Secondary Hydraulics: An Engineering Approach

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Abstract

Secondary hydraulics is used in many systems, for instance for winches on offshore cranes, but also for amusement rides. The main advantage of secondary hydraulics is the low inertia of the hydraulic motor, leading to a low response time, making the system very accurate. The main goal of this article is to explain the basics of secondary hydraulics and demands on a secondary driven system. Furthermore it will give inside in the "do's and don'ts" for this kind of system and the reasoning behind the "do's and don'ts".

To explain secondary hydraulics, it is useful to compare it to primary hydraulics. Primary hydraulics are in general more simple and often cheaper, but secondary hydraulics have other benefits [2]: the response time is very low, which makes it more accurate when used for instance for heave compensation. The speed and tension control is more accurate and fast changes between these modes are possible. Also energy recovery is possible, which is not that easy for primary controlled systems.

For primary controlled systems, the flow normally controls the speed of the actuator, while the pressure over the actuator determines the force. The main difference of secondary controlled systems to 'normal' or 'primary' hydraulics, is that the system is not controlled by flow and pressure, but by the swash plate angle of the hydraulic motor. Due to the constant pressure drop over the motor, the swash plate determines the torque of the motor.

As shown in figure 1, the forces on the winch are the torque provided by the motor and the force on the wire. If the provided torque to the winch is equal to the torque created by the load on the drum, the winch will stay in position or rotate with a constant angular velocity. When the torque is not equal to the torque created by the load on the winch, the winch will have an angular acceleration, meaning it will speed up or slow down, depending on the angular velocity, directions and magnitude of the torque.

This however shows a danger and a drawback of secondary hydraulics: if the control system fails and the swash plate goes to a maximum position, the winch will start to hoist or lower and it will accelerate while doing so. This is why there are multiple security checks required and also why the winch is stopped once one of the requirements is violated. This makes the system more complicated.

The best way to evaluate the design of a secondary driven system, is to look at the power balance or energy balance, instead of looking at the pressure and flow as one would do for a primary controlled system. The power or torque balance is a good tool to evaluate the entire sys-

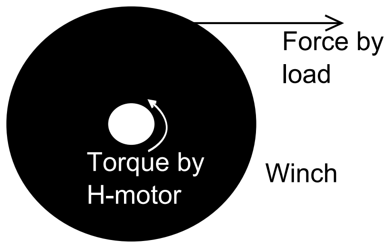

Figure 1: A simplified representation of a winch, including the torque delivered by the hydraulic motor and the force of the wire on the winch. The force of the wire is dependent on the load and multiplied with the distance to the center of the winch, the torque it introduces on the winch is known.

tem and also to understand how the system will respond. Thinking in terms of flow and pressure will be confusing, because due to the influence of the swash plate, these physical quantities do not directly determine the torque and speed. To illustrate: a certain flow to the motor at the constant pressure, can mean a very slow movement with high torque with a large swash plate angle, or a very fast rotation of the winch with a very low torque and a small swash plate angle. Also, the pressure over the motor is constant, but the delivered torque is not the same, but can be high (large swash plate angle) or low (small swash plate angle). Therefore, flow and pressure are not representative for the behavior of the system. Note that the power is the same for both situations mentioned above, which makes the power a better physical quantity to use while designing and analyzing a secondary controlled system.

To recapitulate: The pump supplies power and keeps the pressure line on constant pressure, the swash plate in the motor determines the torque and speed.

1 The Basics

Secondary driven systems are most commonly closed loop systems, meaning that the return line goes back to the pump directly, instead of to a reservoir as in open loop systems. A simplified example is shown in Figure 2.

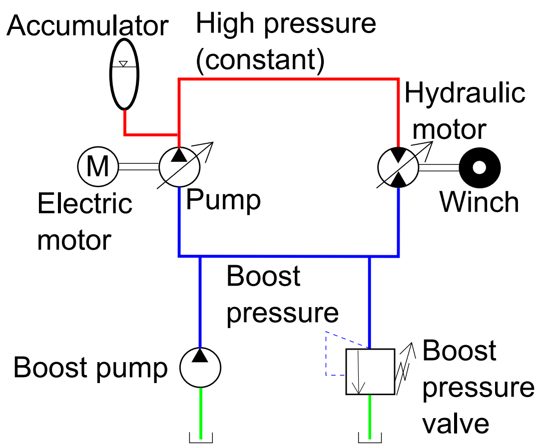

Figure 2: A simplified hydraulic diagram of a secondary controlled winch

The hydraulic motor (or motors, as more motors can be used on one winch) is the element to start the system design with. The motor should be able to be large enough to deliver the maximum torque at the maximum rotational speed, as are stated in the system specifications. Keep in mind that a gear box can change the torque and angular velocity at the winch, but it cannot change the delivered power (multiplication of the torque and angular velocity)

The electric motor and pump are part of the HPU. Secondary systems are used in heave compensation systems. Their power consumption shows a period signal, which is why the HPU can be fitted with a piston accumulator, as is shown in [1]. The pump keeps the high pressure line at a constant pressure. The boost pump and the boost pressure valve are also in the HPU. The boost pressure valve limits the pressure in the boost line, to keep the pressure constant and allows used and warm oil to be flushed out of the system. The boost pump flushes fresh oil into the system, keeping the oil in the system within the working limits. These working limits are mainly determined by the temperature of the oil, which influences the viscosity. This will be further explained in section 2.3.2.

1.1 Hoisting and Lowering

Assume that the winch has no rotational speed. The brakes are lifted, so the hydraulic motor needs to hold the load. To hold the load, the hydraulic motor must provide some torque. To achieve this, the displacement of the hydraulic motor can be changed by changing the angle of the swash plate. The torque of a hydraulic motor is calculated as follows:

$$T_{motor} = \frac{\Delta p \, V_{sg\ actual}}{20\,\pi} = \frac{\Delta p \, V_{sg\ max}}{20\,\pi} \frac{\alpha}{\alpha_{max}} \quad (1)$$

The last formula assumes that the change in displacement is linearly dependent on the angle of the swash plate, which is normally the case for axial piston motors. The variables are explained in table 1.1 below.

Table 1.1 **Explanation of the variables**

T_{motor}	Theoretical torque provided by the motor in $[N\ m]$
Δp	Pressure difference over the motor in $[bar]$
$V_{sg\ max}$	Maximum volume displacement per rotation in $\left[\frac{cm^3}{rev}\right]$
$V_{sg\ actual}$	Actual volume displacement per rotation in $\left[\frac{cm^3}{rev}\right]$
α	Actual angle of the swash plate
α_{max}	Maximum angle of the swash plate

Now a speed command in the hoist direction is given to the winch. The hydraulic motor must provide the torque to hold the load plus the torque to give the winch an angular acceleration. This is done by changing the swash plate angle, to increase the volume displacement of the motor, which increases the torque. Mind here that the pressure at the high pressure and boost side are kept the same, so the pressure drop over the winch is constant. The winch starts to accelerate due to the extra torque. When the winch is at the desired speed, the swash plate is set back to the position where the torque of the motor matches the load again, with the distinction that the winch and motor are now rotating at a constant speed.

Now that the winch and motor are rotating with a constant speed, a speed reduction command is given. The torque of the motor needs to be lowered to decelerate the winch. The swash plate is moved to decrease the volume displacement of the motor, so the motor delivers less torque than the load. This causes an angular deceleration and decreases the angular velocity. The deceleration continues even when the winch stops and then the winch gets an angular velocity in the opposite direction as where the winch was turning. Now the load is being lowered. When the desired velocity is reached, the swash plate of the motor is again put in the position where the torque of the load is equal to the torque of the motor, which means that the winch does not accelerate any more and the velocity is constant.

1.2 Mooring

When the load is accelerated in hoisting direction or when the winch has a constant angular velocity, the oil flows from the high pressure side of the motor to the boost side. This is because the motor delivers a torque larger than or equal to the load. This process requires energy provided by the pump. When the load is accelerated in lowering direction or is lowered at a constant velocity, the oil flows from the boost side of the motor to the high pressure side. Keep in mind that if the winch rotates in lowering direction, the motor also rotates in lowering direction. Now the energy of the load is transferred into hydraulic pressure, thus hydraulic energy. There are three ways to handle the extra hydraulic energy:

- Store the energy in hydraulic accumulators and nitrogen, causing a small pressure increase;
- Dissipate the energy over a relief valve, transferring the energy into heat in the oil;
- Absorb the energy by the pump.

If the energy is absorbed by the pump, it transfers the energy to the motor that drives the pump.When the pump is driven by a diesel motor, there is a maximum amount of power that it can absorb (rule of thumb is maximum 20% of the the total installed power). When the pump is driven by an electrical motor, this conversion to electrical power is called **mooring**. The electrical motor can absorb the energy and transfer it back to the electrical grid.

When a system uses mooring, it is important that there are sufficient other users in the electrical grid to use the extra electrical energy. On vessels it is often not a problem, because during offshore operations the thrusters are also running, which often have a large power demand. Although there are advantages of mooring, most (electrical) engineers are worried about the phase shift induced by the mooring (cosine ϕ shift) and higher order harmonic distortion. In practice it is however not a problem when the electrical grid is designed to handle this. During mooring the angular velocity of the electrical motor and pump will become higher. The electrical motor is dominant in this effect, but in practice an angular velocity increase of 50[rpm] is feasible. The electrical motor and pump must both be able to handle this higher angular velocity. Another advantage of mooring is that less cooling power is required then when the energy is converted into heat over a relief valve. This is mainly due to the fact that the oil, pump or electrical motor are not heated as much, because the energy is fed back into the electrical grid. Mind that when mooring is not applied, it is required that the cooling system has sufficient capacity to also cool the energy of the load during lowering. Another thing to keep in mind is that a winch, especially an offshore winch, can lower for quite some time.

