the decentralized hydraulic system into closed-loop system. This would require further research, but is something to consider when making decisions about the future of hydraulic system in PF filters. A closed-loop system can reduce energy consumption in many cases, and it can provide additional benefits for the hydraulic operations.

Remote condition monitoring could increase the intelligence of PF filters and variable speed drive controlled hydraulic system can enable it. By including an inclinometer to the system with feedback control, it is possible to predict changes in the condition of the hydraulic cylinders. PF filter could then remind of upcoming maintenance tasks and provide an online tool for ordering the correct spare parts. The customer could receive information of the condition of the hydraulic system and plan maintenance operations in advance according to the requirements of the system’s condition.

7 SUMMARY

This master's thesis discussed an improvement to a hydraulic system of a stationary industrial machine. The machine is a Outotec Larox PF60 filter that is used in the mining industry, for instance. The modifications to the system were done by utilization of rotational speed-controlled fixed displacement pumps. The current hydraulic system of PF60 was divided into sub-systems on a concept level. The focus in this thesis was on opening and closing movement of the plate pack. Movements of the plate pack are done with four double-acting double-rod hydraulic cylinders. A test system was constructed for these cylinders to conduct tests with the PF60 filter.

The basis for the test system was that minimum modifications can be done to the structure of the PF60 filter, as the test filter is a production model filter. Objective was to find ways to improve energy efficiency and synchronization of the cylinders. In addition, spare part availability and maintenance aspects were part of the research questions. The test system included a separate motor-pump unit for each cylinder. The four motor-pump units were mounted on a same oil tank. The test unit was mounted on top of the PF60 filter, because hydraulic connections between the motor-pump units and the cylinders wanted to be kept in a minimum.

The tests included two different test connections with the hydraulic test unit. Active power was measured during the test drives in addition to cylinder pressures, pump pressure and inclination angle of the plate pack. An inclinometer was mounted on top of the topmost part of the plate pack to measure the synchronization of the cylinders. When the measurements with the test system were done, the same measurements were repeated with the current hydraulic system of PF60 for comparison.

Based on the results the test system was more accurate in synchronization of the four cylinders than the current hydraulic system. The difference in angle of inclination was maximum 0,2 degrees with the two different test connections. The same inclination angle was maximum 0,8 degrees for the current system. Three comparisons were done between the two connections of the test system and the two connections of the current system. The test system reduced power consumption in only one of the three comparisons. Reason for this was that the pressure level was significantly lower than the optimal pressure level for the pumps. The hydromechanical efficiency of the pumps increased losses inside the pumps and caused the high measured active power levels.

It was suggested that the pressure level is increased to improve the efficiency of the system before the test system is further developed into actual hydraulic system of PF60 filter. A simple way of increasing the pressure is to reduce the piston area of the cylinder. If the cylinders are modified, the stability of the cylinders must be analyzed again. The test system can be modified so that the system includes feedback control from the inclinometer to adjust the rotating speed of each motor separately. This is important as there is a need to maintain the synchronization of the cylinders at all times, even if the cylinders are not new and fully functioning.

Additional benefits of the test system included improved spare part availability and straightforward troubleshooting. In addition to increasing the pressure level and including the feedback control, reducing the oil tank size is important for further development. Changing the system into closed-loop system is another possible direction of development. If a decentralized hydraulic system is implemented to PF60 filters, remote condition monitoring of the hydraulic system can be developed.

LIST OF REFERENCES

Altare, G. & Vacca, A. 2015. A design solution for efficient and compact electro-hydraulic actuators. Procedia Engineering. No. 106, p. 8 – 16.

Dieckmann, J., McKenney, K. & Brodrick, J. 2010. Variable Frequency Drives, Part 1: The Technology. ASHRAE Journal. No. 52(4), p. 60 – 62.

Ekman, A. & Börjesson, P. 2011. Life cycle assessment of mineral oil-based and vegetable oil-based hydraulic fluids including comparison of biocatalytic and conventional production methods. The International Journal of Life Cycle Assessment. No. 16(4), p. 297 – 305.

Fixed Displacement Vane Pump. Datasheet. (Publishing place unknown): Hydac, 2014. 14 p.

Hydac Systems & Services. 2019a. Hydraulic diagram of the DVA kit [private e-mail]. Confidential document.

Hydac Systems & Services. 2019b. Proposal letter [private e-mail]. Confidential document.

Hydac Oy. 2019c. Measurement plan [private e-mail]. Confidential document.

Hydac Oy. 2019d. Measuring report [private e-mail]. Confidential document.

Hydac Oy. 2019e. Efficiency of gear pumps [private e-mail]. Confidential document.

Jalayeri, E., Imam, A., Tomas, Z. & Sepehri, N. 2015. A throttle-less single-rod hydraulic cylinder positioning system: Design and experimental evaluation. Advances in Mechanical Engineering. No. 7(5), p. 1 – 14.

Kaipainen, J. 2018. Product development manager, PF filters. Interview. 18.7.2018.

Kauranne, H., Kajaste, J., & Vilenius, M. 2013. Hydraulitekniikka. 2nd edition. Helsinki: Sanoma Pro Oy. 496 p.

Koitto, T., Kauranne, H., Calonius, O., Minav, T. & Pietola, M. 2018. Experimental Investigation of a Directly Driven Hydraulic Unit in an Industrial Application. In: H. Murrenhoff (ed.), Proceedings of the 11th International Fluid Power Conference, March 19 – 21, 2018, Aachen, Germany. Vol 2, p. 348 – 361.

Lovrec, D. & Ulaga, S. 2007. Pressure control in hydraulic systems with variable or constant pumps? Experimental Techniques. No. 31(2), p. 33 – 41.

Maré, J.-C. & Fu, J. 2017. Review on signal-by-wire and power-by-wire actuation for more electric aircraft. Chinese Journal of Aeronautics. No 30(3), p. 857 – 870.

OxiStop OXS. Datasheet. (Publishing place unknown): Hydac, 2016. 4 p.

Quan, Z., Quan, L. & Zhang, J. 2014. Review of energy efficient direct pump controlled cylinder electro-hydraulic technology. Renewable and Sustainable Energy Reviews. No. 35(C), p. 336 – 346.

Tan, K. K. & Putra, A. S. 2011. Drives and Control for Industrial Automation. London: Springer London. 182 p.

