POLITECNICO DI TORINO

MASTER's Degree in Automotive Engineering

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Simulation of Hydraulic Motor Performance in Hydrostatic Transmission Under Various Operational Scenarios

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Abstract

This thesis aims to analyze and simulate the operation of a hydraulic motor within a hydrostatic transmission, a system widely used in industrial and mobile applications for its efficiency and versatility. By utilizing advanced simulation software like Simcenter Amesim, the main components of the transmission have been modeled. The study has been focused in particular on the motor control systems, studying the performance of the hydrostatic transmission analyzing the main parameters. The study starts with a general introduction about the hydrostatic transmissions and then continues with a brief general description of hydraulic motors. Then has been analyzed in detail the specific hydraulic motor used for this activity, with particular attention to control system. After that has been explained how has been assembled the complete model of the transmission used to perform various simulations with the software Simcenter Amesim.

The first simulations have been performed simulating the circuit without a vehicle (usually an off-highway vehicle) in order to verify if the circuit and the motor controls works properly. This result has been obtained performing simulations with variable loads. The last simulations have been performed with the vehicle simulating different working conditions. First of all the vehicle on a flat ground and then analyzing the system on a slope, both positive and negative.

This thesis wants to highlight the advantages of this type of transmission, but also the drawbacks looking for the identification of potential areas of improvement. This research also contributes to a better understanding of the dynamics of hydraulic motors.

Acknowledgements

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Chapter 1

Introduction

1.1 Hydrostatic transmissions

The transmission is a key component in every vehicle, playing a crucial role in transforming the available power supplied by the engine to meet the speed and torque requirements of the vehicle. Classified as a continuously variable transmission, according to ISO standards 5598 and 4391, a hydrostatic transmission exists at any time that a hydraulic pump is connected and dedicated to one or more hydraulic motors. Versatility is achieved by making either or both the pump and motor(s) variable displacement. This kind of transmission is preferred over shifted gear transmission in many cases because of the stepless way in which the hydrostatic transmission's speed ratio can be changed.

As it's represented in Fig.1.1, hydrostatic transmission has torque characteristics more similar to the ideal traction hyperbola, which means that the use of hydrostatic transmission allows a greater range of use, and it can operate in scenarios not achievable with mechanical transmission.

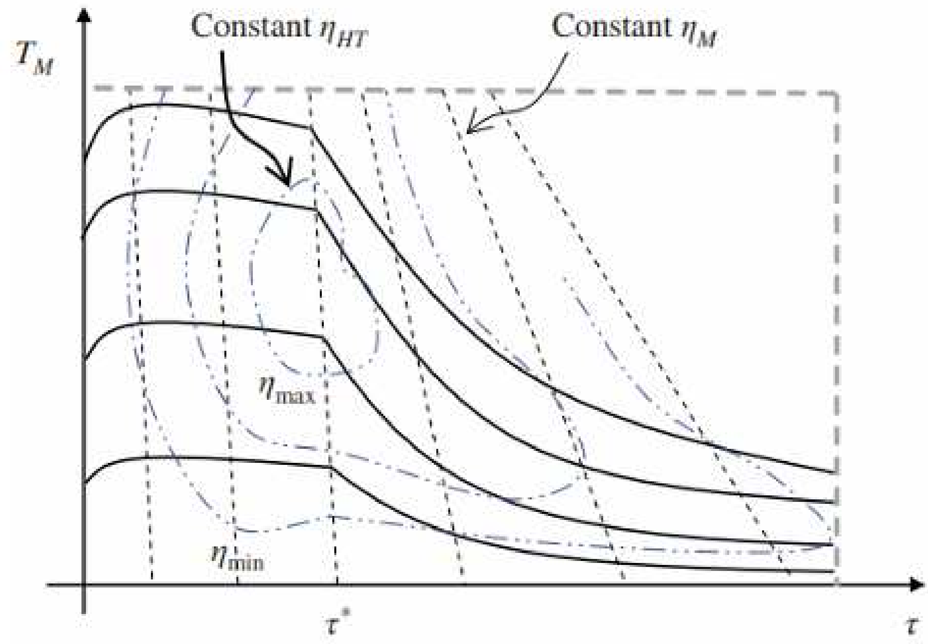

Figure 1.1: Torque characteristic of a hydrostatic transmission

At present, in the construction machinery market, most medium-sized and large loaders use hydro-mechanical transmission systems. In particular, hydrostatic transmissions are equipped in off-highway vehicles.

This kind of solution can assume two different configurations:

- closed circuit;
- open circuit;

The one considered in this thesis is the first one.
A hydrostatic transmission is composed of two hydraulic units (HU), usually, the unit connected to the prime mover is referred to as the primary unit, the one connected to the load is called secondary unit. Both units can be fixed displacement units or variable displacement units, but to be considered a continuously variable transmission, at least one of them must be a variable displacement unit.

In conclusion, a hydrostatic transmission can have four configurations combining fixed or variable displacements of the two hydraulic units. To ensure proper internal lubrication flow, both units must have external drains that are directly connected to the tank. The fluid lost through the drains could be detrimental to the system, because it reduces the pressure in the low-pressure line and the pump would work under severe cavitation. To avoid this scenario a hydrostatic transmission must be equipped with a charging circuit which imposes a minimum pressure in the low-pressure line, composed of:

- Charging pump;
- Relief valve;
- Two check valves;

It is also necessary to add two relief valves to limit the maximum pressure in the high-pressure line, and a flushing circuit for the proper cooling of the transmission, composed of a relief valve and a 3/3 directional control valve. A general scheme can be observed in Fig.1.2.

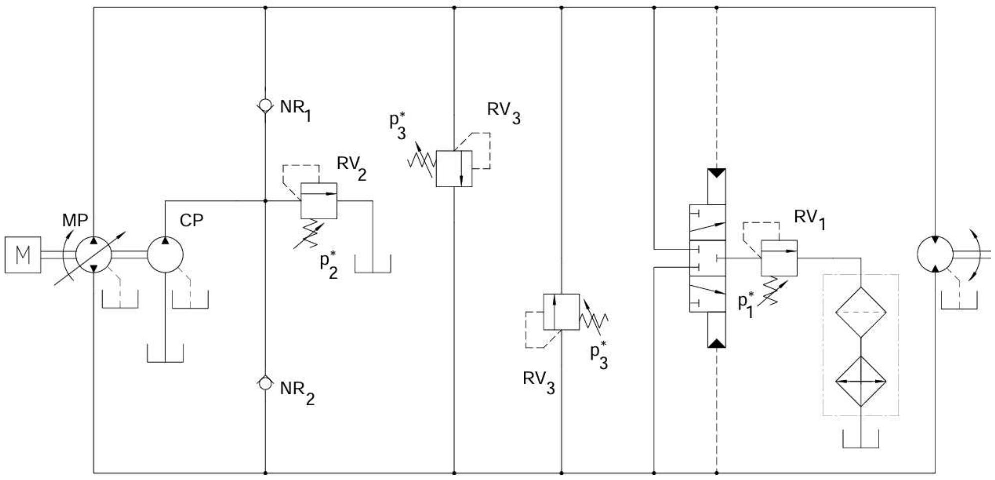

Figure 1.2: General scheme of a hydrostatic transmission[1]

A hydrostatic transmission can work in four working modes, as can be observed in Fig.1.3, which refers to a closed circuit system.

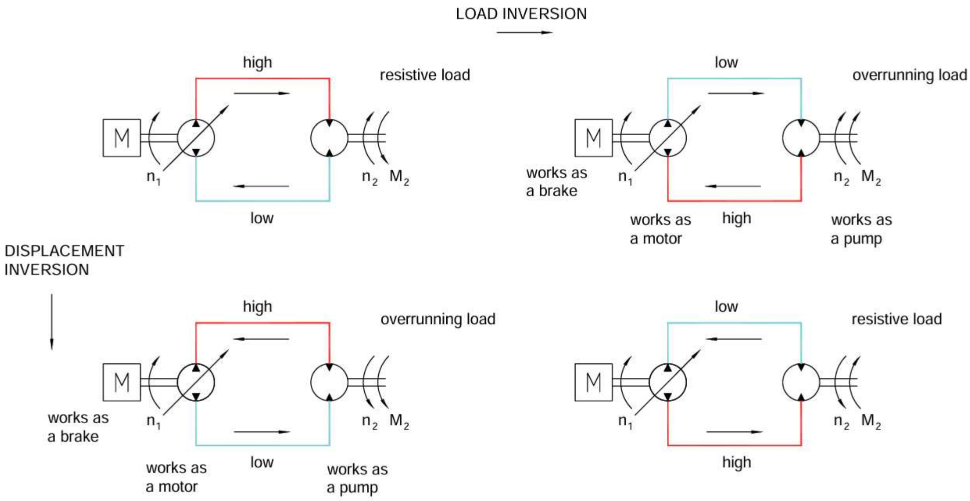

Figure 1.3: 4 working modes of a hydrostatic transmission[1]

Observing the Fig. 1.3 can be noticed how a displacement inversion changes the loop, in fact the pump will act as a motor and viceversa, whereas a load inversion affects the HP and LP sides.

Hydrostatic transmissions can have a flow coupling or a pressure coupling of the hydraulic units, which means that the flow delivered by the primary unit is received by the secondary unit for the first case, whereas it means that the torque delivered by the pump is the same delivered by the motor. Obviously this can happen only if the hydraulic units are considered ideal and not considering the various losses or leakages along the circuit. Considering the case with variable primary unit and variable secondary unit the 1.1 refers to the flow rate coupling case, the 1.2 refers to the pressure coupling case.

$$V_1 \cdot n_1 = \alpha_2 \cdot V_2 \cdot n_2 \tag{1.1}$$

$$V_1 \cdot \Delta p_1 = \alpha_2 \cdot V_2 \cdot \Delta p_2 \tag{1.2}$$

Where:

- V_1 is the primary unit displacement;
- n_1 is the rotational speed of primary unit;
- α_2 is the fractional displacement of secondary unit;
- V_2 is the secondary unit displacement;
- n_2 is the secondary unit rotational speed;

- Δp_1 is the pressure drop between suction and delivery port of primary unit;
- Δp_2 is the pressure drop between suction and delivery port of secondary unit;

Taking into account this two equations is clear how, ideally speaking, the relations between the quantities exchanged by HU depends only from the coefficient α_2. With the following hypothesis:

- n_1=*const.*
- Δp_1=Δp_2
- V_1=V_2

It is possible to introduce the concepts of transmission ratio in (1.3) and torque ratio (1.4).

$$\upsilon = \frac{n_2}{n_1} = \frac{\alpha_1 \cdot V_1}{\alpha_2 \cdot V_2} = \frac{\alpha_1}{\alpha_2} \tag{1.3}$$

$$\tau = \frac{M_2}{M_1} = \frac{\alpha_2 \cdot V_2 \cdot \Delta p_2}{\alpha_1 \cdot V_1 \cdot \Delta p_1} \tag{1.4}$$

In 1.4, α_1 is the fractional displacement of primary unit.

$$\eta_{HT} = \frac{P_2}{P_1} = \tau \cdot \upsilon \tag{1.5}$$

The 1.5 is the equation to calculate the overall efficiency of a hydrostatic transmission.

1.2 Hydraulic motors

A hydraulic motor is a machine able to convert the hydraulic power delivered by the pump, into mechanical power.

These motors are robust, reliable with minimal noise emissions and maintain high efficiency in medium and high pressure applications.

The application of these motors, due to their good performance, the ease to vary the rotational speed, the simplicity to reverse the movement direction, and the ease for installation, has a large use in many industrial sectors, in particular in the agricultural sector. This solution allows to have high torque with reduced encumbrances, where a mechanical solution could be difficult or the usage of electric power is not permitted or dangerous for specific reasons due to the working environment. In general all ideal hydraulic motors are governed by the following equations:

$$Q = V \cdot n \tag{1.6}$$

$$M = V \cdot \Delta p \tag{1.7}$$

$$P_{us} = M \cdot n \tag{1.8}$$

$$P_{exp} = Q \cdot \Delta p \tag{1.9}$$

The 1.6 calculates the flow rate in l/min as the product of the motor displacement per revolution (V) multiplied by the rotational speed (n). In 1.7 the torque (M) is the result of the product between V and the delta pressure between the inlet and outlet port of the motor. The 1.8 refers to useful power and it's the torque multiplied by rotational speed, whereas the expended power in 1.9 is the product between flow rate and delta pressure. Taking into account that real motors are not ideal, the concepts of volumetric efficiency and mechanical-hydraulic efficiency can be introduced. The volumetric efficiency for a hydraulic motor is given by the theoretical flow rate divided by real flow rate:

$$\eta_v = \frac{Q_{th}}{Q_R} \tag{1.10}$$

In a hydraulic motor the real flow rate is higher than the theoretical one. The mechanical-hydraulic efficiency described in 1.11, is the ratio between the real torque and the theoretical torque and the real torque is always lower with respect to the theoretical one.