1.3 Pressure settings

There are three basic pressure settings for the system:

- The HP low pressure setting;
- The reduced speed setting;
- The LP low pressure setting.

The HP low pressure setting is the pressure at which the system is just able to deliver the maximum Safe Working Load (SWL). When this pressure is reached, the system is to be stopped, as the load could be dropped when the pressure is too low.

The reduced speed setting is normally 20bar above the HP low pressure setting. When this pressure is reached, it is a sign that the power demand is larger than available power of the HPU and possible accumulator and that the accumulator is almost drained. It can also be caused by a large leakage of the system: internal or external. At the reduced speed pressure the speed, is to be lowered, to lower the power demand and thus prevent that the HP low pressure setting will be reached.

The LP low pressure setting is a setting well below the normal operating pressure. This value should be just above the minimum pressure required by the main pumps. Often a warning is added a few bar above this setting, to warn the operator. The low pressure can be caused by a large leakage, by too few boost pumps running or when the accumulator is being filled.

2 Part Selection

This section will show the main components and the main reasoning of how to select them. The explanation will start with the winch, because the design of the whole system will need to comply with the specifications of the winch. The next step is the HPU, which is the main power supply. After discussing all the aspects of the winch and HPU, also the connections of other users to the system will be discussed.

2.1 General setup of the winch

As already explained in section 1, the best way to evaluate the system, and thus the winch, is to use to amount of required power. The power can be calculated at full speed hoisting or lowering with the maximum load. The maximum load can be expressed in a torque at the winch, dependent on the layer of the wire on the winch. Best is to choose the outer layer, unless the specifications are specific about the maximum load per layer.

$$P_{total} = T_{load}\,\omega_{winch} \tag{2}$$

The definitions of the variables are given in Table 2.1.

Table 2.1 **Explanation of the variables**

P_{total}	Total required power $[W]$
T_{load}	Maximum torque on winch due to load $[Nm]$
ω_{winch}	Angular velocity of the winch at maximum hoisting or lowering $\left[\frac{rad}{s}\right]$

Mind that ω_{winch} is only the speed of the winch, not of the hydraulic motor(s). There are gearboxes and / or a gear-ring between the winch and the motor(s).

2.2 Motor

From the total required power (P_{total}) the motor torque can be determined. Therefore, the gear ratio needs to be determined, which can go up to $i = 500$, but normally is in the order of $i = 100$. The most important factors for choosing the motor are:

- Maximum rotational speed of the hydraulic motors. It is advised to choose the maximum rotational speed at least 100[rpm] below the maximum rotational stated by the supplier, to have some operational room left for the overspeed safety on the motors. To stop the motors when their speed comes too close to the end of the working range of the motor.

- Maximum pressure is determined by the lowest design pressure of all parts in the system. Normally the servo valves controlling the swash plates have the lowest design pressure. When there is no separate pressure line installed to the servo valves, the maximum pressure is 315[bar]. Therefore, the safety should be set to 315[bar]. When there is a separate pressure line to the servo valves, the maximum pressure is the design pressure of the motors and pumps, which is normally 350[bar].

- If the system has a large accumulator installed in the pressure line, the minimum pressure should also be stated. The benefits of a large accumulator are shown in [1]. Normally the minimum working pressure is 50[bar] under the nominal working pressure (normally 300[bar]). It is very important to ensure that the winch can hold the maximum load (maximum torque) at the minimum working pressure.

- The efficiency with which normally the motors are calculated, is 95% volumetric and 95% mechanical. Furthermore, the servo valve to control the motor requires a flow of 5[$L\,\text{min}$]. The efficiency of the motor will decrease for smaller swash plate angles and lower working pressures. In practice the mechanical efficiency at full swash plate angle is 98%. The volumetric efficiency will decrease with increasing motor size. For 1000[cc/rev] motors the volumetric efficiency will be approximately 92%. This shows that the 95% volumetric efficiency is an average.

To summarize: for the choice of the motor, the maximum rotational speed and the minimum and maximum operating pressure are important. The torque of the motor can be calculated using:

$$T_{motor} = \frac{9550\ P_{total}}{n_{motor}\ \eta_{MechMotor}} \tag{3}$$

The minimum required motor size is them calculated using the following formula.

$$V_{sl} = \frac{T_{motor}\ 20\,\Pi}{\Delta p\ \eta_{VMotor}} \tag{4}$$

Table 2.2 **Explanation of the variables**

T_{motor}	Required motor torque [Nm]
$\eta_{MechMotor}$	Mechanical efficiency motor [$-$]
n_{motor}	Rotational speed of the motor [rev/min]
V_{sl}	Minimum required motor size [cc/rev]
Δp	Minimum pressure difference over the motor [bar] (Note: Substract the boost pressure of the minimum operating pressure)
η_{VMotor}	Volumetric efficiency motor [$-$]

2.2.1 Type of motor

For a secondary controlled motor, the most important sensor is the Linear Variable Differential Transducer (LVDT), which measures the position of the swash plate.

The hydraulic motor which is often used, is the Rexroth A4VSG...HD, which has a short and long spindle for emergency operations. The short and long spindle limit the swash plate angle during emergency operations. Other types can be used as well, although the author has no experience with these motors. Most often used are the 1000[cc/rev] and 500[cc/rev], but the 750[cc/rev], 355[cc/rev] and 125[cc./rev] are also available.

The maximum torque can be limited by limiting the swash plate angle in hoisting direction on the motor, in case this is required. Note that the efficiency will decline with smaller swash plate angles.

In the lowering direction, the maximum swash plate angle can be limited on the motor. This helps to prevent overspeed in that direction.

When limiting the swash plate angles, leave enough swash plate angle to stop the winch with no load on the system, including the efficiencies.

For the Rexroth motors, there is a check valve between port A and B, see figure 3, which is not always shown in supplier documents. When the manifold on the motor is designed the wrong way around, the check valve is installed incorrectly. This is easy to change on the motor, but is good to know up front and to notify production.

The motor housing must always be filled with oil. This can be achieved either by placing the motors below the tank of the HPU, or by installing a check valve in the drain piping above the motors.

The motors are normally also flushed from the boost system. An orifice is installed here to prevent too high pressure inside the motor. This has several advantages, such as a guarantee that the motor is filled when starting up, cooling of the bearing (longer life time) and the motor and bearing are within the correct temperature range.

The Rexroth motors have two manifolds on the motor: the first manifold is called the Motor manifold, controlling the flow to the motor and the other manifold is to control the swash plate of the motor and is called the servo manifold.

2.2.2 Motor manifold

The motor manifold is mounted on the high pressure port (Main P line) and low pressure or boost pressure port (LP line) of the motor.
The motor manifold is installed such that the swash plate is set completely to hoisting direction when not actuated. In practice it can be hard to determine on paper what port of the motor is the high pressure port, as this is dependent on the way the winch will rotate in hoisting direction, possible gear-ring and the type and number of stages in the gearbox. In the past this was solved by making the motor manifold such that it can be mounted on both ways, so exchange is relatively easy.

The motor manifold has several valves installed, see figure 3. The motor manifold has a logic valve, which in closed position will disconnect the HP port from the HP line. This logic is called the main logic and is controlled by a directional valve on the manifold. In neutral position, the directional valve closes the main logic for safety reasons. Between the directional valve and the main logic an orifice or throttle check valve is installed, to control the opening time of the logic valve. This should prevent to fast opening of the valve, which can result in pressure peaks in the motor. After the main logic a pressure sensor is installed, to check the functioning of the main logic.

Important function of the main logic, is that it can close in a very short time, so less than $300[ms]$. This is required for swift closure of the valve in the case of overspeed.

On the motor manifold is also a relief valve installed. The relief valve should be capable of handling the flow to the motor, so this is often a logic valve as well (as drawn in figure 3. A smaller relief valve is installed on top, to make the logic valve act as a relief valve. The setting of the relief valve is normally above the setting of the general relief valve on the HPU, so that the motor relief valve will only act as safety when the main logic is closed. This relief valve prevents an overpressure due to load put on the crane when the system is not active.

To control emergency hoisting and lowering, manual valves are installed on the motor manifold. The emergency operations will be explained in section 2.4.