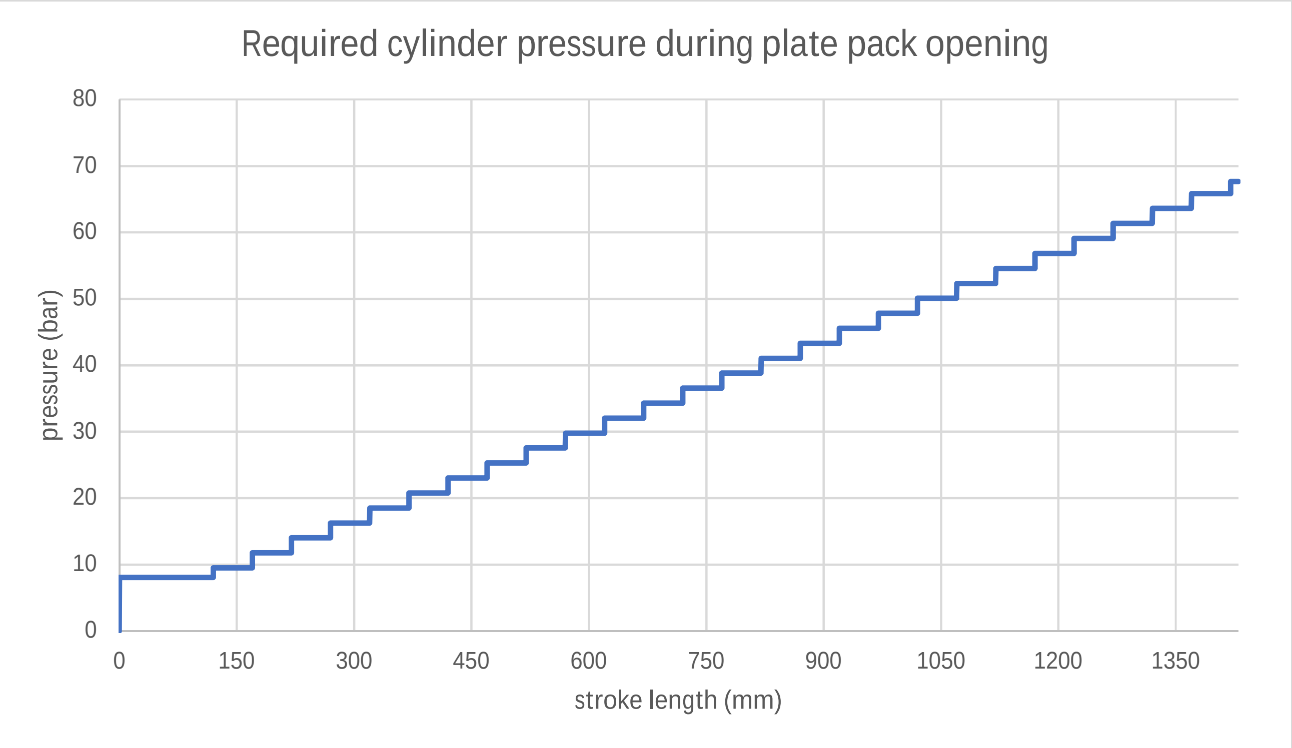
Initial data for QAC circuit
standard model with slurry
PF 156/156 M60 1 60

Cylinder data (mm)	
piston	250
rod	140
stroke	1430

Part	Weight (kg)
top pressure plate	10000
filter plate, topmost	1765
filter plate	1706
filter plate with aux. drive	1708
filter plate, lowest	1143
slurry / chamber	1080
total (156/156)	83650
(each cylinder)	20912,5

velocity	0,08	m/s
volumetric efficiency	0,99	
hydromechanical efficiency	0,9	
total efficiency	0,891	
area	0,0336936	m2

volume flow	163,4	l/min	max. pressure	67,7	bar
max. hydraulic power	18419,9	W			
plate pack opening time	17,9	s			



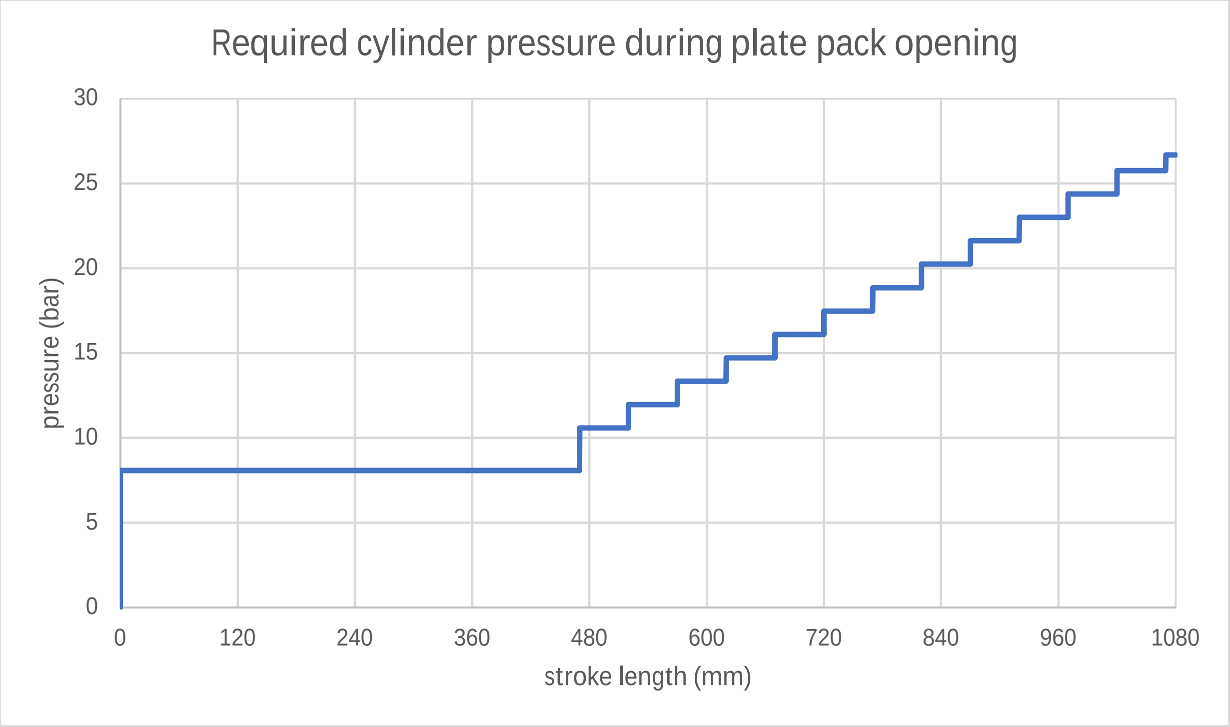
Initial data for QAC circuit test assembly (without slurry) PF 72/96 M60 1 60