$$\eta_{mh} = \frac{M_R}{M_{th}} \tag{1.11}$$

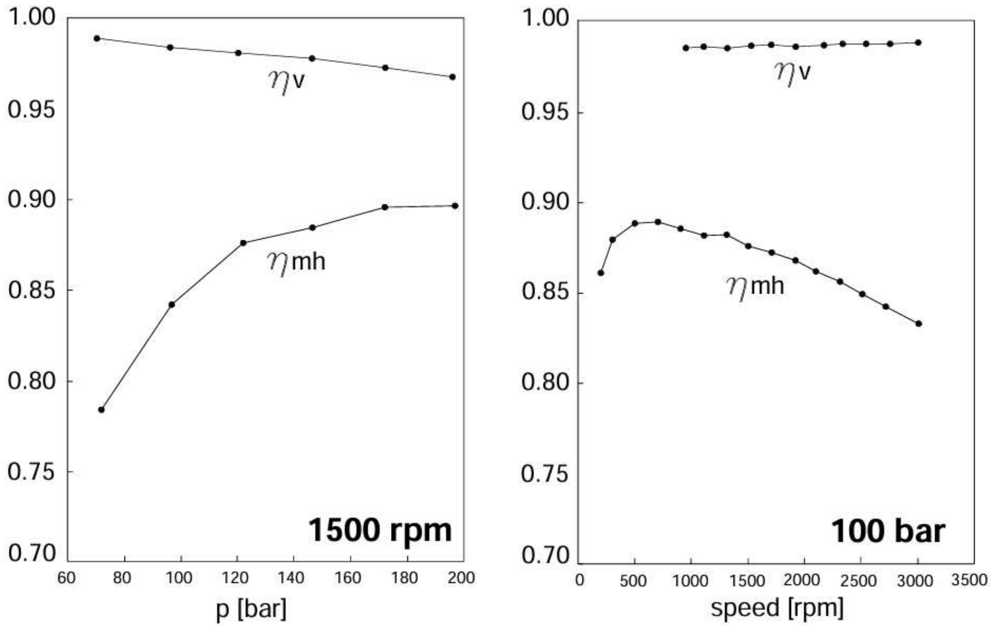

Figure 1.4: Trend of efficiencies for a generic hydraulic motor[1]

The product of the two efficiencies gives the total efficiency of the motor.

$$\eta_t = \eta_v \cdot \eta_{mh} = \frac{P_{us}}{P_{exp}} \tag{1.12}$$

where P_{us} is the useful power and P_{exp} is the expended power. Looking at Fig.1.4, with constant velocity and an increment of pressure the volumetric efficiency tends to decrease because of the laminar model of leakage flows whereas with constant pressure and variable rotational speed, the volumetric efficiency grows up to a certain level and remains more or less constant.

For what concern the mechanical-hydraulic efficiency, with constant rotational speed and variable pressure it starts to grow from a value of pressure obviously higher than zero and achieves a saturation value with big values of pressure. Looking at the second plot is evident that mechanical-hydraulic efficiency is affected a lot by the rotational speed because of the friction losses which becomes more severe at high speeds.

1.2.1 Types of motors

It's possible to distinguish the hydraulic motors into three categories depending on the device that permits the variation of the displacement volume (piston, hydraulic gear, hydraulic vane). In particular for the first category of piston motors, it is possible to do a subdivision into axial piston motors and radial piston motors. For axial piston motors there are:

- swash plate motors (rotating cylinder block);
- bent axis motors;

For radial piston motors:

- rotating cylinder block (variable displacement)

Talking about gear motors (fixed displacement), they can be:

- external;
- internal (gerotor or orbit);

Vane motors exists only with balanced rotor and constant displacement.

1.2.2 Motor's working principle

Independently from the type of motor, the working principle is the same for all. The pressurized fluid (oil) enters in the machine through the suction port, generates a force able to move the components, leading the fluid, not pressurized now, to the delivery port and transmitting a torque to the output shaft of the motor. All volumetric machines have a proportionality between the velocity and the admissible flow rate in the suction port, whereas the pressure drop is related to the load applied to the output shaft of the transmission.

1.2.3 Bent axis motors

Bent axis motors belong to the family of axial pistons motors and it is possible to design them with fixed or variable displacement. In figure 1.5 is represented an example.

Figure 1.5: Motor Sauer-Danfoss Size 060 with variable displacement[2]

Bent axis motors belong to the family of axial pistons motors and two possible versions are available:

- fixed displacement;
- variable displacement;

They have the feature of allowing for the variation of displacement volume between the suction chamber and the delivery chamber, laying on an inclined axis, with respect to the output transmission shaft, the swash plate. The variation of the inclination of this element allows the change in the displacement volume according to the working conditions.

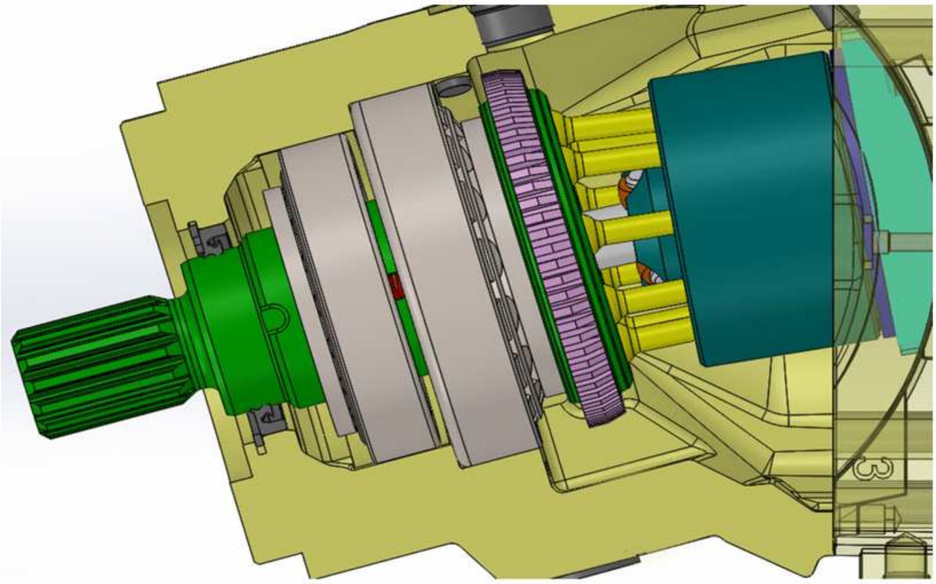

Figure 1.6: Bent axis motor with minimum displacement configuration

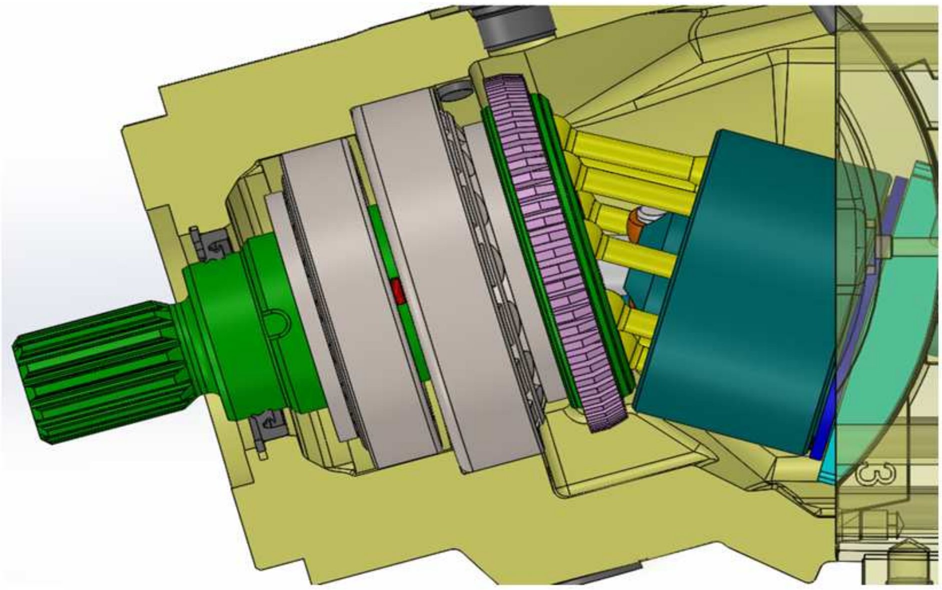

Figure 1.7: Bent axis motor with maximum displacement configuration

In figure 1.6 and 1.7, it is possible to observe the minimum and maximum displacement configurations respectively.
In the first image a lower inclination corresponds to a smaller displacement, whereas in the second, an higher inclination leads to a bigger volume.

1.2.4 Functioning of bent axis motors

The fundamental elements of these motors are the fixed distribution plate, the rotating drum where the cylinders are placed, the pistons and the transmission shaft. The suction and the delivery of the oil is obtained through the distribution plate by means of two kidney-shaped control slots, one is for the admission of the fluid whereas the other for the exit. An opposite rotation leads to a switch between the two slots.
During the first half revolution, the oil flows in the chamber and the pistons, pushed by it, apply a torque to the shaft ,thanks to a particular universal joint able to transmit the torque which allows the rotation.
In the second half the oil flows through the second slot and the piston helps the fluid to be discharged.

1.2.5 Displacement variation

As it's been observed before, the motor displacement depends on the inclination of the axis of the piston with respect to the axis of the output shaft. The more the angle increases, the more the displacement will increment and viceversa.

$$V_0 = V_{max} - V_{min} \tag{1.13}$$

$$V = V_0 \cdot N \tag{1.14}$$

In 1.13 is reported the definition of the volume of fluid that every single cylinder is able to move in a complete revolution of the shaft, instead equation 1.14, takes into account the effective volume of the motor, where "N" is the number of cylinders. The 1.13 depends on the inclination of the rotating drum with respect to the shaft. The piston stroke depends on this angle, as it could be seen in 1.7 and in 1.6, to a bigger angle corresponds a higher stroke. Theoretically speaking, it is possible to have a null angle, but this means a difference of volume between maximum and minimum, equal to 0 and so an infinite velocity of rotation.

For this reason, in real applications, the configuration with null displacement is avoided and the minimum angle is limited to a small value. The displacement affects the rotational speed and also the torque, so thanks to this feature, with a change in the geometry, bent-axis motors can adapt their behavior to every working condition.

1.3 Motor H1060 Sauer-Danfoss

In this thesis activity the object of study is the bent axis variable displacement motor produced by Sauer-Danfoss, with a displacement that ranges between 12 cm^3 and 60 cm^3 per complete revolution, which is the smallest for this series. H1 variable displacement motors are bent axis design, incorporating spherical pistons. These motors are designed primarily to be combined with other products in closed-circuit systems to transfer and control hydraulic power. Series H1 motors have a large maximum/minimum displacement ratio of 5:1 and high output speed capabilities.

Figure 1.8: Motor's photos

As it is possible to see in Fig.1.8 this model is composed by two main blocks, the first one has the output shaft, the pistons, the rotating drum and all the components needed to transmit torque, whereas the other one is smaller and hosts all the electric and hydraulic controls. In this second block there are two holes marked with "A" and "B", one for the suction and one for the drain, finally another smaller block is fixed to the last one and two solenoid are clearly visible, they are responsible for the displacement control.

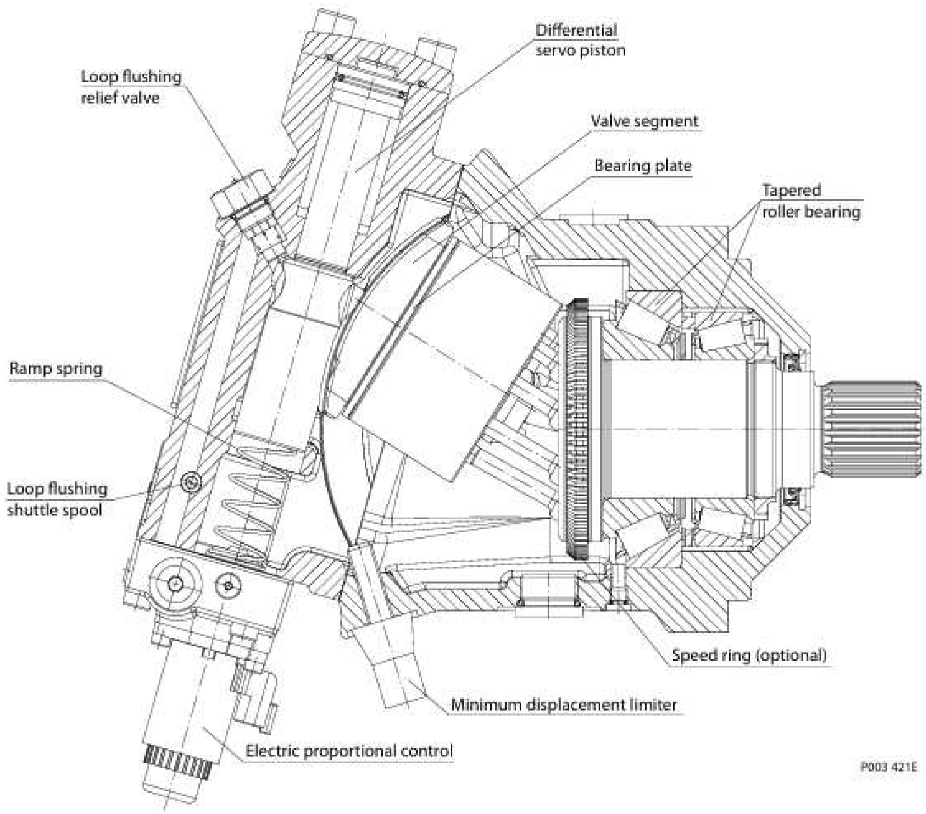

Figure 1.9: Cross section H1 electric proportional control[2]

In Fig.1.9 is shown a cross section that shows the electric control that acts in order to push the spring, and so on the linear actuator, in order to move the bearing plate, responsible for the angle of inclination and, as a consequence, for the variation of displacement.
A complete family of controls and regulators is available to fulfill the requirements of a wide range of applications. Motors normally start at maximum displacement, this provides maximum starting torque for high acceleration.