The brake on the motor has often a lower design pressure than the motor. Therefore a 3-way pressure reducing valve is used, often of the brand Moog. When this Moog type reducing valve is used, a point of attention is that the spring chamber should be completely filled. When this is not the case, the valve can break and cause a too low pressure in the brake. This means that the brake then cannot be released. The line to the brake also requires an orifice to prevent too large flows to the brake, which leads to a too violent release of the brake. The brake is controlled by a control valve. This control valve should not have open connections when switching (so it should have positive overlap), to make sure that the valve does not get stuck due to too large flows. A pressure switch is normally installed in the brake line, to tell the PLC that the brake is lifted. When the brake is not fully lifted, the brake will wear quickly and will not be able to hold the load anymore. This is why the PLC will stop the opera-

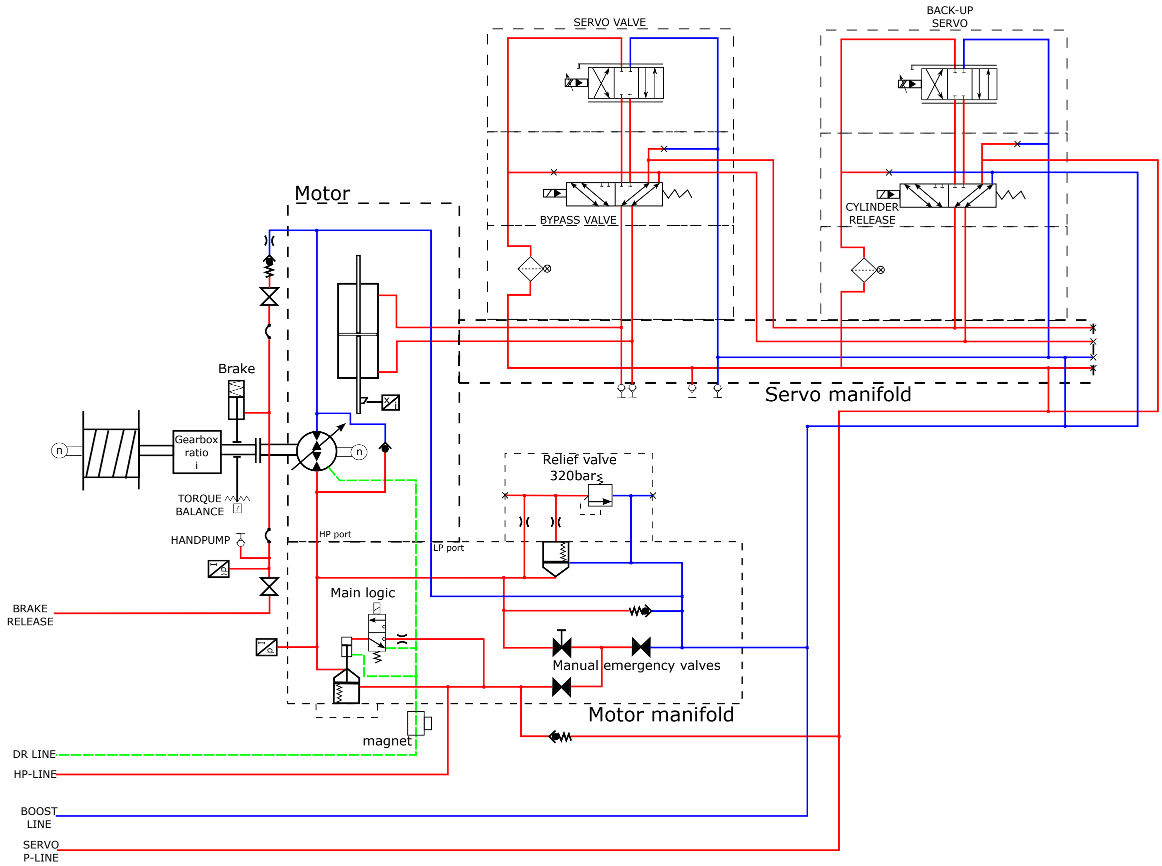

Figure 3: A hydraulic diagram of a secondary controlled winch

tion when the brake is not fully lifted.

2.2.3 Servo manifold

As shown in figure 3 the servo manifold is mounted directly on the motor, just like the motor manifold. The motor has a specific interface, so the manifold must be designed to fit on that interface.

For the servo valves the Rexroth 4WS2EM105X are often chosen, because these have a good performance at the middle position. The middle position is the most important region for this valve, as the adjustments to the swash plate are often small and therefore the control loop needs accurate control. Drawback of this valve is the small negative overlap, causing a relative large leak flow of about 5[L/min] per servo valve. During the design phase it is important for the engineer to check the control cylinder of the swash plate. If the control cylinder is asymmetrical, the servo valve should be asymmetrical as well.

The motor (or one motor if there are multiple) has a back-up servo valve with cylinder release valve installed. The back-up servo must take control when one servo valve fails and stop the operation to ensure a safe stop.

When the back-up servo fails or when the servo valve fails in a system without a back-up valve (tuggers), a fast stop is initiated. This means that the main logic in the motor main logic valve closes within 300[ms], the swash plate is switched to full hoisting and after 500[ms] the brake will apply[1]. This will mean that the main force dur-

[1]The 500[ms] before the brake applies is to give the motor time to stop the winch, before the brake applies, as the brake is not well suited for dynamic braking.

ing braking is about 2.5 times the maximum load (maximum swash plate angle multiplied by the maximum pressure, plus the dynamic braking torque of the brake). Between failure and braking the load is uncontrolled, which is the reason that the system has to respond so rapidly.

The servo valve as well as the back-up servo valve have a special Rexroth 6/2 directional valve. This valve is expensive and has a long lead time, which should be accounted for in the project planning.

Both the servo valve as the back-up servo valve have filters installed in the P-line to prevent dirt from entering the servo valve, as these servo valves are very sensitive to dirt.

The (short) piping between the motor manifold and the servo manifold is flared piping, instead of Walform couplings. This is done because of the vibrations, which caused deformations and leakages in the piping.

The back-up servo can be connected using hoses. Although this influences the accuracy (control become less direct), this is acceptable as the back-up servo is seldom used. Even when used, it will only stop the winch, and it will not be used to continue operation. Hoses can also introduce more dirt, so the hoses should be properly flushed.

2.2.4 Extra inertia

For cranes it is common to add extra inertia (flywheel) between the motor and the brake. There are different opinions on how much inertia is needed. Rexroth documentation mentions 10 times the inertia of the hydraulic motor, although many cranes only have 1 time the motor inertia. Especially when the load has a large inertia, the extra inertia is not required.

The extra inertia will cause a slower response of the control system, although this effect is small, as the gain in the control loop can be enlarged. It will also result in a higher power demand for the same acceleration, but during heave compensation this energy is also again retrieved when the load is decelerated.

2.2.5 Gear box

The used gear boxes are similar to the planetary gear boxes used in other hydraulic systems, although there are two main differences: the used brakes and the torque balance. Furthermore, the gear ratio of the gearbox has a large impact on how the hydraulic motors 'feel' the inertia of the system. For a typical active heave crane (gear ratios between 80 and 120), the inertia of the drum, wire and load would be approximately 50% of the entire 'felt' inertia. For more direct driven systems (gear ratio of 4 for example) this inertia can be up to 95%. The inertia of the planetary gearbox is for a first order of magnitude estimation just as large as the hydraulic motor. The inertia on the motor is more important for fast acting cranes, as inertia decreases the resonance frequencies. When these frequencies are low (2Hz to 10Hz), they can become close to the frequency of the control loop, especially when the gain of the control loop is high. Decreasing the gain would solve this issue, but would also mean that the system responds slower to the error in the control loop, and thus has a slower response.

Brakes Every winch should have parking brakes. For secondary hydraulic systems lamella brakes are often used. These brakes are only used statically. Only in case of failures or a fast stop these brakes are used dynamically.

The lamella of the brake are often wet ('benatzt'), which means that these are dipped in ATF oil before mounting. This reduces the chance that the lamella will stick together when used dynamically. Filling the house of the brake completely is however not required. This will actually reduce the braking torque of the brake.

As the brake is hydraulically lifted, there is a seal between the brake lamella and the hydraulic oil. In the past it has been seen that this seal is not completely tight, which means that some hydraulic oil will reach the lamella, reducing the performance of the brake. This is why it is advised to drain the housing regularly during maintenance. The engineer must make it possible to drain the brake, so make sure there is a drain plug, and that this is reachable.

In normal installation, the brake is a dead end of a hydraulic line, meaning that the oil is normally relatively cold, especially when the lines are long. This can be solved by flushing the brake lines to the boost line, as shown in figure 3.

If there are problems with the installed brakes, there are some things that can be done. By changing the lamella of the brake, or their orientation, the stiffness and thus their braking force can be adjusted. These lamella work as a spring, so changing their position changes the stiffness. It is also possible to replace the lamella by stiffer versions. Keep in mind that these changes also change the required pressure to lift the brake.

In case that the winch has manriding capabilities, a second independent brake is often required. This can be achieved by mounting a caliper brake on the drum of the winch.

Torque balance In most secondary controlled systems a torque balance is used. The torque balance is mounted between the brake housing and the fixed world. The torque balance has two springs, one on each side. When the torque balance is in the middle, there is no force on the springs and the torque of the hydraulic motor is equal to the load. At this point it is allowed to lift the brake and start the operation. This is a check by the control system to make sure that the motor can handle the load, the swash plate control is functioning well and the system pressure is sufficient.

The springs need space to compress and extend. The springs are normally chosen so the spring can hold two times the motor torque, to ensure that the springs do not fail when the motor torque and load are in the same direction. The springs are also not linear. Around the middle position the spring must be soft, to make the detection reliable. Further away from the middle position the springs need to be stiff, to make the compression length smaller. This is achieved by using multiple springs in series.

The switch is a detection switch (Turck) to detect the steel of the torque balance. Normally only one switch per torque balance is required.

On winches it is common that all motors have a torque balance. If there is no load possible in the rest position (for instance amusement rides) the torque balance is not installed.