Cylinder data (mm)	
piston	250
rod	140
stroke	1080

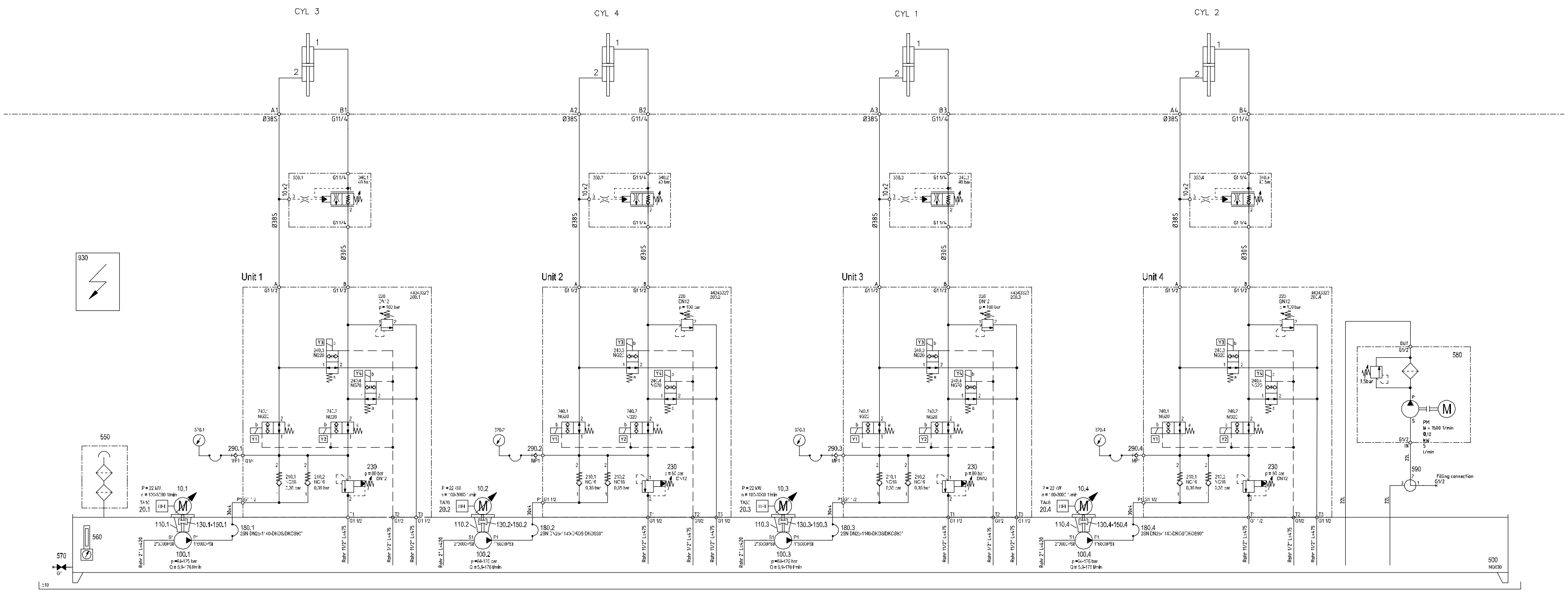
Part	Weight (kg)	
top pressure plate	10000	
filter plate, topmost	3083	(includes filling plates)
filter plate	1706	
filter plate with aux. drive	1708	
filter plate, lowest	1143	
total (72/96)	33000	
(each cylinder)	8250	

velocity	0,08	m/s
volumetric efficiency	0,99	
hydromechanical efficiency	0,9	
total efficiency	0,891	
area	0,0336936	m2

volume flow	163,4	l/min	max. pressure	26,7	bar
max. hydraulic power	7266,7	W			
plate pack opening time	13,5	s			



APPENDIX III

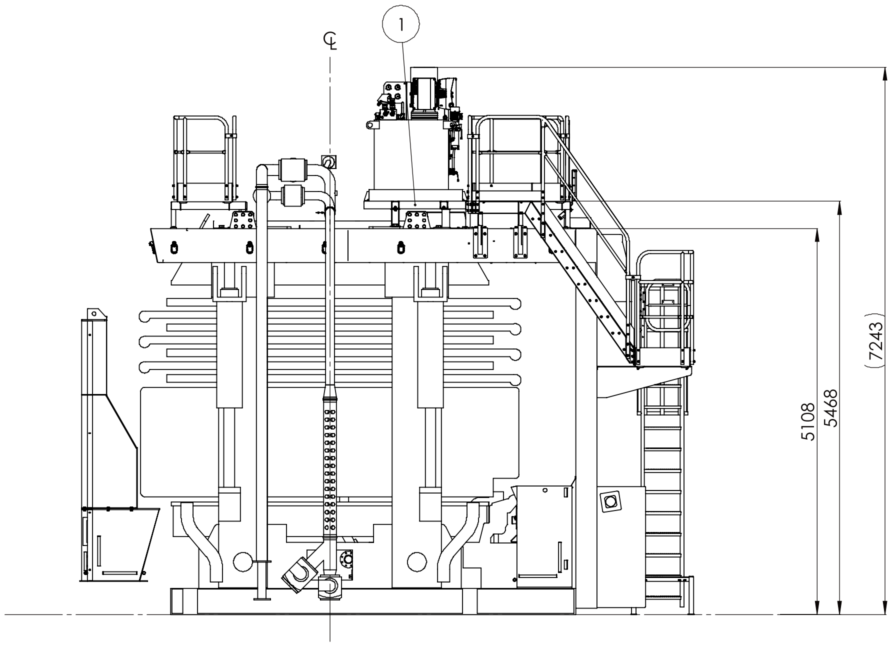


Dimensional tolerances	Machining EN 22768-1-m Thermal cutting EN ISO 9013-1	Welded constr. EN ISO 13920-B Welding Quality Level EN 5817 C
Outotec	Description HYDRAULIC DIAGRAM PF60 TEST ASSEMBLY 1	Weight (kg) Drawn 03.07.2019 ANESAI Check Scale / Size 1:1 A0 Drawing.No F733443 Revision A

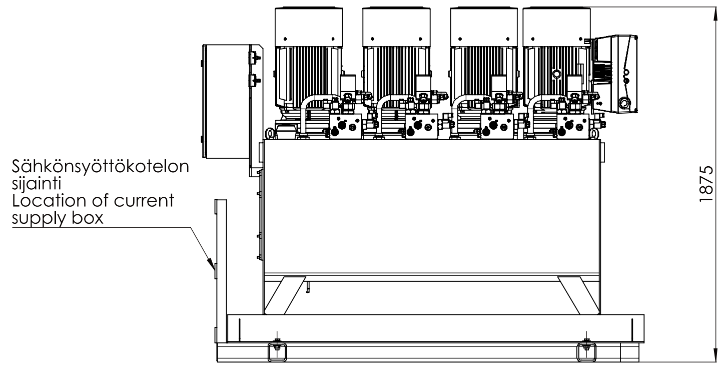
Sheet 1/ 1

1122045.dwg

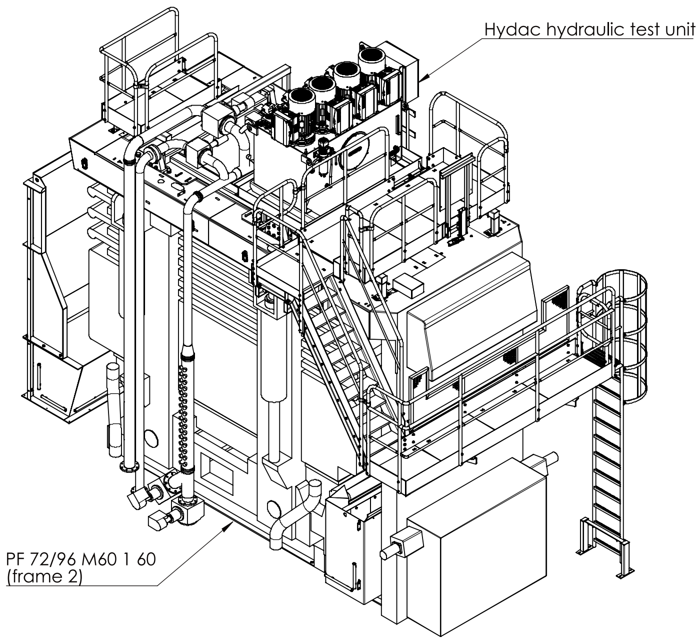
Rev	Info
A	
B	Structure updated.