As previously said this motor is suitable to be mounted in a closed-loop system, and in Fig.1.10 an application is presented. The gray portion in Fig.1.10 highlights the motor and the controls that are the main object of study of this thesis.

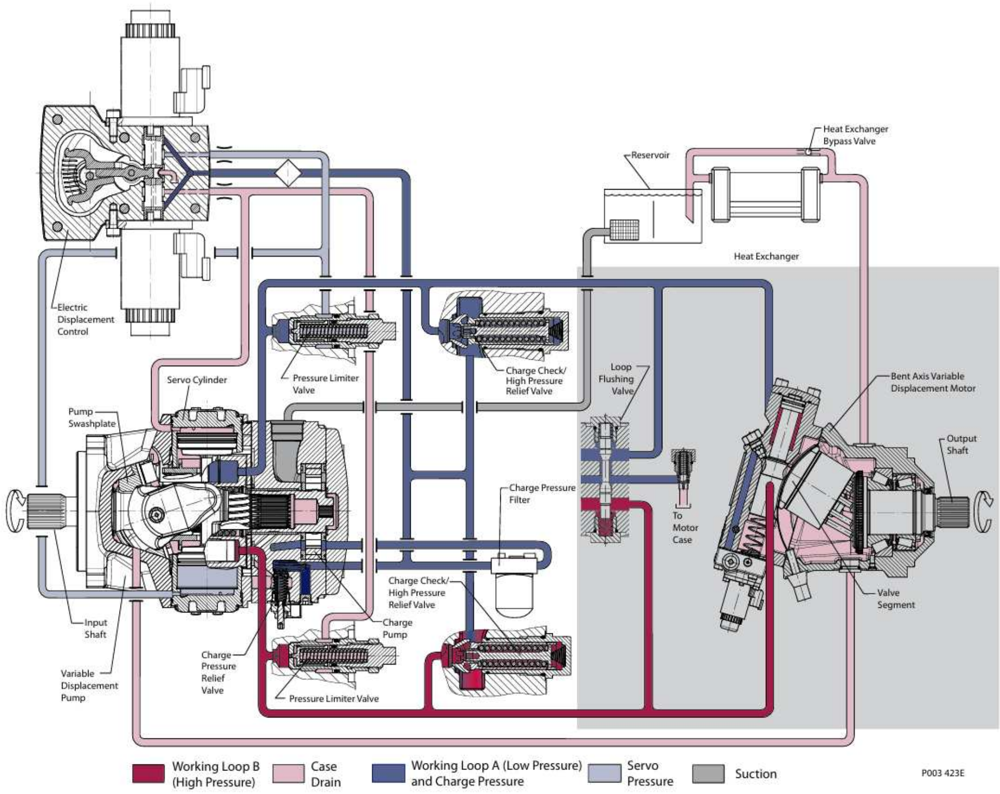

Figure 1.10: H1 Pump and H1 Motor with Electric proportional control[2]

1.3.1 Functioning

In this subsection is described the functioning of a generic bent axis motor

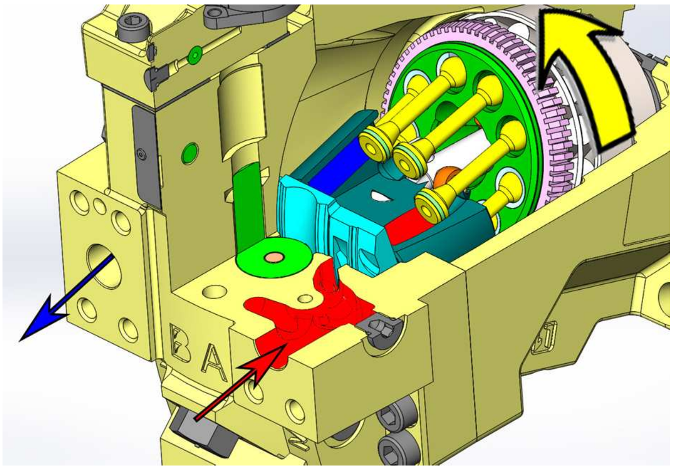

Figure 1.11: Rotation scheme of the motor H1060

Looking at Fig.1.11 it can be noticed how the pressurized fluid (in red) enters from the admission port, flows through the distribution plate, and reaches the cylinder, the action of the oil pushes the pistons towards the shaft. Instead after half a rotation the piston moves the fluid towards the plate and so to the drain (blue arrow).

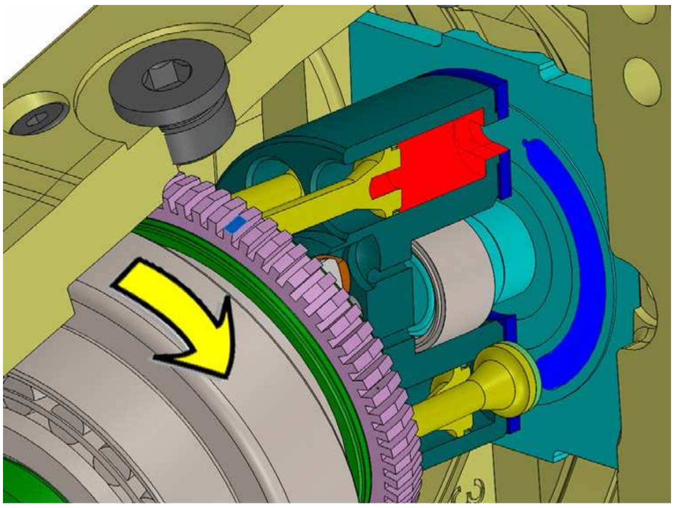

Figure 1.12: Section on the kidney-shaped control slot

In Fig.1.12 is highlighted the flow of the oil from the cylinder(red portion) to the slot which guides the fluid to the drain. The rotational motion generated by the pistons is transmitted to the shaft through a universal tripod joint shown in Fig.1.13, designed to adapt to all the inclinations reachable by the motor and to hold the maximum torque generated.

Figure 1.13: Tripod joint

At the center of the drum there is a hole where a pivot is located on which the joint shaft is supported through a conical recess, on the other side of the shaft a second pivot is inserted placed on a small plate constrained to a preloaded spring. These elements maintain the joint on its location, but contemporary allow a certain degree of freedom, in order to transmit the torque in all the intermediate configurations between maximum and minimum displacement.

In the drum there are 9 cylinders, one for each piston. The cylinders have a diameter of 1,6 cm and a stroke of 4,2 cm, calculating the volume of one cylinder, multiplied by 9, a value of 76 cm^3, but considering that zero degree configuration is not allowed, the pistons complete a stroke of 3,3 cm each. Taking into account this value, with the same procedure described above, a value of 60 cm^3 emerges, in agreement with the manufacturer. The distribution plate is crescent-shaped, and thanks to this feature it is able to slide on the main block of the motor.

1.3.2 Displacement variation

The other main block is totally dedicated to motor controls. The main output is the displacement control due to the sliding of the distribution plate on a crescent-shaped milling, this motion modifies the rotational axis of the drum, in fact the plate has a fixed pivot on which the drum can rotate.
In this way, the angle between the shaft and the drum axis changes according to the working conditions, into a range starting from 6 ° up to 32 °. In particular, with a small angle, the motor works at maximum rotational speed and mininum torque, whereas the opposite is valid for bigger angles. The component responsible for the distribution plate's position is a linear actuator placed into the secondary block, dedicated to motor's controls. The piston of the linear actuator is constrained to the plate via a pin with spherical head, which transmit the motion. According to the control input is possible to feed the chamber with bigger section in order to push the piston and bring the motor to decrease its displacement, the opposite happens when the oil fill the other side of the actuator. Usually these kinds of motor start with maximum displacement in order to reach maximum torque and so maximum acceleration.

The linear actuator is supplied through ducts cut into the motor head, which start from the inlet ports and cross the block along its entire length.

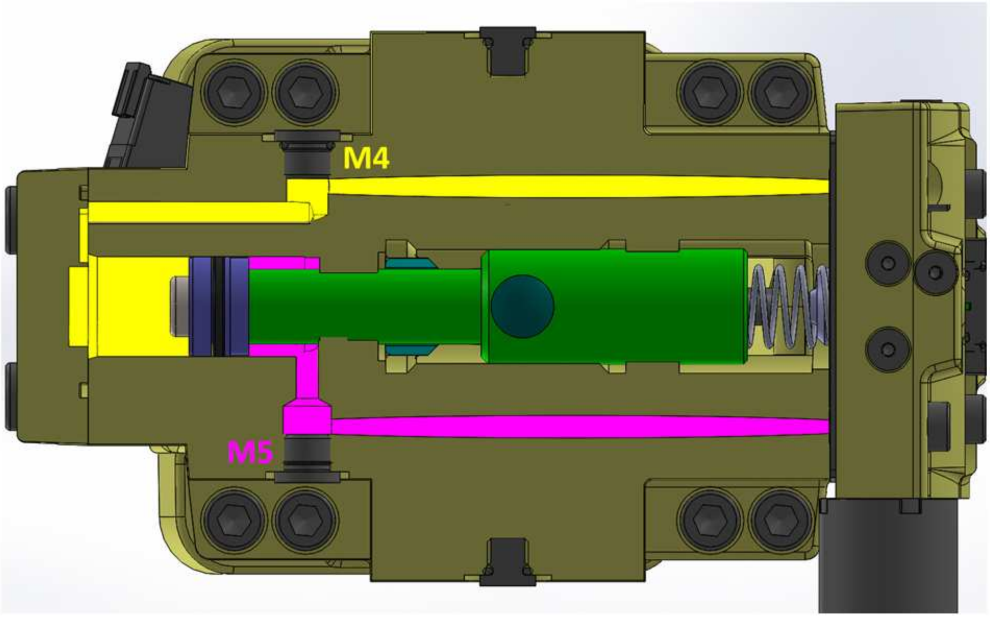

Figure 1.14: Section of motor block and linear actuator

Looking at Fig.1.14, are clearly visible two ducts, the yellow one feeds the chamber with major section, the purple one the opposite side of the piston. Moreover two ports are present, marked with M4 and M5, they are connected to pressure sensors in order to send information from the linear actuator.
The amount of oil intended for the linear actuator is regulated by a series of control valves that will be analyzed in the next section. The valves need to intercept the oil just after the inlet port of the motor, so as to be adequately supplied.

1.4 Motor controls

Sauer-Danfoss H1060 is equipped with:

- loop flushing relief valve;
- loop flushing shuttle spool;
- proportional control valve controlled by a solenoid;

Two additional control valves are also present:

- PCOR system (Pressure Compensator Over Ride);
- BPD system (Brake Pressure Defeat);

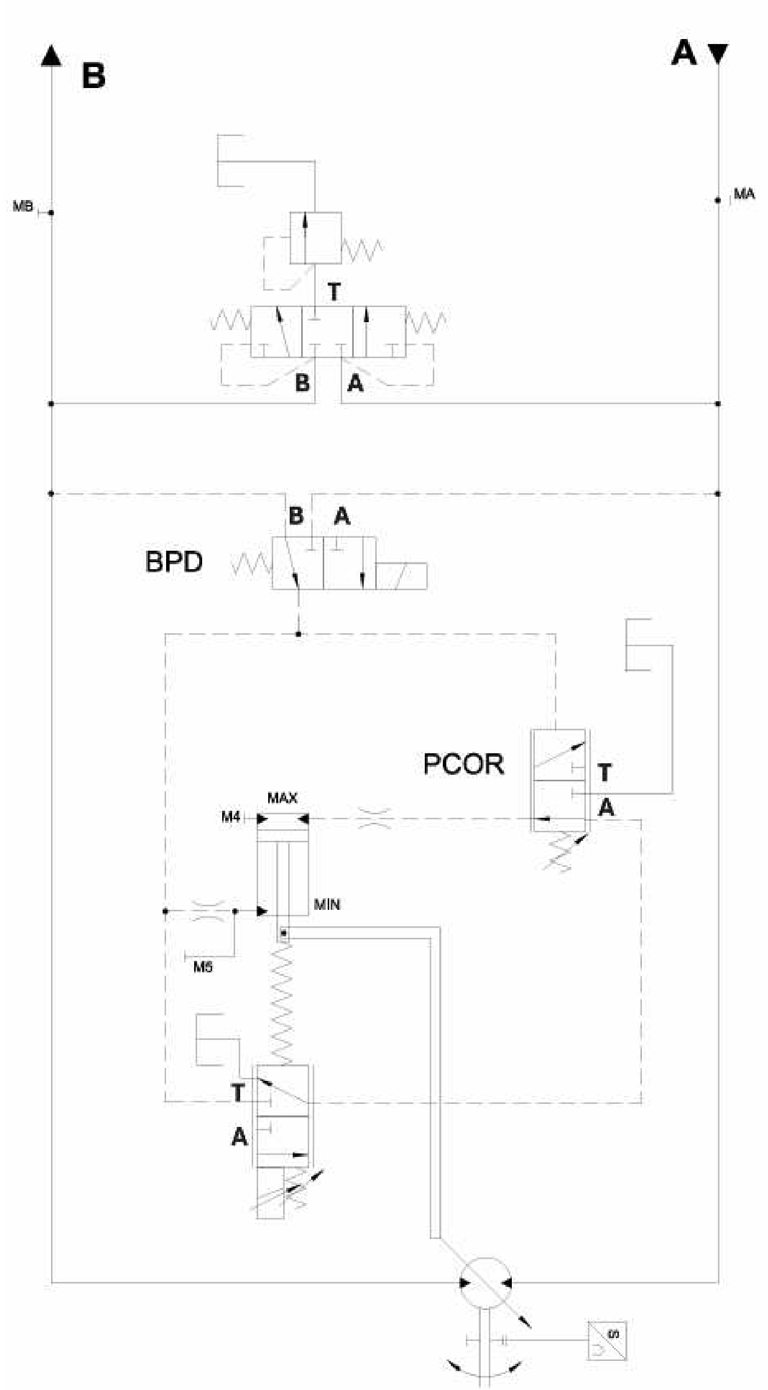

Figure 1.15: Hydraulic scheme of Sauer-Danfoss H1060[2]