2.2.6 MOPS

Manual Overload Protection System, or shortly called MOPS, is a protection system that is more and more demanded by offshore rules. It is a protection system for when a crane on one vessel is picking up a load from another vessel. When rigging up it is very undesired that the crane pulls on the load with a significant force, because that might lead to the load being lifted up and coming back down on the other vessel due to heave motion. Therefore, there is an emergency button in the control cabin at the operating console, which prevents large forces by changing the maximum pressure drop over the hydraulic motor. A motor with MOPS installed is shown in figure 4.

MOPS can be installed by changing the valve stack on the safety relief valve on the motor manifold. The extra relief valve is set to a much lower pressure than the original one, so when MOPS is activated, the A and B line of the motor are connected when the line in hoisting direction reaches the set pressure. Mind that the relief valve is connected to the boost line, so the pressure of the boost line is added to the spring setting. It is very common to place two control valves in series, to ensure that when one valve fails, MOPS is not immediately is activated.

The other change shown in figure 4 is the emergency power line instead of the servo P-line. This line is normally connected to the large piston accumulator or other accumulators. This is necessary to operate MOPS even when the HPU is not started. This also requires the piston accumulator or other accumulators to be charged at all time. It is however possible to disconnect this line from the accumulator when the system is off and MOPS is not active, but it should be connected when the MOPS button is pressed. This isolation of the accumulator is used to prevent the accumulator from being drained by leak flows while the system is not operated.

2.2.7 AOPS

An Automatic Overload Protection System, or shortly called AOPS, is also more and more demanded by the notified bodies and rules. The EN13582-1 makes it obligatory, especially for offshore cranes meant for supply lifts (lifts from or

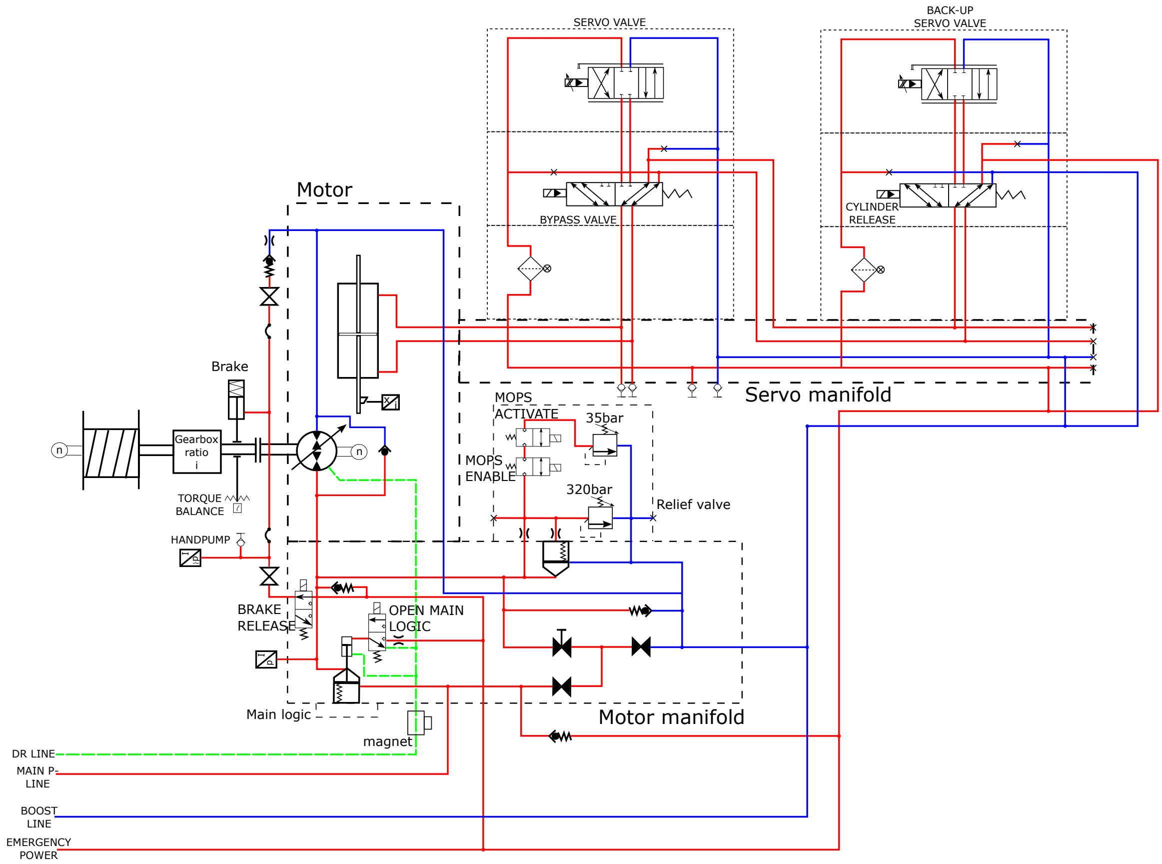

Figure 4: A hydraulic diagram of a secondary controlled winch including MOPS

to a supply boat). AOPS needs to become active when the tip is not above the vessel and the crane is not in manriding mode.

An AOPS function needs to take into account that the maximum SWL changes with the boom angle. As the boom goes down, the tip is further away and thus gives a larger moment at the pedestal. This means that the AOPS also needs to change with the boom and/or knuckle angle.

AOPS functionality can be achieved by controlling the swash plate angle. With the slew, boom and knuckle angle, the crane knows where the tip of the boom is, so it knows at what load the AOPS should become active. During an overload situation (so a load above the SWL at the boom angle) the swash plate has a maximum allowable angle to make sure the winch pays out wire. The maximum allowable swash plate angle is different for different boom angles.

2.3 HPU

An example of the HPU diagram is shown in figure 5. This diagram contains the main components for a closed loop secondair hydraulic system:

1. Reservoir including appendices
2. Boost pumps and enable manifold
3. Main pump
4. Piston accumulator with pressure vessels
5. Flushing and safety manifold

The parts will be discussed in detail in the following subsections. Furthermore, the important cooling and filtering functions are discussed.

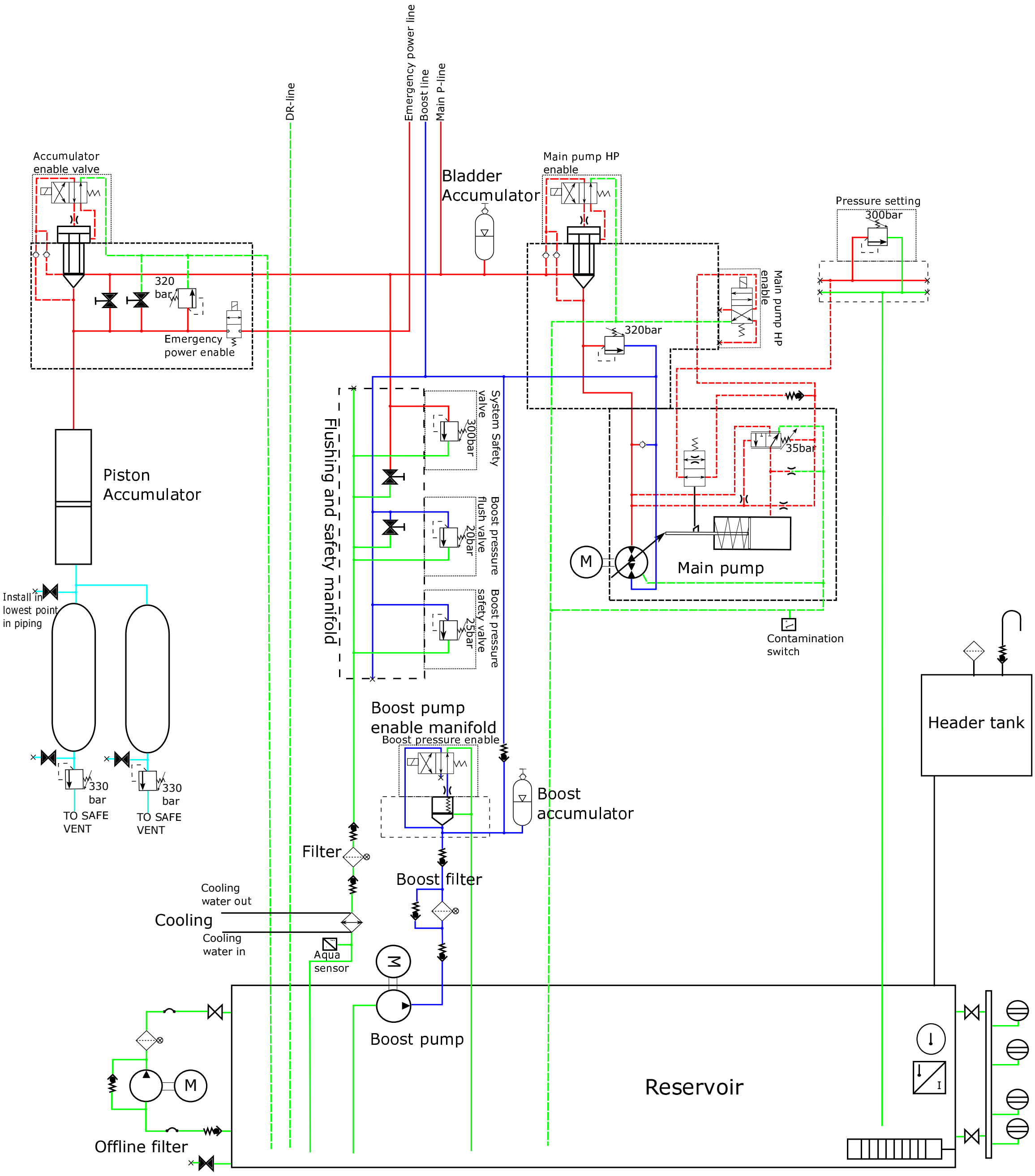

Figure 5: A hydraulic diagram of an HPU for a secondary controlled winch including MOPS

2.3.1 Reservoir and type of oil

The reservoir has the same functions as for normal primary systems, allowing air bubbles to escape, cooling and filtering. There are however a few things the engineer should keep in mind.