Hydraulic test unit with support frame
scale 1:20



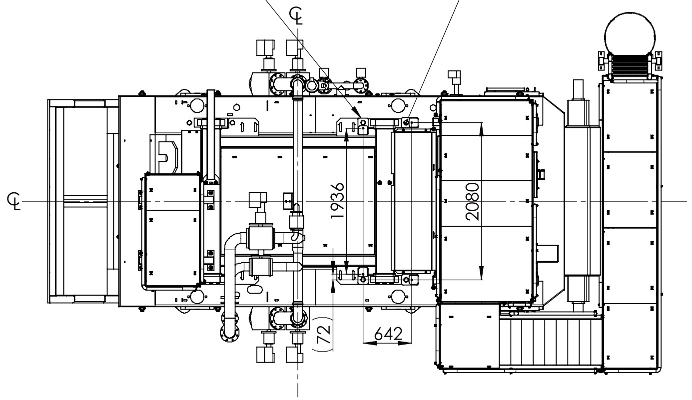
Asenna lattiatasossa testihydrauliyksikkö tukirakenteen päälle. Huomaa, että tukirakenteen jalat ovat irrallaan, koska jalat asennetaan ensin suodattimen ylärungon päälle.
Test hydraulic unit is installed on top of the frame on ground level.
Note that the supports of the frame are first installed to upper body.



Asenna ensin testihydrauliyksikön tukirakenteen jalat ylärungon päälle ja tarkista reikien etäisyydet.
Lopuksi nosta testihydrauliyksikkö tukirakenteen kanssa jalkojen päälle.
Install supports of the frame before installing the test hydraulic unit.
Check the distance of holes before lifting the hydraulic unit (with frame) on top of the supports.

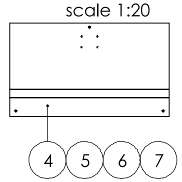
Tukirakenteen jalat
Supports of the frame

2 Korvaa perusrakenteen M36x80-pultit pidemmillä M36x90-pulteilla.
Replace M36x80 bolts with M36x90 bolts.



Cover (to upper pressure plate) with holes for HIT 1500 inclinometer.

scale 1:20



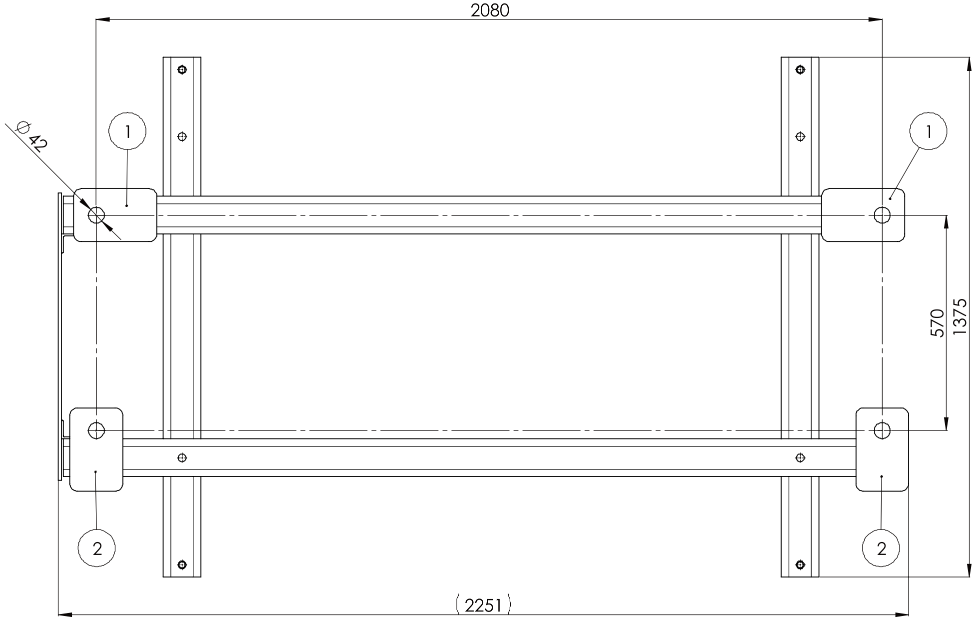
3
Testihydrauliyksikön öljysäiliö täytetään öljyllä ennen kuin hydrauliyksikkö nostetaan suodattimen päälle.
Säiliöön täytetään 600 litraa öljyä.
Oil tank of the test hydraulic unit is filled with oil before the hydraulic unit is lifted on top of the filter.
Volume of hydraulic oil in the tank is 600 liters.

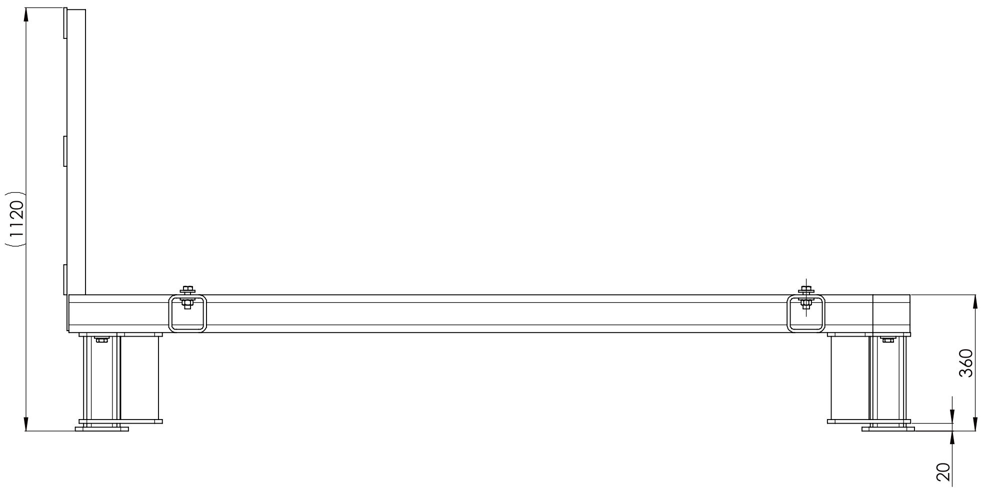
Assembly	F730883 , B	~~438~~	
CONFIGURATION	ITEM	WEIGHT	MATERIAL

Dimensional tolerances	Machining EN 22768-1-m Thermal cutting EN ISO 9013-I	Welded constr. EN ISO 13920-B Welding Quality level EN 5817 C
Outotec	Description: ASSEMBLY PARTS	Weight: ~~438~~
		Drawn 13.05.19 ANESAI; Check 04.06.19 ANESAI; Scale 1:50 A2
	HYDRAULIC TEST UNIT	Drawing no. F730883; Revision B

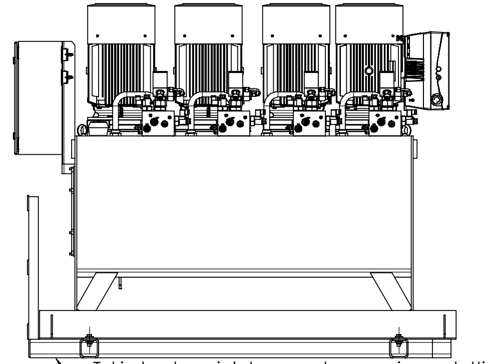
PREVIOUS CODE:
3D

Rev	Info
A	
B	Part 3 changed.