In Fig.1.15 is depicted the hydraulic scheme of this hydraulic motor. A and B are the two inlet ports, the choice of the inlet port affects the direction of rotation, it

can be either clockwise or counterclockwise.
Supposing that the chosen inlet port is A, the oil flows to the drum in order to put in rotation the output shaft and to the 3-way valve too, because it has to connect the pressure relief valve with the LP side. Simultaneously the BPD valve is activated to supply the motor controls, from here the oil flows to the proportional control valve and to the chamber with minimun sectional area of the linear actuator. Depending on the command coming from the driver the oil continues to flow in order to increase the displacement or flows to the opposite chamber of the linear actuator, thanks to control valves, reducing the motor displacement. The PCOR system intervenes only if the HP pressure reaches its setting pressure, bypassing the proportional control, it forces the motor to increase the displacement in order to limit the maximum pressure and maintain it constant.
Starting from the top of the picture there are:

- a relief valve necessary to impose the maximum pressure in the LP side;
- a 3-way valve intended for connecting the LP loop with the relief valve;
- the BPD valve;
- the PCOR valve;
- linear actuator connected to the proportional valve through a preloaded spring;

At the bottom of the image, connected to the actuator, there is the motor.

1.4.1 Pressure relief valve

The pressure relief valve is a valve needed to not exceed the limit pressure on the low-pressure side of the transmission and is incorporated into all H1 motors, this valve constantly regulates. It is inserted under the ports A and B and from the outside it is observable its hexagonal head. In motor H1060 it is a cartridge valve. In Fig.1.16, the duct above with respect to the valve is connected with the drain, whereas the lower part of the valve, where the oil flows, is in communication with the inside of the block at null pressure, in turn connected with the drain.

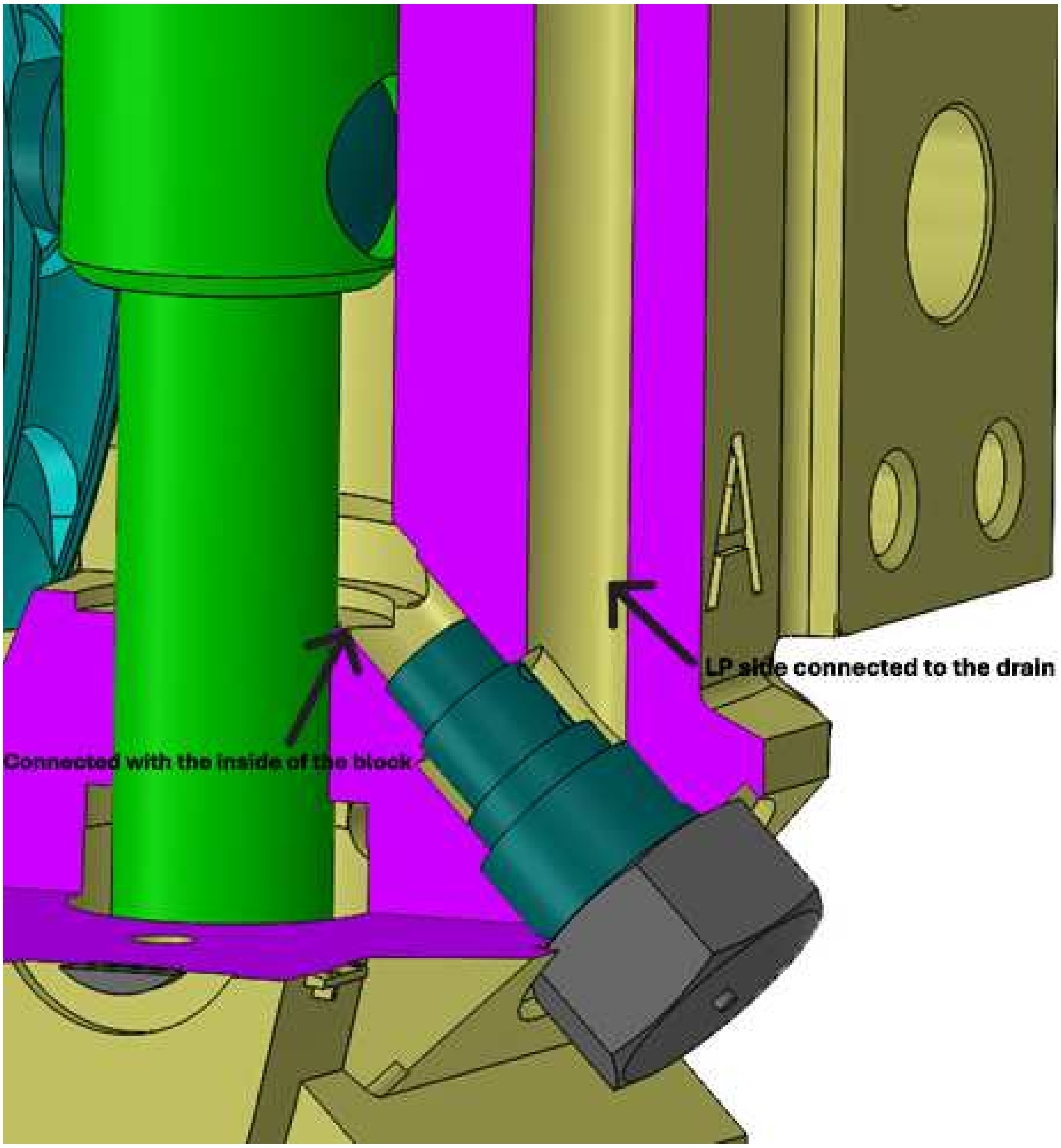

Figure 1.16: Relief valve position

Use the loop flushing option in installations that require fluid to be removed from the low-pressure side of the system circit due to cooling requirements and also used to facilitate the removal of contaminants from the loop.
The loop flushing valve is equipped with an orificed charge pressure relief valve designed with a specific cracking pressure, the parameter that imposes the pressure limit to not overcome on the LP side.

It is normally closed and when it works lets the oil flow to the tank, maintaining the pressure under the set limit. In this case the relief valve is a cartridge valve.

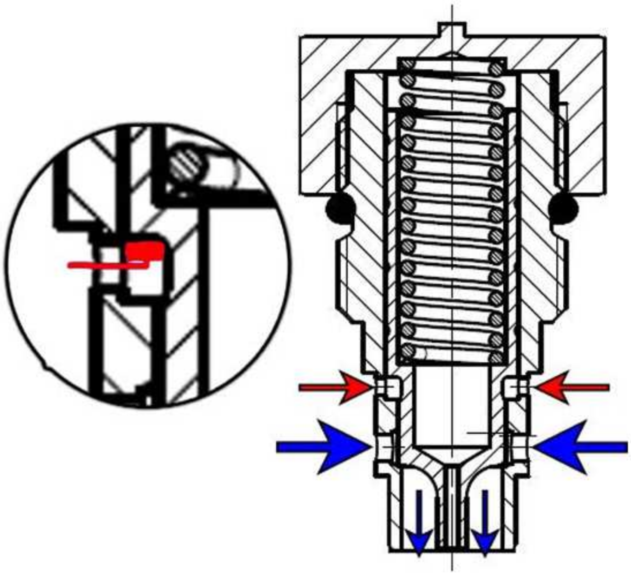

Figure 1.17: Relief valve section[2]

Regarding the functioning of this valve, looking at Fig.1.17, it is surrounded by the fluid, the oil enters from the small upper holes (red arrows) and pushing on the poppet surface, moves the internal spool. Once the cracking pressure is been reached, the preloaded spring is compressed and this movement generates a flow area between the poppet and the sleeve (blue arrows). In this way the fluid is able to flow to the drain and the value of the cracking pressure won't be exceeded.

Figure 1.18: Real relief valve

The Fig.1.18 clearly shows the two holes that allow the valve to work. The left hole is the one indicated with the red arrow in the previous picture, the right one was indicated with the blue arrow.

1.4.2 Pendulum valve

Pendulum valves are 3-way valves, used in closed circuits, that compare two input pressure signals from HP and LP sides. This system separates system A and system B pressures; this delta pressure causes the spool to shift, allowing the low side pressure to flow to the relief valve.
This valve is placed above with respect to inlet ports and its location is housed inside the motor block as can be observed in Fig.1.19.

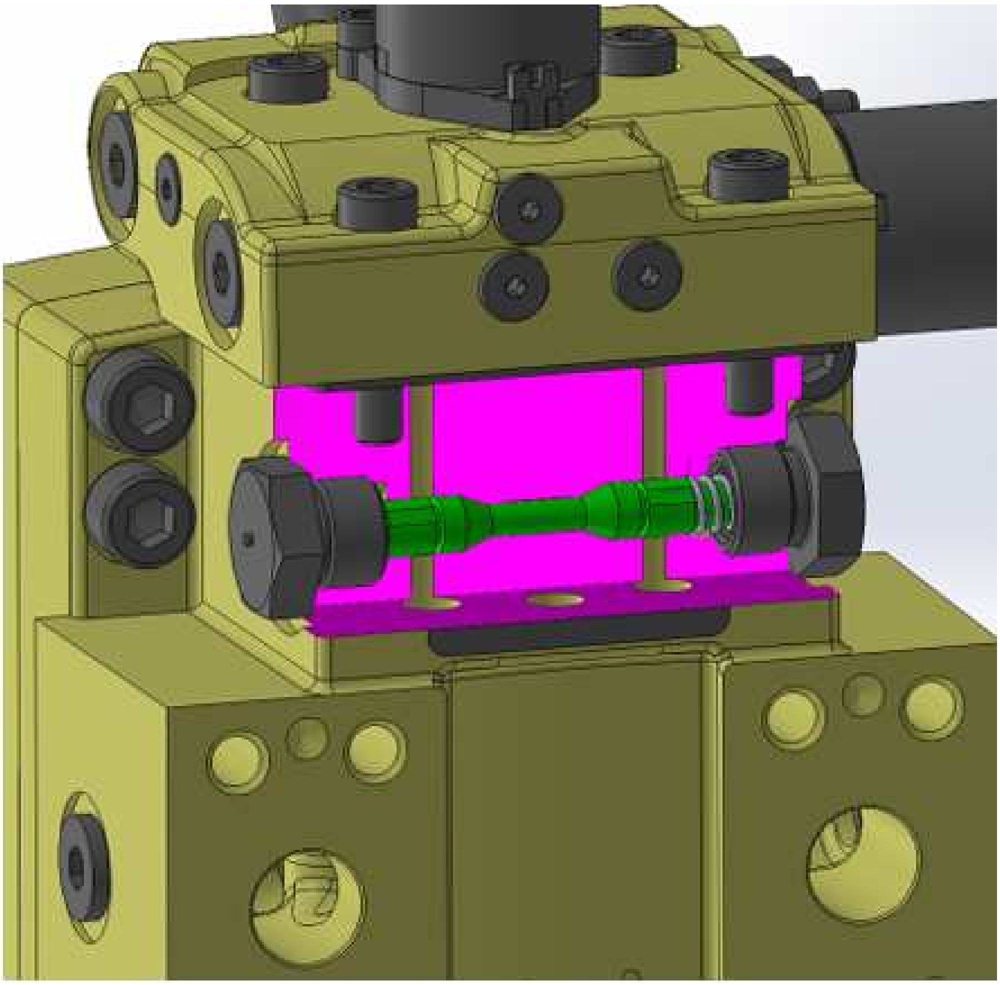

Figure 1.19: Pendulum valve location inside the motor block

Referring to Fig.1.20 and Fig.1.21, supposing that the fluid is admitted from port A (red portion), the pressurized oil will flow in the channel and, through a flattening, will be able to reach the chamber with the spring. At this point the fluid generates a thrust against the elastic force of the spring and as soon as this force is higher than the combined force from the other side (spring plus low pressure oil), the valve moves and the fluid in the blue region can flow through a third channel placed at the center of the valve, which is connected to the pressure relief valve. In this way, in the LP side, the maximum allowed pressure is the one imposed by the relief valve.
In Fig.1.22 the real valve is shown.