To start with the size of the reservoir. For closed loop secondary controlled systems, the size of the reservoir should be 3 times the boost flow plus the volumes required to extend cylinders, fill accumulators and other displaced volumes.

For man-riding applications it is important that during an emergency operation the motors are able to get oil from the reservoir. For DNV cranes it is mandatory that emergency lowering is still possible when all pumps are off (so also the boost pumps are not running). Therefore it is important that for this type of applications an header tank is installed, placed above all motors. The function of the header tank is to keep the motors filled at all times, which is required for the emergency operation for manriding operations. By placing the header tank above the motors, the oil will flow to the motors during the emergency operation due to the hydrostatic pressure. The header tank can be placed outside and on top of the crane house, leading to smaller space claim within the crane house. A header tank does however mean that the reservoir near the pumps is pressurized and completely filled. For cold environments and when the header tank is placed outside (which is often the case), the header tank can be isolated to prevent that the oil becomes too cold. The line between the header tank and the reservoir needs to be large enough to handle the maximum flow of the HPU. Especially the maximum flow of the piston accumulator during heave compensation can be very large and thus it often determines the size of the line between the header tank and the tank in the crane house. When the piston accumulator delivers oil to the system, the oil is flushed out of the system through the flush valve, which results in the same flow through this pipe. The same is true for the breathers on top of the header tank.

When a header tank is used it is very common to use butterfly valves in the line to the reservoir or other lines. Especially the line to the reservoir needs position detection, because the system is not allowed to start when the butterfly valve is closed.

The oil to be used in the system is normal ISO VG 46, but especially for larger motors the efficiency becomes better with ISO VG 68 oil. As the system contains servo valves, the oil needs to be compatible with the the valves. Also the cleanliness is an issue: Hydraulic oil as bought is often NAS class 10 or 11, while this system requires it to be much cleaner to guarantee the lifetime of the system. This means that the oil needs to be filtered before it is used in the system.

2.3.2 Boost system

The boost system has four main components: The boost pump, the boost accumulators, the boost pump enable manifold and the flushing and safety manifold.

The boost system is often required due to the chosen hydraulic motors, which commonly demand a boost pressure of at least 14bar. It is common that the boost system is chosen between 20 and 30bar. The largest advantage of the higher pressure is that the boost accumulators have a larger capacity compared to a lower pressure. The boost system keeps the return line of the closed loop system on a certain pressure with the use of boost pumps. The oil coming back from the winch to the HPU is partly flushed out of the system through a cooler and filter. Because the secondary controlled system is a closed loop system, it means that when the load is lowered, the oil will be pumped from the boost system to the high pressure side by the motor. This means that the oil can go in both directions, as shown in figure 2.

Boost pumps The boost pumps are pumps which normally deliver a lot of flow, although the pressure is relatively not that high. This means that screw pumps can be used, which have been used frequently in the past. Advantages are that the screw pumps can have their own pressure safety integrated, the pumps have a continuous flow (no pulsations) and have a good efficiency for cold oil. Due to this last advantage these pumps can also be used to warm up the oil in the reservoir.

As the boost pumps deliver flow to the boost system, the flush valve will allow just as much flow back to the reservoir. The connection of the boost pump is normally closer to the main pump than the flush valve connection. These connections are at least a few meters apart to prevent that the fresh oil from the boost pumps is directly flushed out of the system through the flush valve. The flow through the flush valve can be larger when the piston accumulator also delivers flow. This extra amount of oil needs to leave the system. When the flow demand drops and the main pump can again deliver all the flow and will fill the accumulator again, the boost pumps need to deliver enough flow to keep up with the filling of the accumulator[2]. This can be quite demanding on the boost system and is in many cases determines the size of the boost pump, flush valve and size of the accumulators in the boost system. The actual amount is very dependent on the maximum heave compensation (vertical vessel motion and the period time).

When the system cannot go into mooring, there should be a relief valve from the HP line to the LP line. Otherwise the boost pumps must be able to deliver the entire relieved flow, which can become demanding on the system.

Boost pump enable manifold A filter is installed normally just after the boost pump. From the filter the oil flows to the boost pump enable manifold. Main goal of this manifold is to keep the pump pressureless during start up, which is less demanding on the electrical motor and makes sure there are no undesired pressure peaks in the boost line.

Boost accumulators The boost accumulators are installed to prevent large pressure drops while the systems goes from hoisting to lowering or the other way around. During the peak flow demands these accumulators can deliver a small part of the flow, although that needs to be calculated for each system.

To have a proper dampening effect in the boost system, the prefill pressure should not be too close to the working pressure.

Flushing and safety manifold The pressure in the boost line is maintained by a boost pressure flush valve, while a safety valve will secure the maximum pressure in the boost system. The difference in setting between these two is normally around 5 bar, to prevent the safety valve from opening during normal operation.

For systems where the force or direction changes rapidly or shifts from one direction to the other, the boost system needs fast responding relief valves to relieve the pressure. Therefore in these kinds of systems, it is beneficial to install multiple smaller relief valves which respond within 100 milliseconds instead of one larger valve. Another option is to install a couple of smaller relief valves with one large relief valves. The smaller relief valves open quicker and deal with pressure peaks, while the larger relief valve requires more time to open, but can relief full pump flow.

2.3.3 Main pumps and piston accumulator

The main pumps, bladder accumulators and the piston accumulator are responsible for the energy supply in the system. As long as the main pumps can deliver the demanded flow, the piston accumulator is at working pressure or possibly delivering flow to the servo valves in the case MOPS is installed. The pressure in the HP line is guarded by a safety relief valve, which is drawn on the flush manifold in Figure 5. The piston accumulator has a separate relief valve, as this piston accumulator can be isolated when pressurized.

Main pumps The most used type of pumps (by the author) are the Bosch Rexroth A4VSG500DP, although other pumps are available. The pump has a hydraulically controlled constant pressure control. This is a closed loop type pump, which means it uses the boost system. There are also open loop pumps available in the market, but then the engineer needs to make sure that there is back pressure and oil supply available for the motors when the system is in mooring. The pressure setting is often set at a separate manifold, to prevent the relief valve from vibrating along with the pump, to prevent relief valve failures. The DP control of Rexroth is used to be able to have several pumps in one system. It means however a 7bar reduction at full swash plate angle. The pressure setting of the pumps is often chosen to be just below 300bar, which is due to the pressure demands of the servo valve. The motors and pumps can handle higher pressures, but that means that the servo valves require a separate supply system.

[2] Normally the boost pumps will only need to supply enough for flushing. When the accumulator is being filled, this amount of oil needs to be added to the system by the boost pump.

The electrical motor that powers the pump is often smaller than the peak power, since the electrical motor can be overloaded for small time periods, as long as it has time to cool down afterward. The maximum speed at maximum load for offshore cranes is normally only a peak value. How small the electrical motor can be chosen, is much dependent on the maximum peak load and period time of the waves. So the motor size should be based on the specifications.

One important difference with primary hydraulics, is that each pump needs to be disconnected from the system once the E-motor is off, especially when a system can go into mooring. This is why there is a main pump HP enable valve on each pump. If this is not done, the piston accumulator or the load can accelerate the pump, get it in overspeed, and the pump and electrical motor can be damaged. This is also why this valve often has a position detection to prevent this from happening.

Due to the HP enable valve, the HP line of the pump needs a relief valve to protect the pump from too high pressure. This relief valve reliefs pressure directly to the boost line, meaning that the pump does not run dry, although the oil can become very hot this way, which is possible when the pump is running and enabled without the main pump HP enable valve to be opened. So an open detection of the HP enable valve is also required. During start up of the HPU, there is a moment when the pump is enabled and runs on full power, but the HP enable valve is not yet open. The same is true for the shutting down of the HPU. Due to the boost pressure (and tank pressure at the pressure setting relief valve) this relief valve is actually used during every start up and shut down of the HPU.

The pumps are often installed with flushing of the bearings from the boost system. The main advantage is that the housing is always filled with oil, as the boost system always has to be started before the main pump is allowed to start. Also the fresh oil at the bearing ensures a longer life time of the bearing and the pump stays at the required working temperature.

The installed power by the pumps must be equal to the nominal demand of the users at one time. The piston accumulator is normally only used for the reoccurring peak values, as this supply is limited.

The installed bladder accumulators close to the pumps are mainly installed as dampeners for pressure peaks of the pumps. For large systems (6x 500cc pumps) 2x 50liter accumulators, prefilled at 65% of the working pressure, is enough. When the lines to the motors are very long, it is possible to install accumulators close to the motors as well, which can help to prevent high pressure peaks during switching of direction or force. Keeping the pipe lines as straight as possible also helps to keep the pressure peaks low.

Piston accumulator As already stated before, the piston accumulator can have several functions. It can be used to limit the size of the HPU by delivering flow at the peak demands, as shown in [1]. It is also possible that the piston accumulator is used to feed the servo valves and for instance feeds the MOPS system. It can also be used as an emergency hoist device, for instance for on stone dumping vessels, where the accumulator is coupled directly to the motors when this function is activated. In the last case the size of the piston accumulator can be become considerably large.