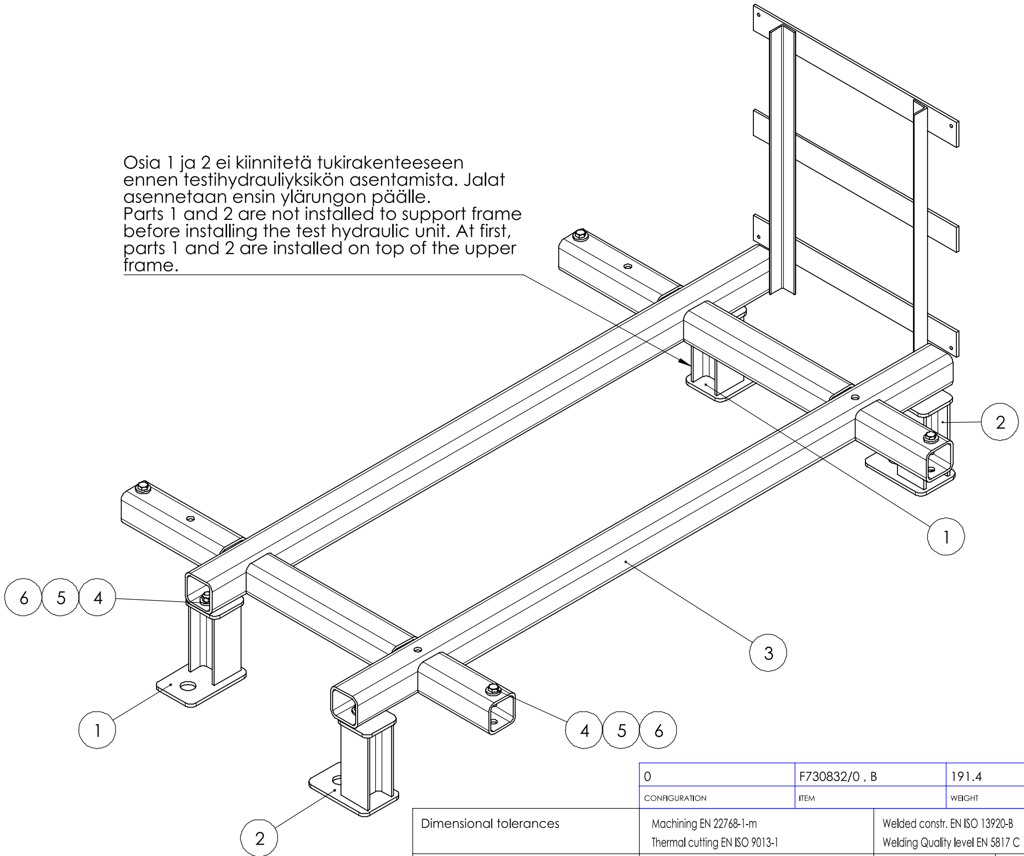


Test hydraulic unit with support frame (scale 1:20)



Tukirakenteen jalat asennetaan ensin suodattimen ylärungon päälle. Lattiatasossa testihydrauliyksikkö asennetaan tukirakenteen päälle.

Osia 1 ja 2 ei kiinnitetä tukirakenteeseen ennen testihydrauliyksikön asentamista. Jalat asennetaan ensin ylärungon päälle.
Parts 1 and 2 are not installed to support frame before installing the test hydraulic unit. At first, parts 1 and 2 are installed on top of the upper frame.



CONFIGURATION	ITEM	WEIGHT	MATERIAL
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Dimensional tolerances	Machining EN 22768-1-m Thermal cutting EN ISO 9013-I	Welded constr. EN ISO 13920-B Welding Quality level EN 5817 C
Outotec	Description: SUPPORT ASSEMBLY HYDRAULIC UNIT HYDRAULIC TEST ASSEMBLY	Weight: 191.4

	Date	Name	Scale
Drawn	10.05.19	ANESAI	1:10 A2
Check	04.06.19	ANESAI	

Drawing no. F730832 — Revision B

PREVIOUS CODE:

3D

Multilevel BOM

Item: F730883, B
Drawing: F730883,B
Description: ASSEMBLY PARTS, HYDRAULIC TEST UNIT

Created: 04.06.2019 A. Sairanen
Checked: 04.06.2019 A. Sairanen
Approved:

Partno	Item	Description 1	Description 3 - 4	Drawing No.	Quantity	Weight
*1	F730832/0, B	SUPPORT ASSEMBLY	HYDRAULIC TEST ASSEMBLY	F730832,B	1 pcs	191 kg
**1	F730811/0, A	SUPPORT		F730811,A	2 pcs	14.0 kg
***1	F730812/0, A	PLATE	S355	F730812,A	1 pcs	2.3 kg
***2	135-46-01000, A	I-PROFILE	HE 100 A, 96x100x5 L = 220; 1 PCS SFS-EN 10034, S355J2G3/4 L=220		0.220 m	3.6 kg
***3	F730814/0, A	PLATE	S355	F730814,A	1 pcs	1.1 kg
**2	F730835/0, A	SUPPORT		F730835,A	2 pcs	14.8 kg
***1	F730812/0, A	PLATE	S355	F730812,A	1 pcs	2.3 kg
***2	135-46-01000, A	I-PROFILE	HE 100 A, 96x100x5 L = 240; 1 PCS SFS-EN 10034, S355J2G3/4 L=240		0.240 m	4.0 kg
***3	F730814/0, A	PLATE	S355	F730814,A	1 pcs	1.1 kg
**3	F730829/0, B	SUPPORT FRAME	HYDRAULIC TEST ASSEMBLY	F730829,B	1 pcs	162 kg
***1	136-01-01005, A	SQUARE TUBE	100x100x8.0 S355 J2G3 L = 2128; 1 PCS L=2128		2.128 m	45.8 kg
***2	136-01-01005, A	SQUARE TUBE	100x100x8.0 S355 J2G3 L = 2226; 1 PCS L=2226		2.226 m	47.8 kg
***3	136-01-01005, A	SQUARE TUBE	100x100x8.0 S355 J2G3 L = 367.5; 2 PCS L=368		0.735 m	15.6 kg

Multilevel BOM

Item: F730883, B
Drawing: F730883,B
Description: ASSEMBLY PARTS, HYDRAULIC TEST UNIT

Created: 04.06.2019 A. Sairanen
Checked: 04.06.2019 A. Sairanen
Approved:

Partno	Item	Description 1	Description 3 - 4	Drawing No.	Quantity	Weight
***4	136-01-01005, A	SQUARE TUBE	100x100x8.0 S355 J2G3 L = 542; 2 PCS L=542		1.084 m	23.4 kg
***5	136-01-01005, A	SQUARE TUBE	100x100x8.0 S355 J2G3 L = 265.5; 2 PCS L=266		0.531 m	11.4 kg
***6	1062, 9 S235JR	ANGLE BAR	L50x50x5 S235JR L = 850; 2 PCS L=850		1.700 m	6.4 kg
***7	1071, 9 S235JRG2	FLATBAR	8x80 S235JRG2 L = 760; 3 PCS L=760		2.280 m	11.4 kg
**4	L24491, A	WASHER	M16 DIN7349 FE/ZN, 17/40X6 GALVANIZED		16 pcs	
**5	1523, 9	HEX SCREW	M16x50 - 8.8 - DIN933 ZINC COATING		8 pcs	0.8 kg
**6	H1283407, 0	HEX NUT	M16 - 8.8 - DIN934		8 pcs	
*2	L613378, A	HEX SCREW	M36x90 - 8.8FEZN - DIN933 ZINC COATING		4 pcs	
*3	L40740, A SP: BP Energol HLP 46, BP Energol SHF 46 CASTROL: Hyspin AWS 46 ESSO: Nuto HP 46 GULF: Harmony 48 AW Oile Hydraulic SHELL: Shell Tellus Oil 46 TEBOIL: Tebo Hydraulic oil 46 MOBIL: Mobil DTE 15, Mobil DTE 25	OIL	ISO VG 46		600.000 l	
*4	F731914/1, A	PROTECTING COVER	UPPER PRESSUREPLATE 1.4301	F731914,A	1 pcs	6.6 kg

Outotec

Multilevel BOM

Item: F730883, B
Drawing: F730883,B
Description: ASSEMBLY PARTS, HYDRAULIC TEST UNIT

Created: 04.06.2019 A. Sairanen
Checked: 04.06.2019 A. Sairanen
Approved:

Partno	Item	Description 1	Description 3 - 4	Drawing No.	Quantity	Weight
**1	L05933, A	DUMMY MATERIAL	1.4301		1 pcs	0.4 kg
*5	1619, 9	WASHER	M5 FEZN DIN126		8 pcs	
*6	2537, 9	HEX NUT	M5 FEZN DIN934 GALVANIZED		4 pcs	
*7	L53370, A	HEX SCREW	M5x20 - 8.8 - DIN933 ZINC COATING		4 pcs	

Rev	Info	Created	Date
A		ANESAI	28.05.19
B	Structure updated.	ANESAI	13.06.19

APPENDIX V, 1

TEST ASSEMBLY 1

Plugs for hydraulic pipelines (pipelines for cylinders):
part 7 (8 pcs)

Elbow connectors (part 19) for ports of cylinders 2 and 3.

HOSES BETWEEN CYLINDERS AND HYDRAULIC TEST UNIT:

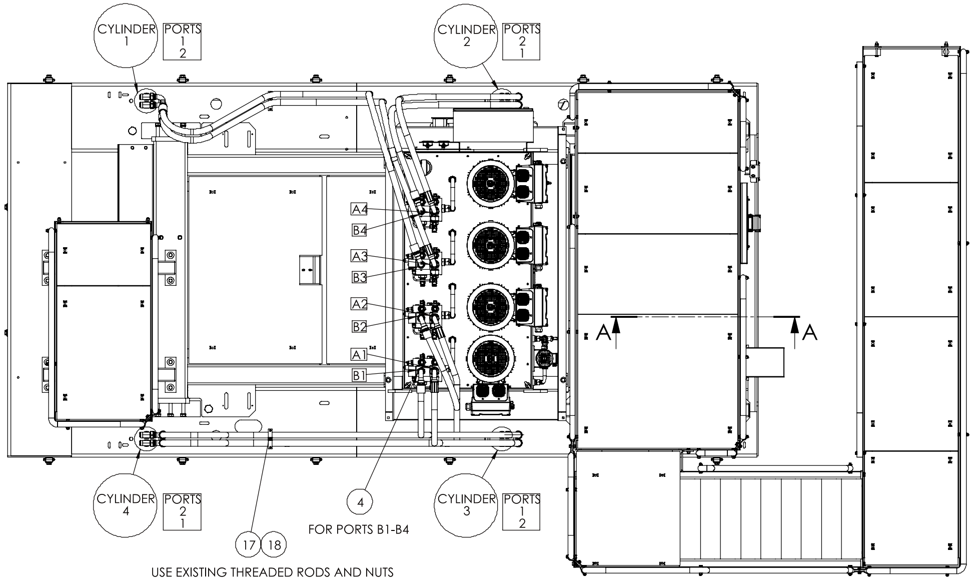
cylinder 1 - A3 B3
parts 9 and 10

cylinder 2 - A4 B4
parts 11 and 12

cylinder 3 - A1 B1
parts 13 and 14

cylinder 4 - A2 B2
parts 9 and 10

Port 1 from cylinder is connected to port B in hydraulic test unit. Port 2 from cylinder is connected to port A in test unit.

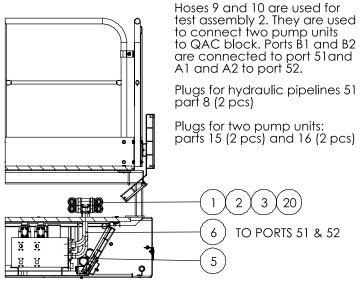


TEST ASSEMBLY 2

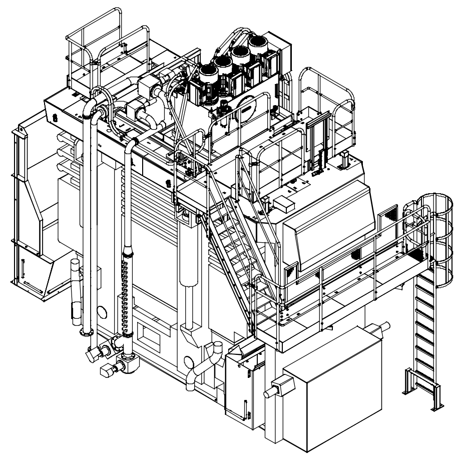
Hoses 9 and 10 are used for test assembly 2. They are used to connect two pump units to QAC block. Ports B1 and B2 are connected to port 51and A1 and A2 to port 52.