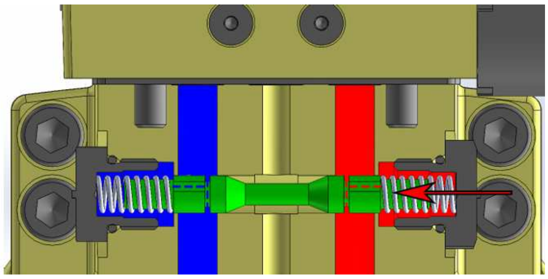

Figure 1.20: Pendulum valve in resting position

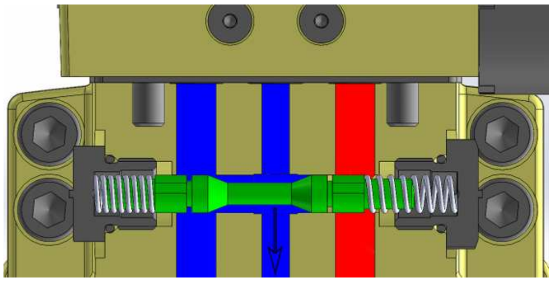

Figure 1.21: Pendulum valve in regulation

In Fig.1.22 there is a real picture of the valve with relative dimensions.

Figure 1.22: Real picture of pendulum valve

1.4.3 Proportional control valve

A proportional valve works not only in two extreme positions (open or closed), but also in all the intermediate positions between them. Because of this valve is possible to change the motor displacement according to the working conditions because it allows the oil to flow through the valve towards the chamber with maximum section area of linear actuator or to the other chamber with minimum section area.

The proportional control valve is positioned in the motor's control block, vertically arranged, and crosses it completely. As it can be seen from the Fig.1.23, on the right of the block is clearly visible the solenoid, that gives the input force to the proportional valve, fixed with three screws. The proportional valve (Fig.1.24) is responsible for direct the flow to the chamber with maximum section of the actuator or to the opposite chamber with minimum section.

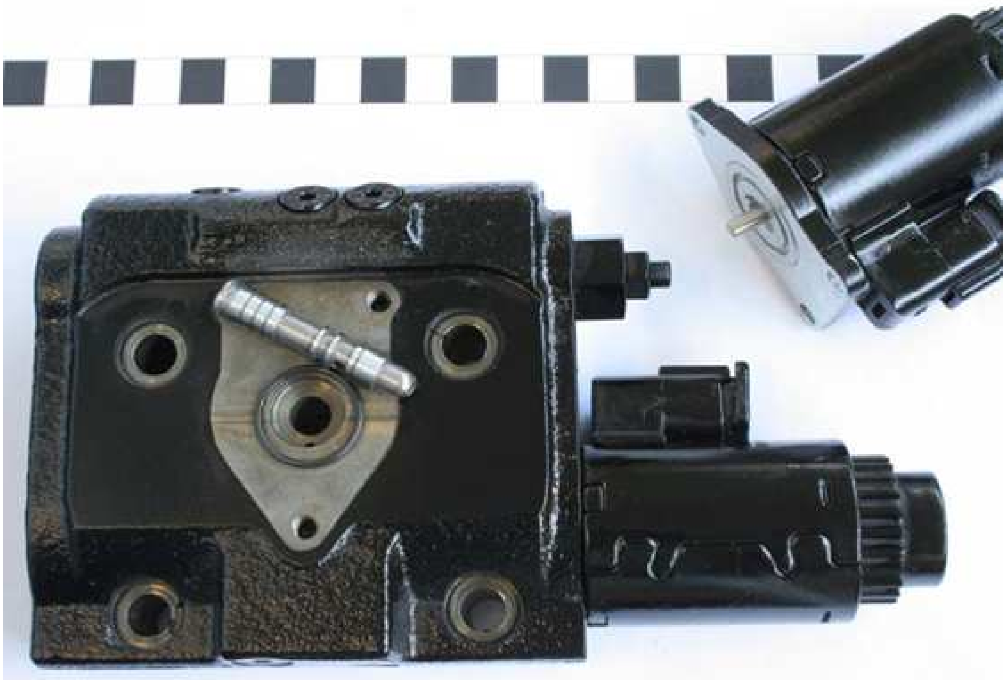

Figure 1.23: Control block

There are two possible configurations for this valve:

- De-energized = maximum displacement:
  With a de-energized to maximum displacement control, the de-energized proportional valve keeps the motor at maximum displacement. When energized, the solenoid pushes on the porting spool which moves to port high system pressure to the larger diameter end of the servo piston. Depending on the

Figure 1.24: Proportional control valve

current supplied to the proportional valve, the motor will stroke between maximum displacement at zero current and minimum displacement at maximum current.[2]

- De-energized = minimum displacement:
  With a de-energized to minimum displacement control, the de-energized proportional valve keeps the motor at minimum displacement. When energized, the solenoid pushes on the porting spool which moves to port high system pressure to the larger diameter end of the servo piston. Depending on the current supplied to the proportional valve, the motor will stroke between minimum displacement at zero current and maximum displacement at maximum current.[2]

The configuration that will be taken into account in this thesis will be the first one. A hydraulic scheme is shown in Fig.1.25.

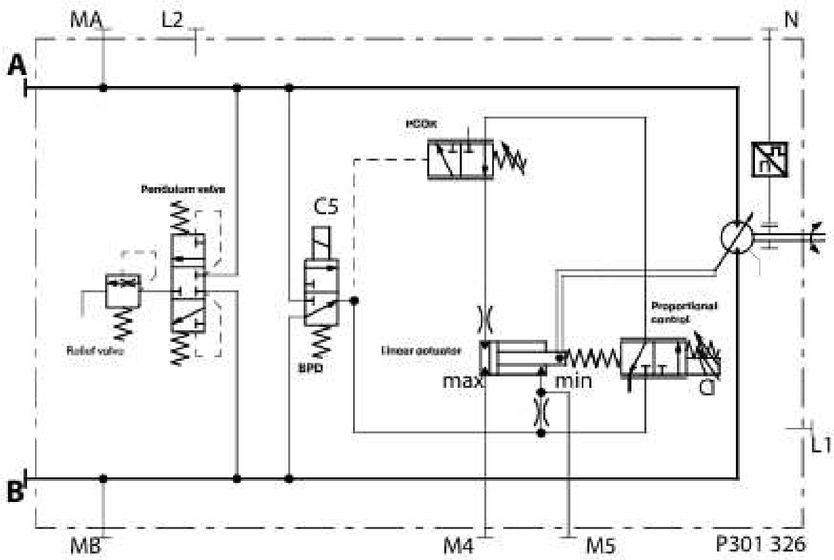

Figure 1.25: Hydraulic circuit of controls[2]

The valve, in its rest position, connects the HP side, independently from the inlet port chosen for the fluid admission, to the chamber of linear actuator with minimum section area (max. displacement), whereas the channel connected to the chamber with maximum section area goes to the drain. For this reason, in rest position, so without electric command, the oil flows in order to keep the maximum displacement. If the motor starts in this situation, the displacement remains constant at maximum value.

When the solenoid acts, gives the input force to the valve, this one is pushed down and the pressurized oil is connected to the chamber with maximum sectional area, causing the motor to decrease its displacement because of the movement of the linear actuator. Another element to highlight is that the proportional valve is connected to the linear actuator also with a spring, so as the actuator moves, it will cause a progressive increase in the force opposing the solenoid. At a certain instant the two forces will be equal and the valve will be in an equilibrium position, no one of the two chambers will be connected to the valve and, as consequence, the motor will maintain constant the reached displacement.

1.4.4 PCOR control system

The acronym PCOR stands for Pressure Compensator Over Ride, this valve allows to limit the maximum pressure on the high-pressure side, acting on the motor displacement bypassing the proportional electric control.
As soon as the pressure reaches the limit value to which the valve is been set, the PCOR intervenes by forcing the linear actuator to move in order to increase the displacement in order to control the pressure.

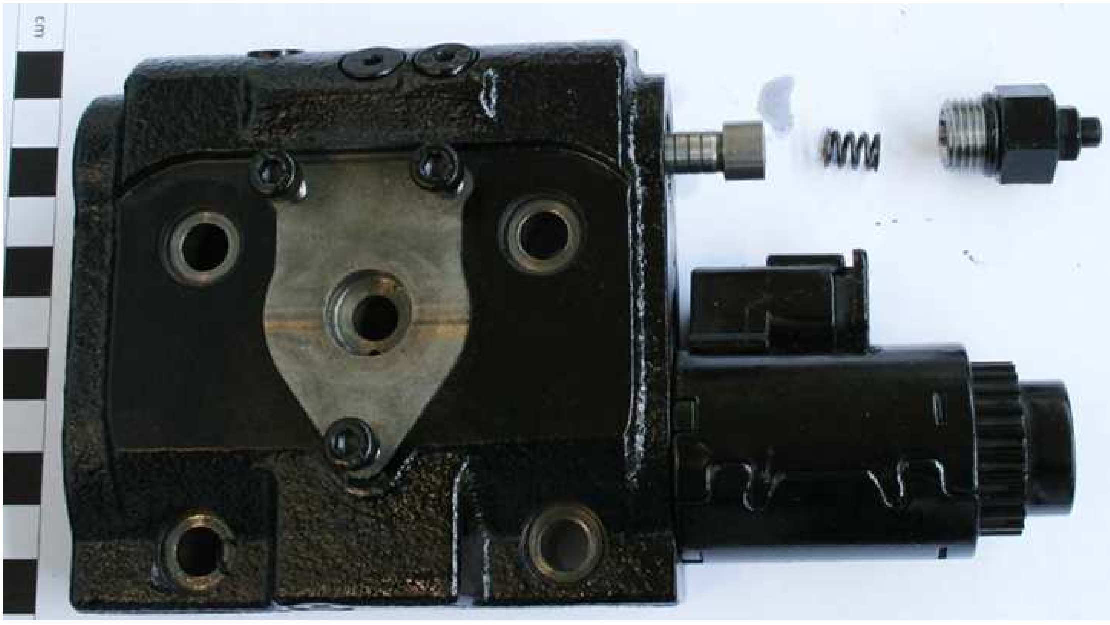

Figure 1.26: Control block with disassembled PCOR

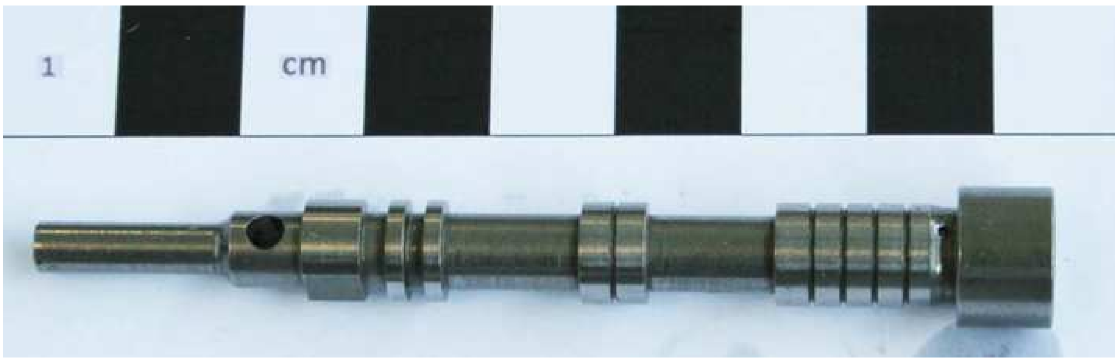

Figure 1.27: PCOR valve

In Fig.1.26 is visible the upper part of the PCOR valve, coming out from the block, the spring, the nut and the screw, all this element are necessary for the valve regulation.
Fig.1.27 is a picture of the valve disassembled from the control block. The valve is placed within the control block, in parallel with respect to the solenoid and cross the entire block along this dimension. From the outside there is the seat cap, placed under the lower part of the valve, and the regulation elements on the upper part. This is a continuous positioning valve.

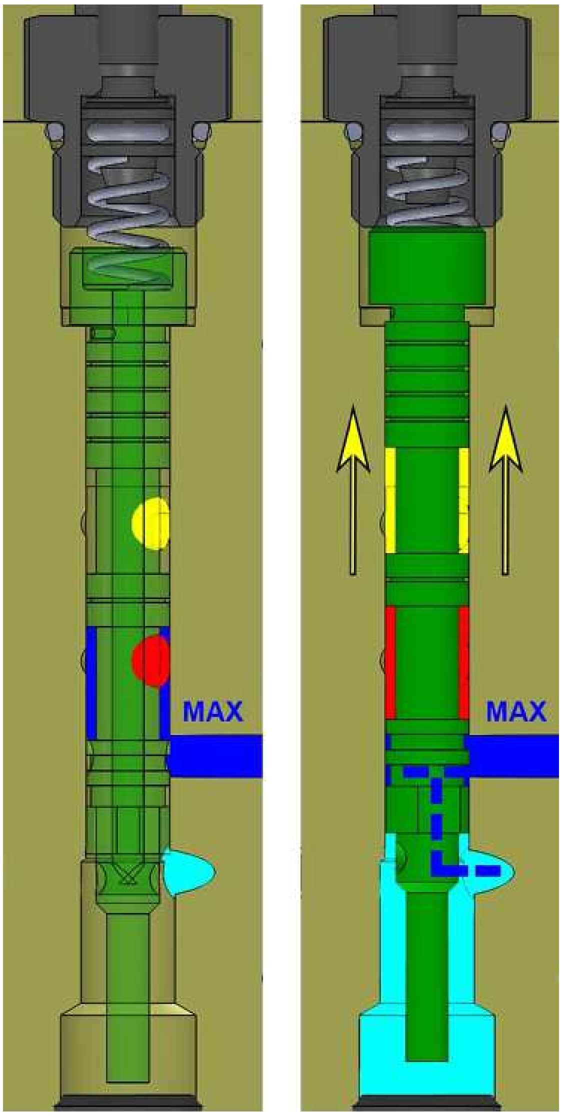

Figure 1.28: PCOR functioning

Taking into account Fig.1.28, on the left side is depicted the rest position of PCOR, whereas on the right side the regulation position. In order to explain the working principle of the valve four different colors, to identify the channels, are used.