In case of MOPS functionality, it is necessary that the piston accumulator can be isolated from the P-line, because otherwise the pressure could be relieved through internal leakages. When MOPS is not required, it can still be advantageous to have an isolation valve installed. The engineer should however be cautious because an isolation valve can also isolate the piston accumulator due to a failure, resulting in dangerous situations or unacceptable high risks.

The nitrogen side of the piston accumulator needs a separate relief valve, securing the design pressure of the system. This can be reached due to temperature rise (i.e. fire), even when the system is off.

The preferred installation of the piston accumulator is vertical with the oil side at the bottom. This has the smallest chances on dirt entering the piston accumulator or dirt settling at the seals of the piston. Dirt at the seals accelerates the wear of them. Often there is not sufficient space for vertical installation or vertical placement would interfere too much with the piping, space claims or other systems, resulting in horizontal or other installation.

The piston accumulator has a position detection. The option of Hydac, which is most often used by the author, is with a rope measurement system. The rope is installed on a small winch inside the piston accumulator, the position is measured by a potentiometer. Big disadvantage is that the piston accumulator needs to be opened to exchange or fix the length measurement. It is possible to mount a laser measurement on the outside of the separator on the nitrogen side. This makes the exchange a lot easier, although the accuracy at different pressure levels is doubtful. Another possibility is an inductive measurement (Balluff). The length measurement is used to check the pre-fill pressure and check for possible leakages. The piston should always be at a certain position when the pumps are running (so the system is on working pressure) and the piston accumulator is coupled to the system. At large deviations, a warning is given to the operator. The maximum speed of the piston is often determined by the rope measurement, as this device has a limited operating speed due to the spooling of the wire (slack wire must be prevented, otherwise the rope measurement can break).

The position measurement is to be calibrated during commissioning. This must be done before the nitrogen side is completely pressurized to the pre-fill pressure, as it will be hard to reach the end of stroke position at the nitrogen side at full pressure. The end of stroke position is not an operational point, as it should be prevented that the piston accumulator hits the end stop at a significant velocity. This means that the pre-fill pressure should be high enough to make sure that the piston never hits the end of stroke. The pre-fill pressure should however be lower than the "HP pressure too low" alarm, to make sure that the system is stopped before the piston hits the end of stroke position at the oil side (with a considerable speed).

The maximum speed of the piston is also important when the piston accumulator is used for pressurizing the nitrogen side during filling. In a lot of countries (especially Asia) it is not always possible to get nitrogen packs at 300bar. So reaching a high nitrogen pressure could mean that the piston accumulator is used to pressurize the nitrogen, so it would actually pump nitrogen into the pressure vessels. The maximum speed should not be exceeded, as this can damage the seals, meaning that there is leakage from the oil side to the nitrogen side and vice versa.

The nitrogen seal of the piston accumulator, bought at Hydac, is a bean seal. This type of seal has the advantage that it creates a good seal between the oil and the nitrogen, which means that the nitrogen will not dissolve in the oil and thus the system will keep its pre-fill pressure. Furthermore, the piston has two oil seals (Turcon Stepseal V) and Slydrings to keep the piston centralized.

The nitrogen pressure vessels connected to the piston accumulator can be divided into groups, and usually these can be isolated separately. Often manual valves are used, as these only need to be switched during maintenance and repair, not during operation. The valves prevent the need to bleed the entire system during repair. When the pressure vessels are quite small, the engineer can also consider using bladder accumulators, as these are considerably cheaper for small volumes than piston accumulators.

2.3.4 Cooling

An important part of the boost system is to make sure that the oil is cooled sufficiently. The engineer should make sure that the oil temperature remains below 70°C[3]. The heating of the oil is mainly due to internal leakages, efficiencies and the flow over relief valves (power to heat conversion). As a general rule of thumb on cooling: if the system does not have a large accumulator in the HP line, the flow of the boost pump should be about 1/3 of the flow of the main pump. When an accumulator needs to be filled quickly, the boost flow becomes about 1/2 of the main pump flow. When the system cannot go into mooring, the boost pump flow can become even higher as the entire power is transferred to heat and needs to be dissipated via cooling.

The oil flushed out of the system by the flushing valve is cooled before it enters the reservoir. This can be done in an oil-water cooler, which is normally the most efficient and requires less

[3]Fresh oil flushed in the system from the tank is about 40°C

space. After the oil-water cooler an aqua sensor is installed, which measures the water content in the oil. When this is above 70% an alarm is given, as this might point to a internal leak in the oil-water cooler which can damage the equipment.

2.3.5 Filtering

As every hydraulic system, a secondary controlled system also requires filters. As these type of systems use servo valves, there are actually extra demands on the cleanliness of the oil, specified by the servo valves.

The boost filter just after the boost pump is the main filter of the system. The filter in this position is very effective as the boost pump delivers a constant flow and works at a constant pressure. This filter normally has a filter element of $5\mu m$. This filter normally has a by-pass and a contamination switch, which will trigger a warning once the filter is contaminated to a certain degree, but will not stop the system.

When mooring is not possible, it is possible to place a filter just after the main pump. This type of filter does not need a by-pass or a contamination switch, as the clogging can be seen at the HP pressure sensor. It is a relatively large filter ($20\mu m$) and needs to withstand the high pressure.
When the system can go into mooring, the pressure filter is hard to implement (flow from both sides is not possible for most high pressure filters).

A filter can be placed in the supply line to the servo valves, although this should be a rough filter ($20\mu m$) to prevent it from clogging. The drawback of this filter is however that it needs to be very large due to the large peak flows through the servo valves.

One last and definitely important filter is the offline filter at the reservoir. This filter helps the main boost filter and can limit the demands on the boost filter. This filter is normally very fine ($2\mu m$).

2.4 Emergency operation

The emergency operation is an important functionality of an offshore crane. Emergency hoisting is often only possible when the main pump and boost pump are running, although it might be possible to perform an emergency hoist with the use of only the piston accumulator.

Emergency lowering is often possible without the pumps running, as long as there is a header tank to add oil to the boost line.

During the emergency hoisting and lowering the swash plate must be set to the maximum angle in hoisting direction. This is absolutely necessary to be able to hold the load, also the maximum load possible. To fix the swash plate, spindles are installed on the motor, one long and one short. The long spindle, which should be at the hoisting side, must be screwed in completely to secure the swash plate in the maximum angle in hoisting direction.

The brakes of the winch are commonly connected together and are controlled by one valve. This is especially helpful for the emergency operation, as the operator does not need to lift each brake separately. To lift the brakes, a hand pump can be used. The brakes should be pressurized well above the brakes lift pressure, to make sure that they do not interfere with the procedure. The pressure should be monitored during the procedure and raised once it drops below a certain threshold, to prevent that the brakes are applied.

For the emergency hoisting and lowering there are valves installed in the motor manifold: one needle valve to regulate the speed and two ball valves. One ball valve connects the HP side of the motor to the HP line, and when the HP line is pressurized by the pumps or piston accumulator, the load is hoisted. The other ball valve connects the HP side of the motor to the boost line, relieving oil and thus lowering the load, even when no pumps are running. In order to make sure that the emergency procedures are done slowly, the needle valve should only be opened half a turn at the start of the process. During an emergency procedure the needle valve can be opened further when required.

2.5 Connecting other actuators to a secondary controlled system

It is possible to connect other systems to a secondary controlled hydraulic system. The engineer should however always consider the simultaneous operations and must ensure that the pressure at the secondary controlled winches are not dropping suddenly, as this can cause undesired motions of the load.

When the other systems are motors, the return line can be connected to the boost line, although it might be beneficial to add a separate tank line, as this reduces the back pressure in the motors due to the pressure in the boost line.

When the other system involves cylinders, it is beneficial to add a tank line. When the cylinder is connected to the boost line at the bottom side as well as the rod side, the cylinder will extend due to the pressure in the boost line and the difference in annulus and bottom area. Furthermore, the difference in flow to rod and bottom side at the same cylinder speed, requires more flow from the boost pumps (this is even worse when the cylinder is connected to a separate tank line, because now the boost pump must deliver the full flow going to the cylinder). Furthermore, the boost line pressure can influence counterbalance or load holding valves and their pressure settings, which can be undesirable.

3 Commissioning and Maintenance

Commissioning and maintenance are required for every hydraulic system. A secondary hydraulic system does have some specific points of attention.

Before starting commissioning or after maintenance, all safeties in the software need to be checked. As shown in the introduction, the system acts directly on the acceleration and thus a run away of a winch is dangerous. Especially during commissioning some parameters need to be tuned. Special attention during commissioning should go to:

- First thing to commission is the swash plate control. When this control is not working as intended, it can be heard by a rumbling sound (can be very low in decibel). A small overshoot (20%) is allowed;

- When the swash plate control is commissioned and working, the speed loop can be commissioned. Always start with a low gain. For this control loop, if not working as intended, it can produce a rumbling sound;

- The accumulator length measurement needs to be commissioned before the nitrogen system is filled, otherwise the sensor cannot be calibrated properly;

- The position of the swash plate in rest should be at the full angle towards hoisting (an exception can be for instance an application where the motor may not deliver any power in rest, such as an amusement ride). The swash plate position can be checked with the emergency hoisting procedure;

- The polarity of the encoder on the motor should be checked. This can be done during the emergency hoisting procedure. During hoisting the encoder should give a positive signal;

- The polarity of the LDVT should be checked. This can be done during the emergency hoisting procedure. During hoisting the LVDT should give a positive signal;

- The torque balance needs to be adjusted before there is any load on the winch;

- During testing of the system, it is wise to set the maximum speed to a very low speed. The speed can then be increased during testing to eventually the maximum speed.