Plugs for hydraulic pipelines 51 and 52:
part 8 (2 pcs)

Plugs for two pump units:
parts 15 (2 pcs) and 16 (2 pcs)



A-A



SCALE 1:50

CONFIGURATION	ITEM F730990 , B	WEIGHT 127.6	MATERIAL

Dimensional tolerances	Machining EN 22768-1-m Thermal cutting EN ISO 9013-1	Welded constr. EN ISO 13920-B Welding Quality level EN 5817 C
Outotec	Description: HYDRAULIC PIPELINES TEST ASSEMBLIES	Weight 127.6 Drawn 28.05.19 ANESAI Check 13.06.19 ANESAI Scale 1:20 A1 Drawing No. F730990 Revision B

PREVIOUS CODE:

3D

Multilevel BOM

APPENDIX V, 2

Item: F730990, B
Drawing: F730990,B
Description: HYDRAULIC PIPELINES, TEST ASSEMBLIES

Created: 13.06.2019 A. Sairanen
Checked: 13.06.2019 A. Sairanen
Approved:
Weight: 131 kg

Partno	Item	Description 1	Description 3 - 4	Drawing No.	Quantity	Weight
*1	50142, A A) Desc 2 was TRTXS 38-30	REDUCING CONNECTOR BODY	24-20TRTXS		4 pcs	2.4 kg
*2	50135, A A) Desc 2 was JTXS 38 Item ownership changed to Enovia	T-CONNECTOR BODY	24JTXS		2 pcs	2.0 kg
*3	50145, A A) Desc 2 was BTXS 38 Item ownership changed to Enovia	SHELL NUT	24BMTXS, STEEL PARKER		4 pcs	1.3 kg
*4	50122, A A) Desc 2 was F40XS 30-1.1/4 Item ownership changed to Enovia	CONNECTOR BODY	20F42EDMXS		4 pcs	1.5 kg
*5	N048008451, 1 Enovia Library Component. Can not be edited / revised in Aton.	SOCKET SCREW	ISO 4762-M10x45-A4-80		8 pcs	0.3 kg
*6	F731450, A	HYDRAULIC PIPE	TEST ASSEMBLY 2	F731450,A	2 pcs	4.6 kg
**1	L70468, A Flange connection including flange (jump size), insert, F37 seal and O-ring.	CONNECTION FLANGE	F37-320-38X4.0TFVCF SAE 3000, ZINC PLATED STEEL PARKER		1 pcs	0.6 kg
**2	L03221, A	PIPE	Ø38x4 1.4436 L = 388; 1 PCS SEAMLESS L=388		0.388 m	1.3 kg
**3	50148, 9 Item ownership changed to Enovia	TENSIONING SLEEVE	TXS 38 , STEEL		1 pcs	0.1 kg
**4	50145, A A) Desc 2 was BTXS 38 Item ownership changed to Enovia	SHELL NUT	24BMTXS, STEEL PARKER		1 pcs	0.3 kg
*7	54121, A A) Desc 2 was FNTXS 30	PIPE PLUG	20FNMTXS		8 pcs	2.2 kg
*8	54281, A A) Desc 2 was PNTXS 38	HOSE PLUG	24PNTXS		2 pcs	0.7 kg

Outotec

Multilevel BOM

Item: F730990, B
Drawing: F730990,B
Description: HYDRAULIC PIPELINES, TEST ASSEMBLIES

Created: 13.06.2019 A. Sairanen
Checked: 13.06.2019 A. Sairanen
Approved:
Weight: 131 kg

Partno	Item	Description 1	Description 3 - 4	Drawing No.	Quantity	Weight
*9	L613593, B	HYDRAULIC HOSE	11C77-38-20 / 787-20 / 10677-20-20 L = 4800 PARKER		2 pcs	33.0 kg
*10	L613598, B	HYDRAULIC HOSE	13977-20-20 / 787-20 / 10677-20-20 L = 4800 PARKER		2 pcs	33.0 kg
*11	L613594, B	HYDRAULIC HOSE	11C77-38-20 / 787-20 / 10677-20-20 L = 3100 PARKER		1 pcs	11.6 kg
*12	L613597, B	HYDRAULIC HOSE	13977-20-20 / 787-20 / 10677-20-20 L = 3100 PARKER		1 pcs	11.6 kg
*13	L613595, B	HYDRAULIC HOSE	11C77-38-20 / 787-20 / 10677-20-20 L = 2400 PARKER		1 pcs	9.5 kg
*14	L613596, B	HYDRAULIC HOSE	13977-20-20 / 787-20 / 10677-20-20 L = 2400 PARKER		1 pcs	9.5 kg
*15	N048091509, 1 Enovia Library Component. Can not be edited / revised in Aton.	BLANKING PLUG	VSTI11/4EDCF PARKER		2 pcs	0.4 kg
*16	H1791888, 0 with nut / mit Mutter	BONNET LOCK	VKAM38SCF PARKER		2 pcs	
*17	L66236, A PARKER Light construction series, inside smooth	FASTENING PART	RAPG 645X SERIES A PARKER		4 pcs	0.4 kg
*18	L62121, A	COVER PLATE	DPA6X STEEL, GALVANIZED, LIGHT SERIES A PARKER		4 pcs	