The valve is normally in rest position and puts in communication the blue channel with the red channel, that are respectively: the connection to the chamber of linear actuator with bigger section area and the connection to the proportional control valve. In this way, in rest position, PCOR doesn't affect the behavior of the system and the motor decrease its displacement according to the solenoid input.
Taking into account the same hydraulic scheme of the previous subsection in Fig.1.25, the fundamental chamber for the correct functioning of this valve is the yellow one, because it receives the information about the pressure from the HP side. The pressure acts on the upper surface and on the lower surface of the chamber, but the upper surface has a slightly bigger diameter and due to this, the force generated by the pressure is not equal. For this reason, the valve moves to the top and the oil flow is deviated with respect to the previous configuration. In this case, the blue channel is connected to the light-blue one which is directed to the drain. In this case, when PCOR regulates, it connects the chamber of the linear actuator with maximum sectional area to the drain and the motor displacement is brought to the maximum independently from the command input of the proportional control valve, allowing for the reduction of the pressure on the HP side until it goes below the setting value of the PCOR valve. To modify the setting value of the valve, simply act on the nut outside of the control block in order to change the spring preload and the necessary force to move the valve.

Since Series H1 motors are also used for off-highway vehicles, in real situations PCOR system acts in specific cases. For example if the vehicle is moving on a flat ground, the driver just push on the accelerator pedal in order to have minimum displacement and maximum velocity, but if the vehicle approaches a slope it happens that the pressure on the HP side rises, because of the resistant force at the tyres and even if the driver keeps the pedal in the same position as before, PCOR system intervenes decreasing the pressure and bringing the displacement to higher values consequently, depending on the pressure level imposed by resistant torque on the motor.
Thanks to this system the driver can always reach the maximum speed according to the road profile and working conditions.

1.4.5 BPD control system

The BPD control (Brake Pressure Defeat) allows to avoid interferences during deceleration or braking manoeuvres, when the pressure rises in the LP side. In this motor the valve is controlled by an electric input through a solenoid. This is a two-position valve, so it doesn't work in intermediate positions and is axially hollow.
BPD valve (Fig.1.30) is positioned in the motor control block, in parallel with respect to PCOR system and in Fig.1.29 is clearly visible the solenoid responsible for the activation. This valve allows the oil flow from inlet port A or inlet port B.

Figure 1.29: BPD positioning in the control block

Looking at the Fig.1.31 the channel originating from A is blue, whereas the one coming from B is red. The yellow is used for the channel that goes to the proportional valve for displacement control and to the piloting of PCOR system, the last channel indicated in purple is connected to the pressure relief valve and has the function to maintain lubricated the spring chambers and the solenoid chamber.

Being axially hollow, the fluid coming from the relief valve flows to the two extreme chambers, even if the oil is pressurized, it doesn't create problems because the acting surfaces are exactly identical and so forces are equal and opposite reaching a balance, the motion of this valve doesn't depend on this factor.
If the solenoid doesn't receive the electric input, the spring to the left push towards the right side and permits the connection between the inlet port B and all the others control valves. In the opposite configuration the solenoid is active, it overcomes the spring force and so the valve moves towards left and the inlet port connected to the controls is the A port.

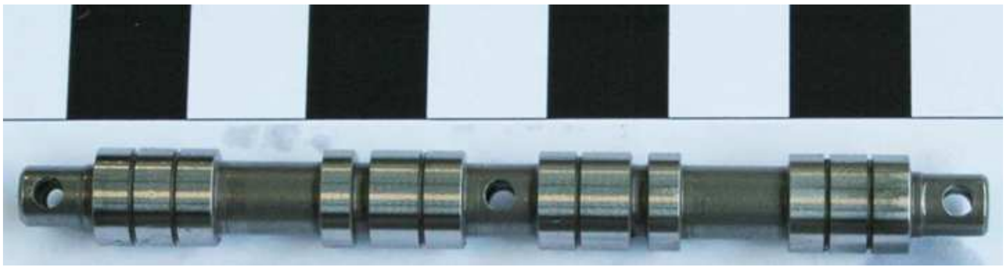

Figure 1.30: BPD valve

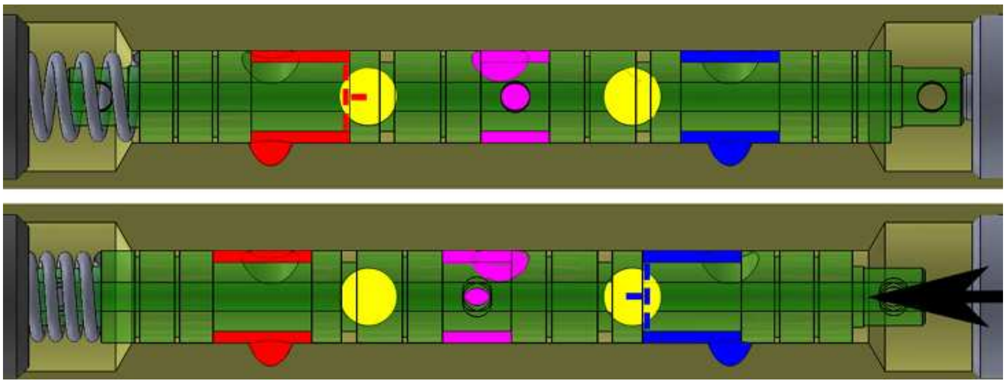

Figure 1.31: BPD functioning

This valve is inserted in the circuit upstream of all other controls, because when the driver brakes or decelerates, the oil pressure in LP side increases and could create disturbances to the system or, in the worst case, substitutes the information needed for a correct functioning of the controls that must arrive from HP side.

For example, supposing that the motor has to decelerate, on LP side the pressure increases, the PCOR system regulates increasing the motor displacement, but this causes a bigger resistant torque and which could leads to an abrupt stopping of the motor. In conclusion this valve is always positioned in such a way that the controls receive the necessary information from the supply branch (HP side).

Chapter 2

Model description

The model used for the simulations is been realized on Simcenter Amesim software, developed by Siemens, it enables the modeling of multi-physics systems and components and allows for detailed analysis relying on a strong numerical core and advanced post-processing.

It's focused on the study of every control valve present in the hydraulic motor object of study.
In fact, the first part of the activity was to create a model for each control valve in order to understand the correct behavior of this components.

In order to do this, the motor is been disassembled, in such a way was easy to perform measurements of all the interested components needed for the construction of the simulation model such as valves, springs and so on. This step was necessary because the construction of the models was realized with the components present in the Hydraulic Component Design library of Amesim.
The second part of the work, instead consisted of assembling the closed-loop of the hydrostatic transmissions regulated by all the control valves designed until now and then, the last step was to simulate the circuit with a 2D longitudinal vehicle model, present in the Simcenter Amesim's library, as load.

2.1 Control valve models

As it's been described in the previous chapter, the motor taken into account for this analysis presents a proportional electric control for the displacement variation and other controls to regulate the pressure in the circuit.
In particular the models of the following valves are been realized with the software:

- pressure relief valve;
- pendulum 3-way valve;

- proportional control valve;
- PCOR system;
- BPD system;

2.1.1 Pressure relief valve

The first model considered is that of the pressure relief valve. As it's been explained in the previous chapter, its function is to impose the pressure of low-pressure side to the value of its cracking pressure.

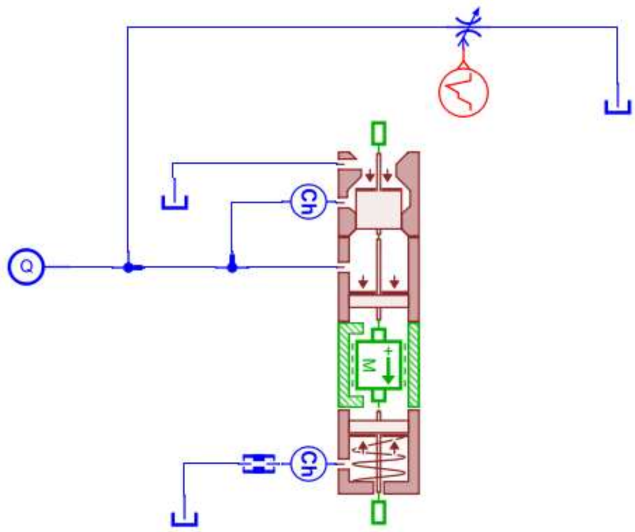

Figure 2.1: Pressure relief valve model on Amesim

The estimation of the dimensions of the valve are been performed manually with a digital calibre, the estimation of the mass from the 3D CAD on Solidworks software, whereas the spring stiffness is been evaluated with the following equation:

$$K = \frac{G \cdot d_w^4}{8 \cdot d_s^3 \cdot n_{coils}} \tag{2.1}$$

Where:

- G is the shear modulus of the material;
- d_w is the wire diameter of the coils;
- d_s is the spring diameter;

- n_{coils} is the number of active coils;

The blocks that make up the valve model come from the library of hydraulic components design in Amesim.

Looking at Fig.2.1, starting from the upper part of the model, it's possible to identify the block that simulates the valve spool(Fig.2.2) and the chamber with the correspondent action surface (Fig.2.3).

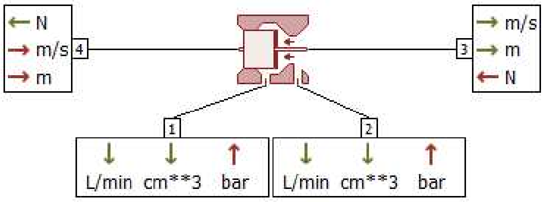

Figure 2.2: Cylindrical poppet with conical seat[3]

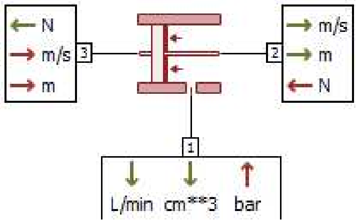

Figure 2.3: Piston or spool with fixed valve[3]

Then the mass block (Fig.2.4) is inserted and finally the last block simulates the spring that decides the cracking pressure through the preload and the stiffness (Fig.2.5).

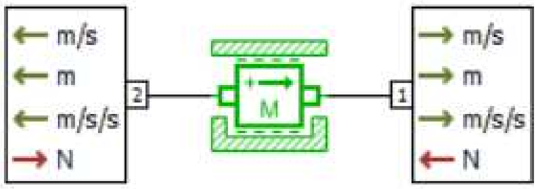

Figure 2.4: Mass with friction and end-stops[3]

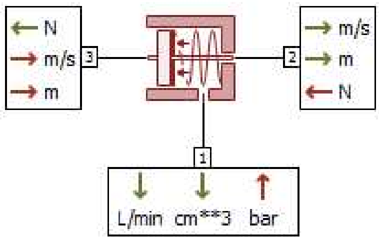

Figure 2.5: Piston with spring[3]

Other components that are present in this valve model, belonging to Hydraulic library in Amesim, are:

- tank;
- junction;
- flow rate source;

- hydraulic chamber;
- hydraulic orifice (fixed or variable);

Here are reported the pictures, one for each element.

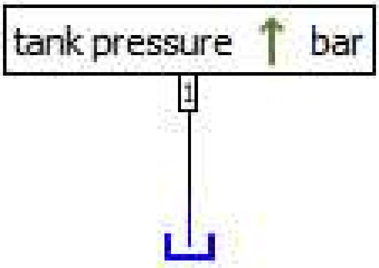

Figure 2.6: Tank model in Amesim[3]

In particular the tank in Fig.2.6, is modeled as an ideal pressure source at 0 *bar*.

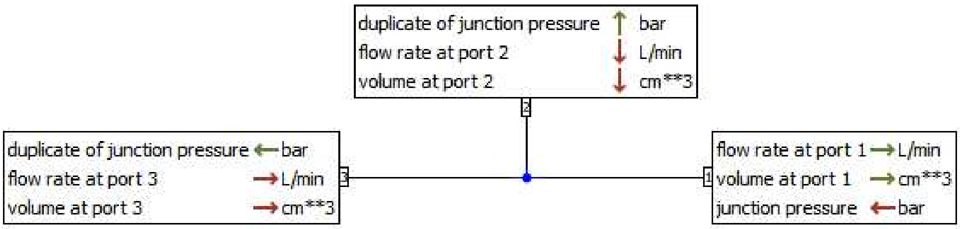

Figure 2.7: Hydraulic junction on Amesim[3]

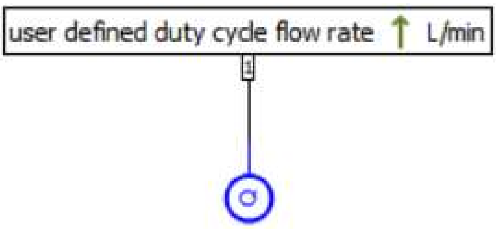

Figure 2.8: Flow rate source in Amesim[3]

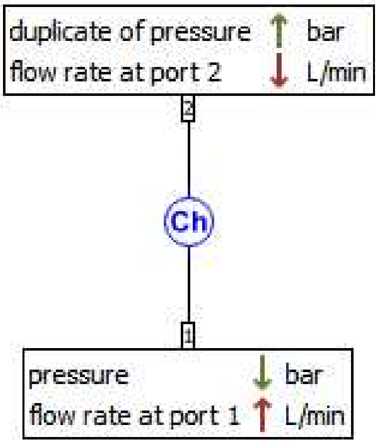

Figure 2.9: Hydraulic chamber in Amesim[3]

Figure 2.10: Hydraulic orifice in Amesim[3]

In Fig.2.7, Fig.2.8, Fig.2.9, Fig.2.10 are represented respectively: a junction, a flow rate source, a hydraulic chamber and a hydraulic orifice. The model of each individual valve is made functionally independent so that its correct functioning can be verified, in this case the valve is supplied with flow rate source block from hydraulic library in Amesim.