Some of these items require the emergency procedure. The reason is that the motion can be controlled accurately and the speed is easy to control using the needle valve. It is recommended that the needle valve is opened slightly before opening the ball valve, otherwise it can require much force to open the needle valve due to the pressure drop over it.

For maintenance there are also some special points of attention:

- The servo valves need to be checked regularly, to see if they still meet the specification. For a frequently used system, this is at least once every 5 years. During this test, a flow versus signal graph is made. When the difference is more than 10% (so it needs 10% more current than state in the specifications for the same flow), the servo valve should be replaced;

- Due to the servo valves, clean oil is essential. Regularly changing the filters (in front of the servo system as well as on the HPU) is essential. Regularly taking and checking an oil sample is advised;
- Motors which leak more than 10% of the maximum flow should be overhauled or replaced;
- The magnet in the drain line near the motors need to be checked regularly. It is normal to have some pieces of metal attached to the magnet, when it is much more (a 'Christmas tree'), the motor is worn and needs to be overhauled or replaced;
- The housing of the brake needs to be emptied regularly. See the description of the brake for more information.

4 Control of a secondary controlled system

To give some insight in the controls of a secondary controlled system, Figure 6 shows the steps from start-up up to and including operation. It also shows where the operator can intervene or what happens when the operator fails to provide the system with input. As Figure 6 is only a simplified representation, its purpose is to give the reader more insight in how the basics of the controls will function. There are four basic modes which the system will stay for a longer period of time:

- Energy save mode;
- Brake applied mode;
- Speed mode;
- Tension mode.

These modes are controlled by a controller, specially designed for these systems. The Hydraulic Motion Controller (HMC), as they are popularly named, control the basics of the system and will have numerous safeties incorporated to prevent accidents due to failures.

Besides the 'normal modes of operation', there are others, such as:

- Cold state (when the system is not being used);
- Maintenance state (Maintenance is being done, some tests are allowed);
- Configuration state (mainly for commissioning, alter system parameters);

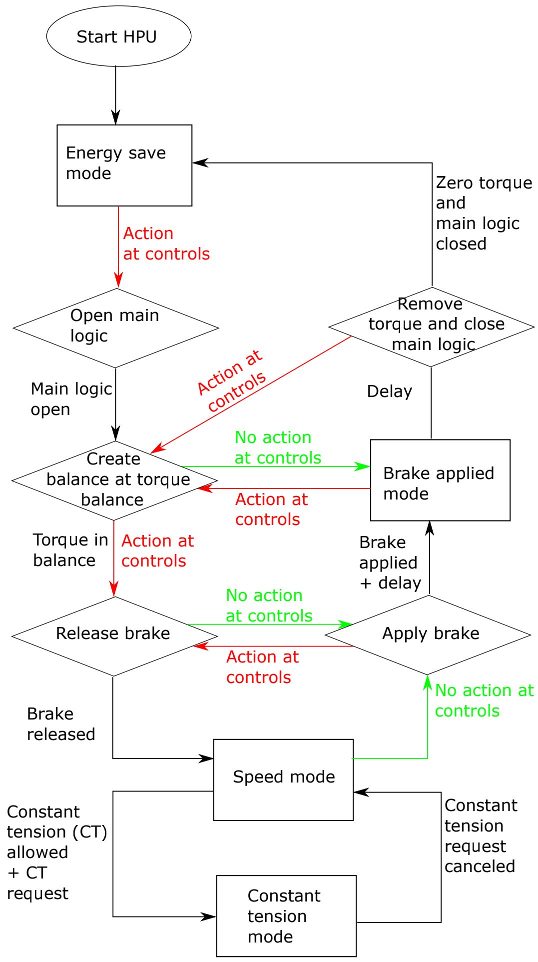

Figure 6: A simplified overview of the normal modes of operation and the steps to go from one mode to another by the controller of a secondary hydraulic actuated system.

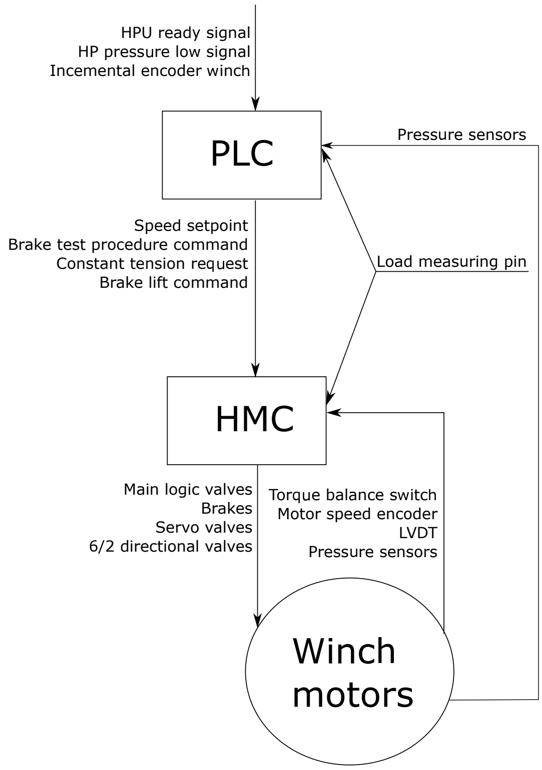

Figure 7: A brief representation of the communication between the PLC and the HMC. Signals like the "HMC Ready" are not included.

- Emergency release state (winch will pay out wire under low tension, as motor will act as pump);
- Stop state (state to stop all motion from normal operation mode before going to cold state).

The Stop state has four subtypes (normal stop, total stop, fast stop and back-up stop) to make sure the winch can always be stopped, even during failures.

When more than one motor is used, the motors can also be connected to the same HMC, depending on the brand.

4.1 Inputs to the Control Loops

In Figure 6 several steps are shown. It was made clear that the basic functions of the system are controlled by the HMC. Figure 7 shows how the user controlled PLC and the HMC communicate. For simplicity, all the checks to and from are not shown. For example, the HMC lets the PLC know that it is ready, before the PLC starts sending signals, but that would make the diagram a lot more complicated, and some of the signals are manufacturer specific. Figure 7 does show however which inputs are used by what controller. The HMC controls the logic valves, brakes and servo valves, which are the most critical for the system. This way the HMC is capable of intervening during a failure, preventing dangerous situations. The PLC gets signals from the operator and for example data on the movements of the vessel or from the incremental encoder on the winch. The PLC then translates these signals to a speed set point or tension set point, to make the system do as the operator commands.

Starting at the loop of the HMC, it is shown that it has four main inputs: the torque balance switch, motor speed encoder, Linear Variable Differential Transformer (LVDT) and the measurement by the load pin.

The torque balance switch is used to confirm that the torque produced by the motor and the load are in equilibrium, before the brakes are lifted. If this is not the case, the load would be accelerated immediately after lifting the brakes. This is considered to be dangerous and uncontrolled, as it is not known how large the difference in torque is.

The motor speed encoder measures the speed of the hydraulic motor. It is used in the speed loop, which will be explained in section 4.2.2. The encoder is most often placed on the back of the hydraulic motor, although this must be taken into account when ordering the motor. A positive signal is preferably used in hoisting direction. With the encoder measurement, a closed loop control of the motor is possible, which makes the system more accurate. This does however mean that a failure of this sensor can make the system become unstable. This single point of failure is normally mitigated by adding a back-up encoder, which signal is compared with the main encoder. When the two sensors are not in agreement (within a certain band width), the system is stopped immediately. The back-up encoder is normally not placed on the same axes as the primary sensor, as this would result in a new single point failure of the axes. For some systems it is however acceptable to do so. The back-up encoder should not be placed on a slow rotating axes, as this would make the resolution of the back-up encoder too large and make the check of the main encoder not accurate enough, resulting in a back-up encoder that is not functioning properly. For systems with multiple motors, the back-up encoder can be put on the other motor. For systems with multiple motors the control can become fuzzy due to the stiffness between the motors (gear boxes, gear ring), especially when the own frequency of the components is close to the control frequency. If so, the controls start to respond to the resonance in the system. When the back-up encoder, an absolute encoder or switch is used at the back of the encoder, the encoder should have a Rotex coupling to ensure its durability.

The LVDT measures the swash plate angle, which is used in the servo control loop, as will be explained in section 4.2.1. This servo loop is used in the speed as well as the tension loop, and is therefore a very important sensor in the control of a secondary controlled hydraulic system. However, this sensor does not need to be very accurate, as a deviation would be compensated for in the speed or tension control loop.

The load measuring pin measures the load somewhere in the mechanical system. For cranes this load pin is used at one of the sheaves. The load is then used for the tension control loop in the HMC. The signal of the load measuring pin is used in the PLC as well, to monitor whether the system is still functioning as expected.

The pressure sensors are used by the HMC and PLC to view the system performance, detect failures, to give the operator more feeling with the system and/or for trouble shooting.

4.2 The Control Loops

The control loop of the HMC has several inputs, which can be sensors as explained above, or commands and set points given by the PLC. As shown in Figure 6, there are several modes, each with its own separate control loop. There are three control loops that are described in this article: the servo, the speed and the tension control loop. The servo control loop is the control loop of the swash plate and is used in the other two control loops.