Outotec

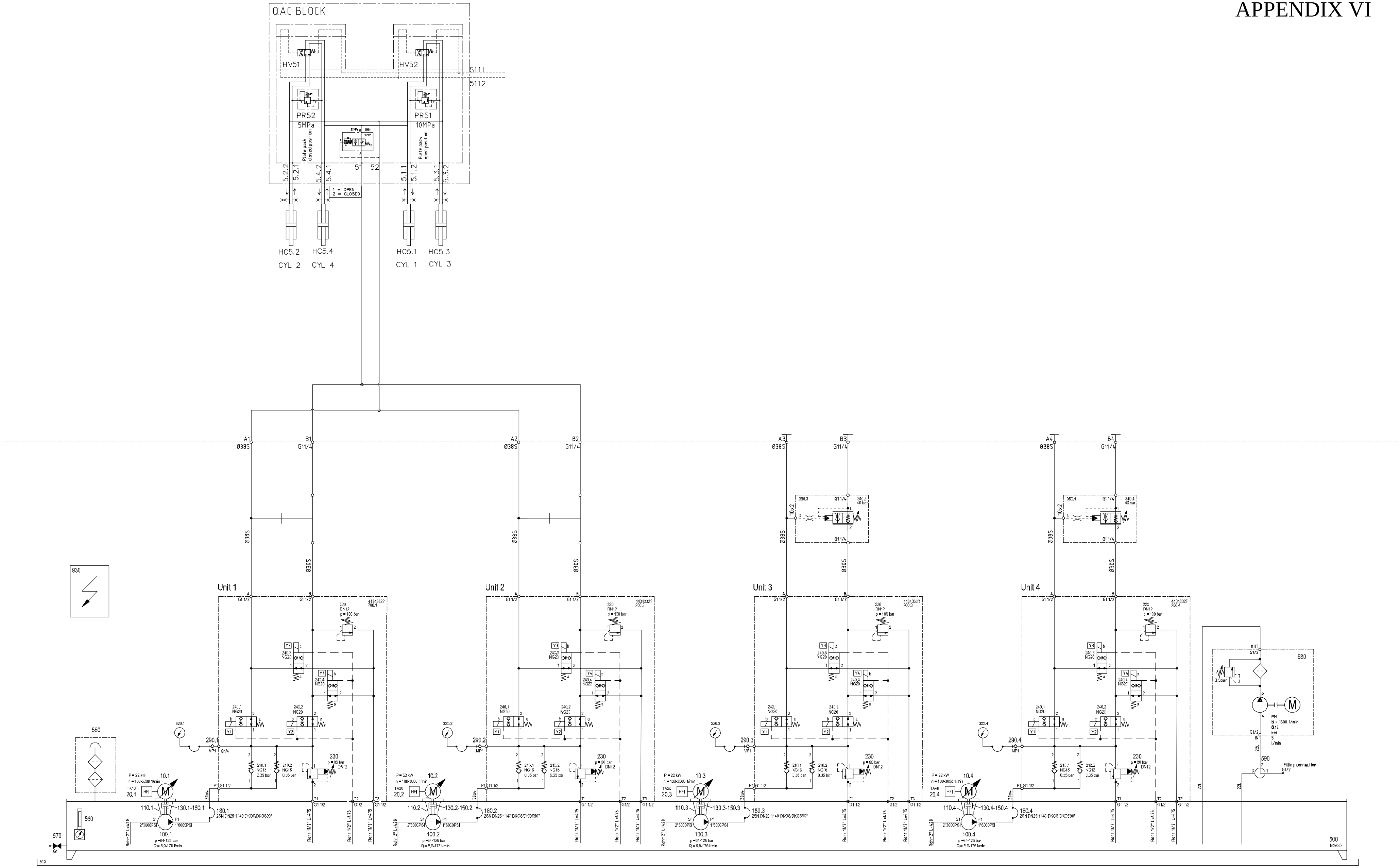
Multilevel BOM

Item: F730990, B
Drawing: F730990,B
Description: HYDRAULIC PIPELINES, TEST ASSEMBLIES

Created: 13.06.2019 A. Sairanen
Checked: 13.06.2019 A. Sairanen
Approved:
Weight: 131 kg

Partno	Item	Description 1	Description 3 - 4	Drawing No.	Quantity	Weight
*19	50131, A A) Desc 2 was C6XS 30	THREADED CONNECTOR	20C6MXS		4 pcs	2.7 kg
*20	51107, A A) Desc 2 was C6XS 38 Item ownership changed to Enovia	THREADED CONNECTOR	24C6XS		4 pcs	4.0 kg

APPENDIX VI



Dimensional tolerances	Machining EN 22768-1-m Thermal cutting EN ISO 9013-1	Welded constr. EN ISO 13920-B Welding Quality Level EN 5817 C
Outotec	Description HYDRAULIC DIAGRAM PF60 TEST ASSEMBLY 2	Weight (kg) Drawn 03.07.2019 ANESAI Check Scale / Size 1:1 A0 Drawing.No F733445 Revision A

Sheet 1/ 1

1122849.dwg

Table with results of test assembly 1.

Test assembly 1, tests 1 – 4											
		opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)			active power (kW)		pressure, pump (bar)		
Target speed (mm/s)	Calculated speed (mm/s)	min.	max.	min.	max.	inclination (deg)	min.	max.	min.	max.	inclination (deg)
40	39,1	19,6	28,4	21,2	35,1	0,1	16,5	18,7	16,2	20,8	0,1
Standard deviation		0,25	0,41	0,23	0,08		0,29	0,21	1,51	0,21	
Test assembly 1, tests 5 – 8											
		opening the plate pack					closing the plate pack				
Target speed (mm/s)	Calculated speed (mm/s)	active power (kW)		pressure, pump (bar)		inclination (deg)	active power (kW)		pressure, pump (bar)		inclination (deg)
		min.	max.	min.	max.		min.	max.	min.	max.	
60	57,3	36,0	48,9	26,6	40,3	0,1	30,5	34,9	23,5	26,3	0,1
Standard deviation		0,35	0,61	0,41	0,43		0,38	0,43	0,04	0,25	
Test assembly 1, tests 9 – 12											
		opening the plate pack					closing the plate pack				
Target speed (mm/s)	Calculated speed (mm/s)	active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
80	75,9	63,5	81,5	31,7	46,4	0,1	52,4	58,1	27,5	31,8	0,2
Standard deviation		0,54	1,06	1,93	0,22		0,65	0,92	0,61	0,43	
Test assembly 1, tests 13 – 16											
		opening the plate pack					closing the plate pack				
Target speed (mm/s)	Calculated speed (mm/s)	active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
85	79,6	71,8	90,8	34,1	47,3	0,2	59,4	65,6	27,3	32,8	0,2
Standard deviation		1,03	0,83	0,22	0,43		0,41	0,32	0,83	0,43	

Table with results of test assembly 2.

Test assembly 2, tests 1 – 4											
		opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)			active power (kW)		pressure, pump (bar)		
Target speed (mm/s)	Calculated speed (mm/s)	min	max	min	max	angle (deg)	min	max	min	max	angle (deg)
40	38,7	14,0	22,1	33,9	62,1	0,2	12,4	15,9	29,8	38,3	0,2
Standard deviation		0,00	0,15	0,22	0,76		0,41	0,22	0,83	0,43	
Test assembly 2, tests 5 – 8											
		opening the plate pack					closing the plate pack				
Target speed (mm/s)	Calculated speed (mm/s)	active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
60	56,1	25,5	38,5	42,4	71,0	0,2	22,6	25,4	36,6	42,5	0,2
Standard deviation		0,50	0,87	0,41	1,00		0,41	0,65	0,41	0,50	
Test assembly 2, tests 9 – 12											
		opening the plate pack					closing the plate pack				
Target speed (mm/s)	Calculated speed (mm/s)	active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
70	65,7	34,3	49,5	47,3	75,3	0,1	28,8	30,9	38,0	43,9	0,2
Standard deviation		1,01	1,06	1,03	0,83		0,43	0,22	0,71	0,22	

Table with results of the current system.

The current system, series connection, tests 1 – 4											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
40	38,5	17,6	27,6	33,0	65,9	0,2	17,3	19,8	31,8	35,8	0,2
Standard deviation		0,41	0,41	0,00	0,22		0,43	0,43	0,43	0,43	
The current system, series connection, tests 5 – 8											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
60	58,0	27,5	40,8	45,5	78,0	0,2	25,3	28,8	39,0	45,8	0,3
Standard deviation		0,50	0,75	0,35	0,71		0,43	0,43	0,71	0,43	
The current system, series connection, tests 9 – 12											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
70	67,0	33,0	49,5	52,0	85,0	0,3	31,0	33,8	44,8	49,9	0,4
Standard deviation		0,71	0,50	0,00	0,00		0,00	0,43	0,83	0,22	
The current system, series connection, tests 13 – 16											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
80	76,0	40,5	59,5	59,0	91,9	0,3	36,0	39,0	49,8	54,1	0,4
Standard deviation		0,50	0,50	0,00	0,57		0,00	0,71	0,43	0,22	
The current system, series connection, tests 17 – 20											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
85	81,8	46,8	65,8	65,3	97,3	0,4	40,8	44,0	53,3	57,3	0,6
Standard deviation		0,83	0,43	0,43	0,43		0,43	0,00	0,43	0,43	
The current system, parallel connection, tests 1 – 4											
Target speed (mm/s)	Calculated speed (mm/s)	opening the plate pack					closing the plate pack				
		active power (kW)		pressure, pump (bar)		angle (deg)	active power (kW)		pressure, pump (bar)		angle (deg)
		min	max	min	max		min	max	min	max	
40	38,2	31,8	40,8	40,0	56,3	0,2	38,8	41,0	53,0	55,0	0,8
Standard deviation		0,43	0,83	0,00	0,43		0,43	0,71	0,00	0,00	