The geometric shape of the valve is simulated setting the parameters in each block, the considered friction model is "simple" and is provided by the software.

2.1.2 Pendulum 3-way valve

The pendulum valve connects the low pressure side to the pressure relief valve, due to the action of high pressure, and thus allows the low pressure side to remain under the cracking pressure of the relief valve.

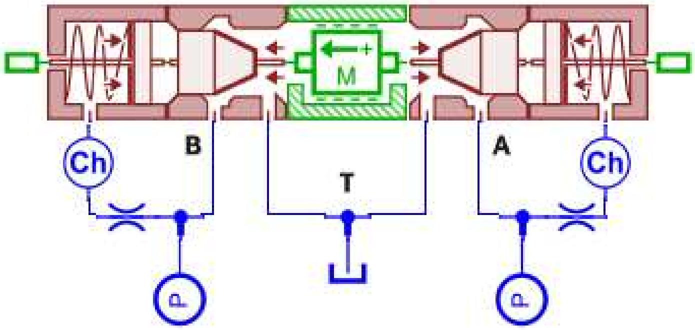

Figure 2.11: Pendulum valve model on Amesim

Observing the Fig.2.11 is visible that it is a symmetric valve.
Also for this valve the geometric measurements are estimated with the same method as before, same for the spring stiffness evaluation. Here there are the same blocks with the spring used before and the blocks with the conical poppet (Fig.2.12), which by moving allow the passage of oil in one side or in the other.
The only new thing with respect to the previous model is the pressure source block that in the model is represented by the blue circle with a "P" inside. This block can be used as a perfect pressure compensated pump.

In this case the correct functioning has been verified providing two pressure source to the valve, in order to simulate the HP side and the LP side. According to the set values in pressure sources, the valve moves to the left or to the right.

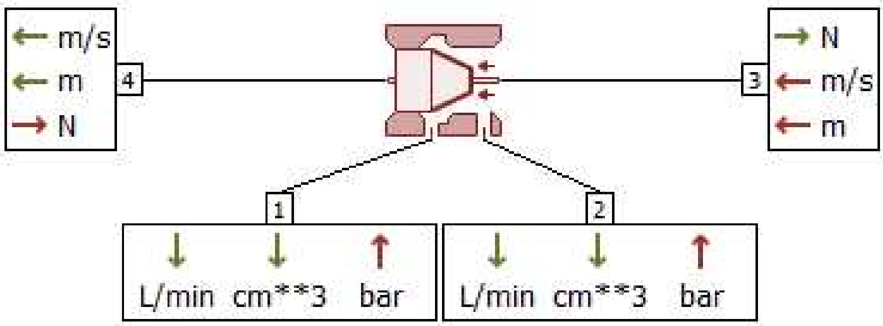

Figure 2.12: Conical poppet valve inside a cylindrical hole[3]

2.1.3 Proportional control valve

The proportional control valve shown in Fig.2.13, activated through an electric input transmitted by a solenoid, is responsible for the motion of the linear actuator, its motion allows for the displacement variation according with the working conditions.

Figure 2.13: Real proportional control valve

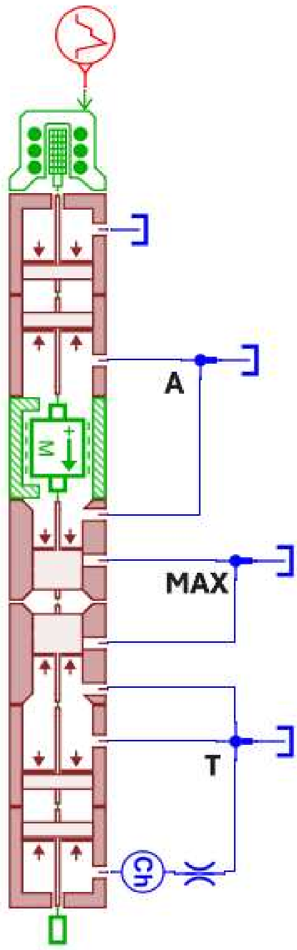

Figure 2.14: Proportional valve model in Amesim

Analyzing in parallel the figures 2.13 and 2.14, starting from the top is the location where the solenoid acts on the valve, in the model this corresponds to the green block which transforms an external input into a force expressed in N. If the signal remains constant to 0 , it means that the solenoid is not active and the valve

doesn't move.
The biggest chamber, that receives the oil from the HP side, visible in the real valve, is the one that has the friction mass block inside of it in the Amesim model. It is connected to the chamber with minimum section area of the linear actuator. Going down through the valve, the smaller chamber is represented by the two blocks described before in Fig.2.2, whereas the hollowed one is delimited by a piston block. The last block represents the action surface at the end of the valve. All the other blocks have already been used in the previous models. For this valve, in order to verify the functioning, it has been simulated a situation with a growing input force and then looking at the flow areas to understand the correct behavior. In fact giving a force, the valve moves downwards, the block with cylindrical poppet above opens a flow area whereas the block below closes the passage for the fluid.

2.1.4 PCOR control system

The PCOR (Pressure Compensator Over Ride) in Fig.2.15 control system limits and maintains constant the pressure on the high-pressure side.

Figure 2.15: Real PCOR valve

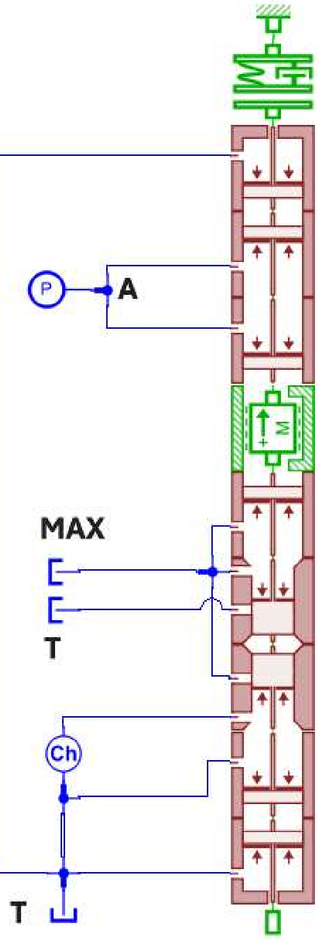

Figure 2.16: PCOR valve model in Amesim

As soon as it reaches PCOR setting pressure, the control system bypass the proportional control valve and forces the motor to increase the displacement to the maximum connecting the proportional valve to the chamber with minimum section area of linear actuator. In this model, depicted in Fig.2.16, the elastic contact between two bodies capable of linear motion, shown in Fig.2.17 is the only new block of this valve. In the real valve it represents the spring inserted upstream of the valve and regulated by a nut and a screw in order to set the pressure. The first chamber starting from the top is represented in the model by two piston blocks connected to a pressure source in order to simulate the HP side. The reason is that the upper action surface of the chamber is slightly bigger, so it generates a force unbalance that moves the valve.

This block simulates the screw and the nut capable of adjusting the system's calibration pressure, as mentioned in the previous chapter. In particular is possible to set the gap with both displacement and the contact stiffness depending on the spring. The way to estimate the spring stiffness is the same as before.

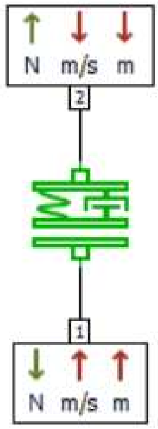

Figure 2.17: Elastic contact model in Amesim[3]

The important thing to highlight in this model is that the chamber supplied by a pressure source, that is the HP pressure in this case, has a slight variation in the action surface. In fact the diameter of the surface that pushes upwards has been set to 8.07 mm, whereas the one of the surface that pushes downwards has been set to 8 mm. It is fundamental for the correct functioning of the valve because this difference create an unbalance on the acting forces and due to this the valve can move.

Looking at the model, the valve moves upwards and so, the block with cylindrical poppet below opens a flow area, instead the block above closes the passage for the oil.

2.1.5 BPD control system

The Brake Pressure Defeat system works with a two-position valve, it allows to supply all the motor controls connecting them with the high-pressure side of the circuit.
Depending on the inlet port, A or B, it moves if the solenoid is activated, otherwise it remains in an equilibrium position according to the spring preload. The chamber in the middle, the one where the mass block is inserted, is connected to the pressure relief valve in order to have a proper lubrication in the valve components.

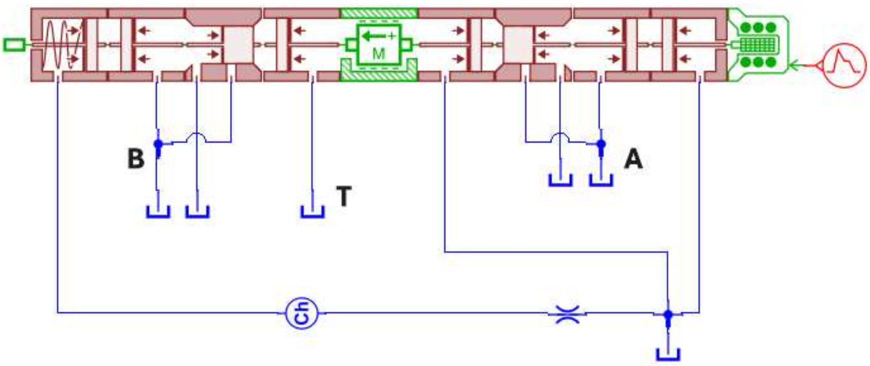

Figure 2.18: BPD valve model on Amesim

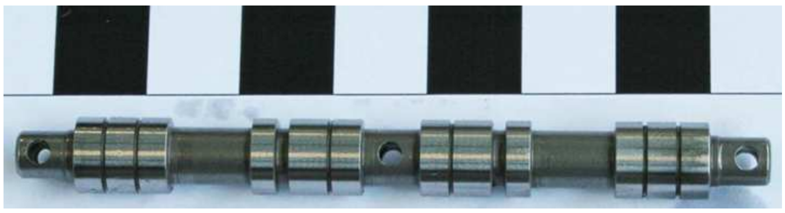

Figure 2.19: Real BPD valve

Giving a look to Fig.2.18, all the blocks present have already been used for the previous models and the equation to evaluate the spring stiffness is the same as before. This model represents the real valve in Fig.2.19.

To verify the correct behavior the solenoid is supplied with a growing input force, and due to this, the valve moves towards left opening a flow area in the block with the cylindrical poppet to the right, whereas the opposite happens on the other side.
An interesting feature to mention observing the model is that this valve is axially hollow as is possible to understand from the hydraulic connections in the figure.

2.2 Hydrostatic transmission model

The next step in the thesis activity was to insert and assemble all the control valves in the circuit.
Initially, this passage was made by slowly inserting the valves starting from the pendulum valve, after which the pressure relief valve was turned. After that, particular attention has been paid to the connection between the proportional control valve, the linear actuator, and, due to this, to the signal given to the motor in order to change its displacement according to the set parameters. Finally, it was the turn of the PCOR control system followed by the BPD control system.

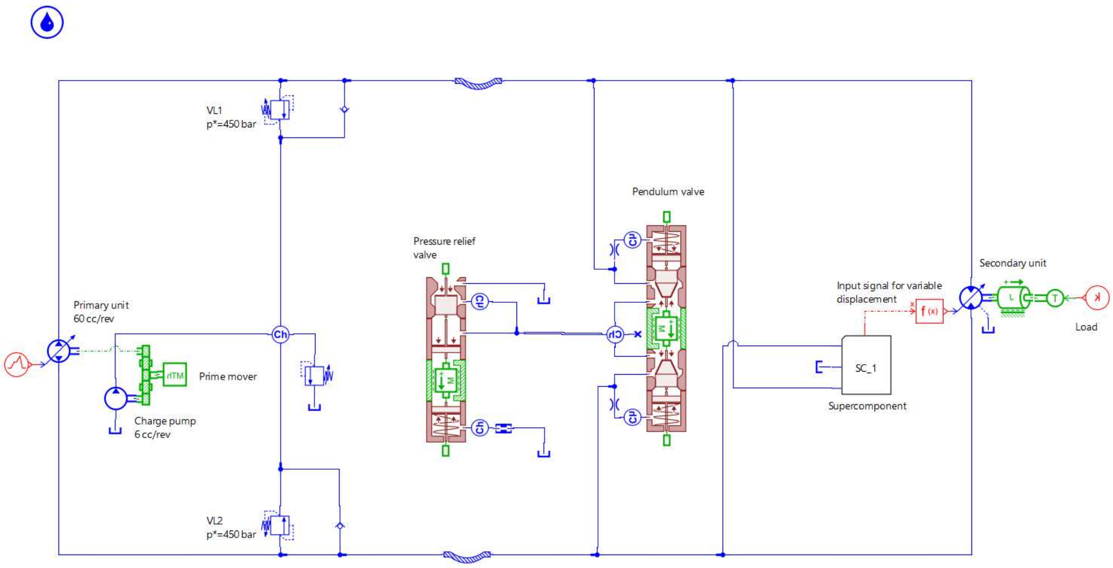

Figure 2.20: Amesim model of the hydrostatic transmission

The closed-circuit hydrostatic transmission model used in the simulations is represented in Fig.2.20. It is composed of two variable displacement hydraulic units, in order to ensure the reversibility of the system is added a flushing system with a 3/3 directional control valve (pendulum valve) previously analyzed. The maximum pressure on the low-pressure side is imposed by the pressure relief valve model connected to the pendulum valve. The prime mover can be an electric motor or an internal combustion engine and feeds both the main pump and the charge pump present to supply the LP side of the circuit in case of leakages and has a displacement of 10% with respect to the main pump.
The maximum pressure on the HP side is ensured by the two relief valves from the hydraulic library, one for each side of the circuit, whereas the relief valve connected to the charge pump is a safety valve that regulates only in particular conditions.