As explained in section 1.3, the flow demand can also be too large for the system's HPU, draining the accumulator and lowering the system pressure. When this happens, the HMC can also go to "reduced speed" mode. In that case the HMC will not allow larger speeds than a certain limit. This has of course influence on for instance the heave compensation, but a system that stops is most often worse than a system that continues on a lower speed. Therefore, the requirements of the system must be critically examined, to prevent the system from going to reduced speed during normal operations.

4.2.1 Servo control loop

The servo control loop forms the basis of the speed and tension loops. This loop consists of the LVDT measuring the swash plate angle and the servo valve controlling it. Assuming that the pressure in the pressure line is approximately constant, the swashplate controls the torque that is delivered by the motor. With the motor torque and the torque induced by the load, the winch is held in equilibrium or accelerated.

The servo control loop has one controller: the swashplate position controller. This is a PID controller, to make sure it has no offset error.

4.2.2 Speed Control Loop

The speed control loop, as shown in Figure 9, controls the swash plate based on the speed of the motor or winch (depending on where the speed encoder is placed, but normally this is on the hydraulic motor). As explained in section 4.1 the encoder requires a back-up, to mitigate the effect of a failure of the sensor. In this control loop the failure is made visible: as the encoder gives

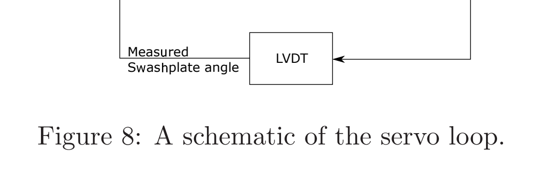

Figure 8: A schematic of the servo loop.

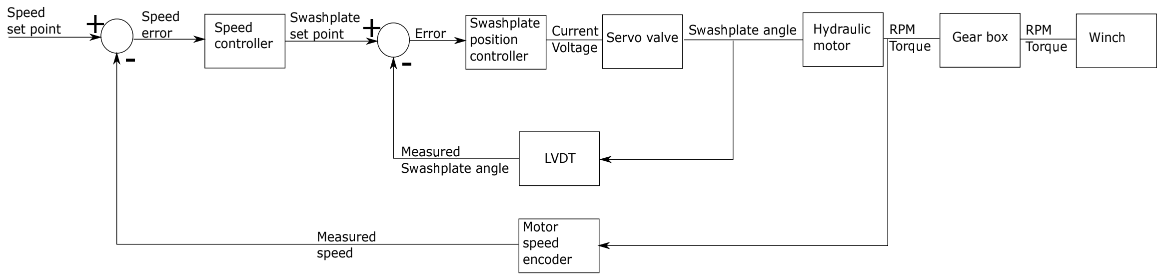

Figure 9: A schematic of the speed control loop, including the servo loop.

a large deviation of the actual speed, lets assume no speed, then the error is as large as the desired speed, meaning that the swash plate is controlled to more torque, accelerating the winch.

The speed control loop has, besides the swashplate position controller, a speed controller. The speed controller converts an error of the speed to a new set point for the swashb plate.

To calculate the new swash plate angle, it is needed to determine how much time is required to get to that speed, as this determines the required acceleration. This acceleration ($\dot{\omega}_{winch}$) is limited by the maximum acceleration based on the load and is normally a requirement of the winch.

$$\dot{\omega}_{winch} = \frac{\omega_{error}}{t} \tag{5}$$

An explanation of the variables is given in table 4.2.2. Now that the angular acceleration is known, the required resulting torque at the winch is calculated (still neglecting the gear ratio):

$$T_{res} = \dot{\omega}_{winch}\, J_{winch} \tag{6}$$

With the resulting winch torque, the required torque of the motor can be calculated assuming linear response of the motor to changing swash plate angle (assuming one motor, otherwise the amount of motors should be taken into account):

$$T_{motor} = T_{res} + T_{load} \tag{7}$$

For simplicity the gear ratio is already included in the T_{motor}, as this is only aimed at giving more insight into the dynamics. Now that the torque is known, the swash plate angle can be calculated:

$$\alpha = \frac{T_{motor}\, 20\, \pi\, \alpha_{max}}{\Delta p\, V_{sg\,max}} \tag{8}$$

This means that the swash plate angle (for a system with one motor) will be determined by:

$$\alpha_{i+1} = \frac{\left(\frac{\omega_{error}}{t}\, J_{winch}\right)\, 20\, \pi\, \alpha_{max}}{\Delta p\, V_{sg\,max}} + \alpha_i \tag{9}$$

Table 4.2.2 **Explanation of variables**

α	Swash plate angle in $[rad]$
α_{max}	Maximum swash plate angle in $[rad]$
ω_{error}	Error on the angular velocity of the winch in $[\frac{rad}{s}]$
$\dot{\omega}_{winch}$	Acceleration of the winch in $[\frac{rad}{s^2}]$
J_{winch}	Moment of inertia of the winch in $[kg\, m^2]$
Δp	Pressure difference over the motor in $[bar]$
T_{load}	Torque on the winch caused by the load in $[N\, m]$
T_{motor}	Motor torque at the winch in $[N\, m]$
T_{res}	Resulting torque in $[N\, m]$
t	Time required to close the velocity error in $[s]$
$V_{sg\,max}$	Maximum volume displacement per rotation in $\left[\frac{cm^3}{rev}\right]$

One last remark on the speed loop: lowering an empty hook will require the hook to be heavy enough to overcome all friction in the mechanical system. A wire cannot push, so an insufficient mass of the hook will mean a slack rope. In such a case, a tugger line is needed to lower the empty hook.

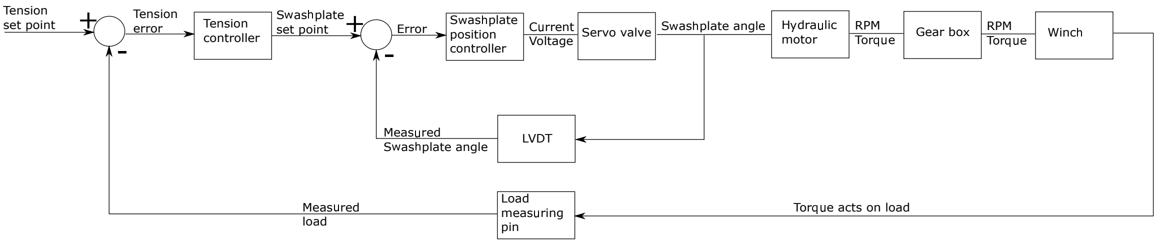

Figure 10: A schematic of the torque control loop, including the servo loop.

4.2.3 Tension control loop

The difference between the speed control loop and the tension control loop, is most visible in the feedback loop where the load measuring pin is used. The swash plate error now needs to be determined on the error in torque. This means that the load needs to be translated into a torque at the winch. The torque would contain the distance between the wire and the point of rotation and therefore includes which layer of the drum the wire is. Using the measured torque and the set point torque, the error is calculated to a swash plate angle by the tension controller. This is similar to the speed control set point. The error will then be based on:

$$\alpha_{i+1} = \frac{T_{error}\, 20\, \pi\, \alpha_{max}}{\Delta p\, V_{sg\ max}} + \alpha_i \qquad (10)$$

where T_{error} is the error on the torque in $[N\, m]$.

4.3 Requirements for Safe Operation

As shown before, there are some safety features required for safe operation. The most important one demands that the system can be stopped in case of a failure. This also means that the operation in which the system is used, must allow the system to stop.

There are four different kinds of stops to be recognized: The normal stop, total stop, back-up stop and the fast stop.

4.3.1 Normal stop

For the normal stop, the system goes to speed mode mode with a set point of 0rad/s and applies the brakes. This is shown in Figure 6 as the Brake applied mode. This stop is the same as when the operator stops the system. Failures that can cause a normal stop, are:

- HMC temperature failure
- HMC board supply failure
- Valve supply failure
- Failure in wiring of back-up servo
- Supervisory circuit failure

4.3.2 Total stop

The total stop is the smoothest of the emergency stops. All causes for this stop have a redundant measure, so there is no direct danger. However, due to the failure the redundancy is gone, thus the system is stopped. For example, during a total stop the back-up servo can take over for a main servo with an electrical wire failure and thus has no communication anymore with the control system. After the total stop the brakes are applied. Failures that can cause this stop, are:

- Pumps running failure
- Main servo wire failure
- Cylinder release valve wire failure
- Bypass valve wire failure
- Brake immediate command (although this means a less smooth stop)

4.3.3 Fast stop

The fast stop is used for failures that can lead to uncontrolled winch motion, over speed or over acceleration. During a fast stop the main logic of the motors are closed, the swash plate of the motors is set to full swash plate angle in hoisting direction and the brakes are applied. Causes for a fast stop are:

- Emergency stop (due to pressing of emergency button)
- Over speed or over acceleration
- Logic valve failure
- Brake failure
- Brake applied failure (sensors show that the brake is applied while running the system)
- LVDT failure
- Swash plate failure
- Encoder failure

4.3.4 Back-up stop

The back-up stop is used during a total or fast stop and when there is a high pressure failure; the system pressure is below the HP low pressure setting. During this stop, the back-up servo takes over all motors, stops the movement and the brakes are applied afterward. Then the system shuts down completely to make sure that the system cannot restart automatically and operator intervention is required to investigate why this failure appeared.

References

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