Looking on the right side of the circuit there is a supercomponent created exploiting a feature of the software which allows to agglomerate more elements in a single icon, in order to improve the clarity of the model. Inside of it is present the proportional displacement control, so proportional control valve and linear actuator, together with the PCOR system and BPD system.

The supercomponent gives as output a signal through a mathematical function which correlates the linear motion of the actuator, having calculated its maximum stroke that goes from 0 mm to 44.5 mm, and the fractional displacement of the motor that varies between 0.2 and 1. In particular, a stroke of 0 mm corresponds to a fractional displacement equal to 1, whereas a value of 44.5 mm corresponds to a fractional displacement equal to 0.2.
Connected to the motor there is the block of a rotary load from the mechanical library which receives the motor torque on one port and the load from the other port, which can be a resistant or an overunning load.

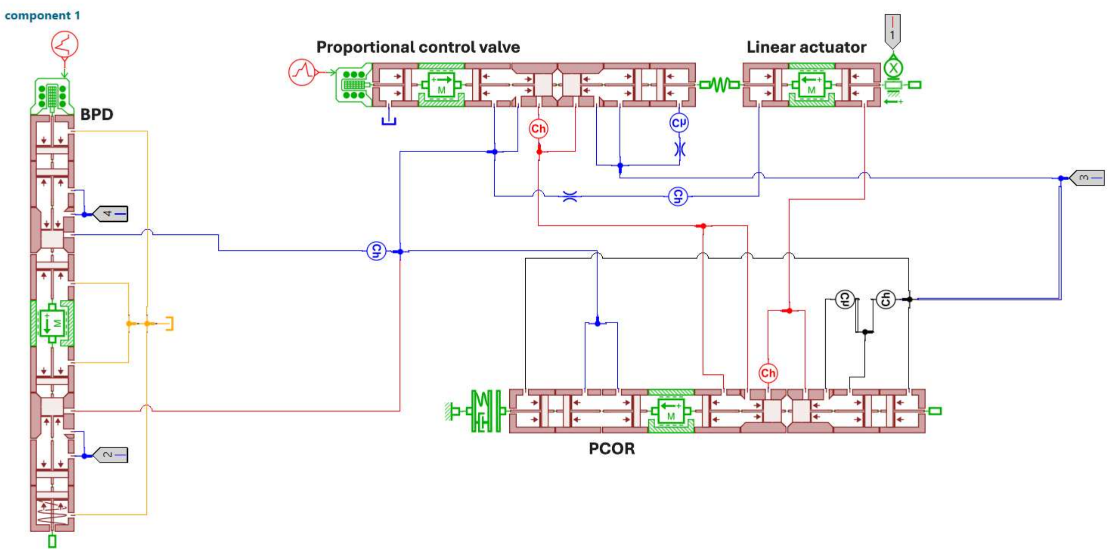

Figure 2.21: Expanded supercomponent in Amesim

In Fig.2.21 it is possible to observe the expanded supercomponent inserted into the hydrostatic transmission circuit. Starting from the left to the right, the BPD valve is the first because it is the one that feeds all the other control valves; in this configuration, its solenoid is activated in order to connect all the motor controls with the high pressure side of the circuit moving downward.
In the upper part of the model there is the proportional control valve connected to the linear actuator through a preloaded spring.

The adopted configuration is the one that brings the motor to minimum displacement when the solenoid is activated, in fact the force generated push to the

actuator that moves along all its stroke.
This situation changes when the pressure on the HP side rises above the setting pressure of the PCOR system which starts to regulate, and bypassing the proportional control forces the motor to maximum displacement in order to decrease the pressure, with the actuator moving in the opposite direction to that set by the proportional valve.

The outputs labeled with numbers from 1 to 4 are connected to the rest of the circuit. In particular:

- port 1 is connected to the motor and returns a fractional value between 0.2 and 1;
- port 2 is connected to the LP side;
- port 3 is connected to the drain;
- port 4 is connected to the HP side;

Now, a brief analysis of the single components present in the circuit is done. Starting from the primary unit, it is constituted by a variable displacement pump from the hydraulic library in Amesim.

This model is an ideal variable displacement bidirectional hydraulic pump; there are no flow losses or mechanical losses.

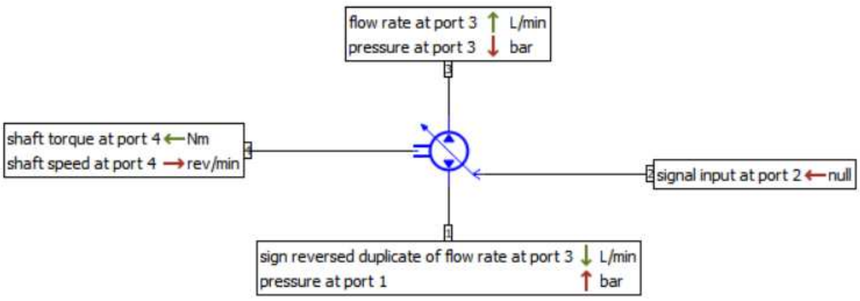

Figure 2.22: Variable displacement pump model in Amesim[3]

Fig.2.22 shows that the flow rate is determined solely by the shaft speed, the swash fraction, and the displacement of the pump. The input signal at port 2 could be a constant or variable fractional value.
The charge pump attached to the same prime mover and with the same rotational speed is the same pump model, but it is not bidirectional and has no variable displacement.
The next element to analyze is the pressure relief valve model in Fig.2.23.

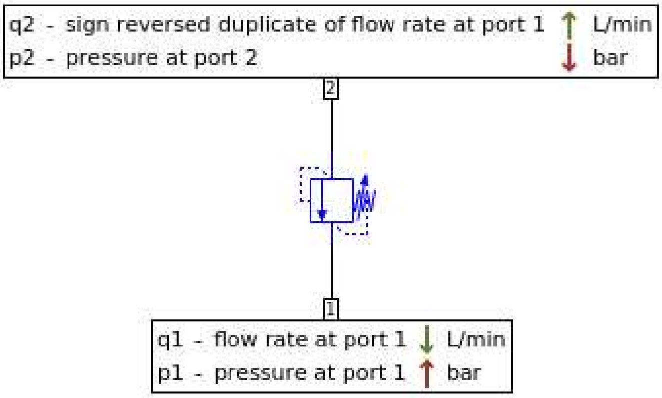

Figure 2.23: Pressure relief valve of hydraulic library in Amesim[3]

The role of this valve is to limit the upstream pressure within the hydraulic circuit and thus protect hydraulic components from over-pressure.
The valve is normally closed; when the pressure drop across it exceeds the cracking pressure (typically set by a spring), the valve opens and lets the fluid flow across so that the pressure drop gets regulated to the cracking pressure. Now it is the turn of another relief valve as depicted in Fig.2.24.

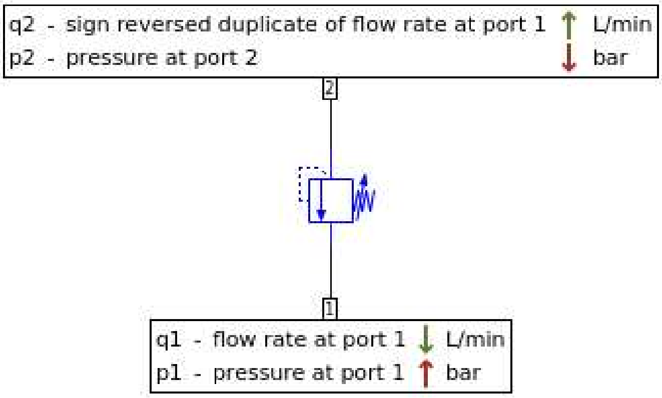

Figure 2.24: Sequence valve model in Amesim[3]

This is the Amesim model for a hydraulic sequence valve or priority valve, but for the usage in this circuit, it is a pressure relief valve. The role of the priority valve is to isolate a portion of the hydraulic circuit, by closing itself, in the case the hydraulic pressure upstream the valve drops below a given threshold.

The following element taken into account is the hydraulic motor model in Fig.2.25. It has an external drain and two directions of rotation.

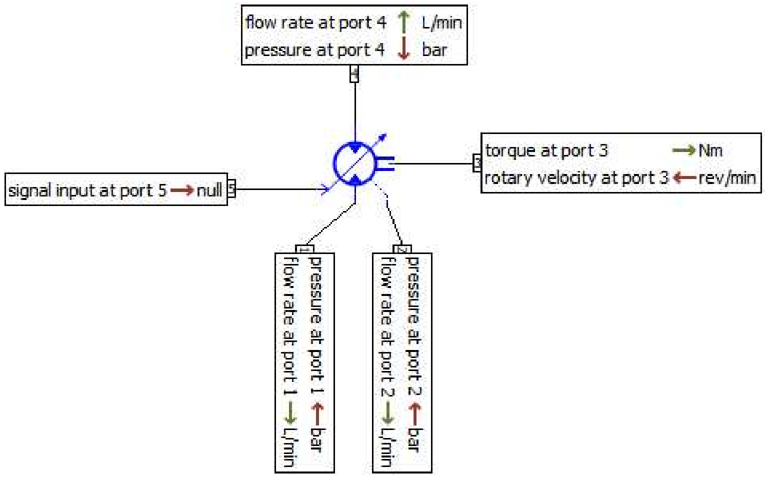

Figure 2.25: Hydraulic variable displacement motor model in Amesim[3]

The last item to be examined more closely is represented in Fig.2.26. It is a simple dynamic submodel of a rotary load under the action of two external torques in $N \cdot m$ applied to its two ports. The existence or absence of friction is configurable, and when it does exist, it is modeled as viscous friction, Coulomb friction, stiction, and windage.

The block connected between the rotary load and the external input is just a submodel that converts the adimensional input into a torque measured in $N \cdot m$.

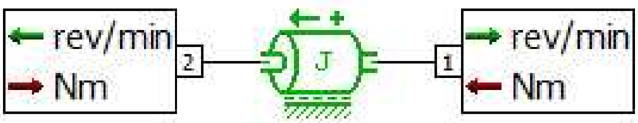

Figure 2.26: Rotary load model in Amesim[3]

2.3 Hydrostatic transmission model with vehicle

For what concern the last step of model assembly, it regards the connection of a real load to the hydrostatic transmission (Fig.2.27).
To achieve this goal, an off-highway vehicle model has been inserted in the model that replaces the load previously present. In particular, the model is a 1D vehicle model. The vehicle is connected to the transmission through a reducer with a set gear ratio, two gears connected to a multi-disc clutch, in turn attached to the rotary load 2 ports block. All these blocks come from the Powertrain library in Amesim, with the exception of the last one, it comes from the Mechanical library.

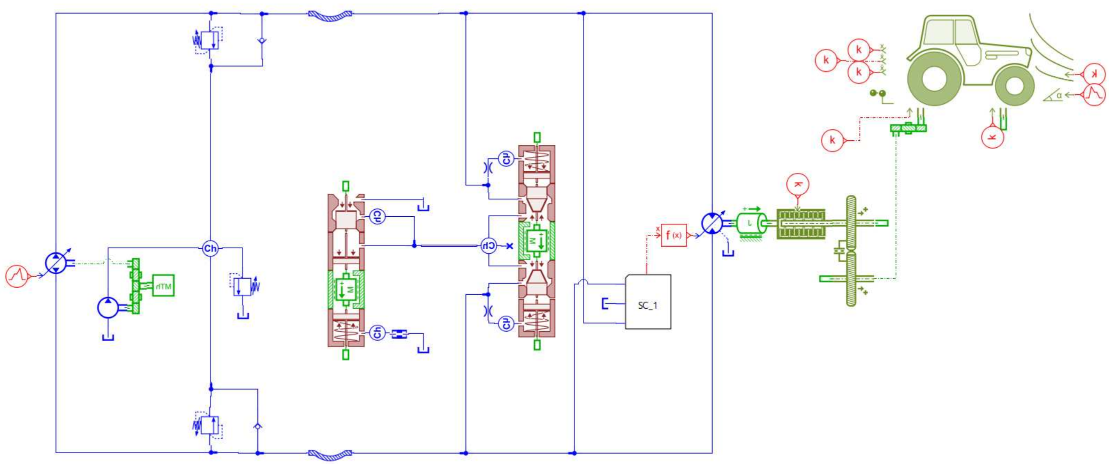

Figure 2.27: Hydrostatic transmission model with vehicle

Looking in more detail at the vehicle model in Fig.2.28, it is possible to analyze all the input received by the model.

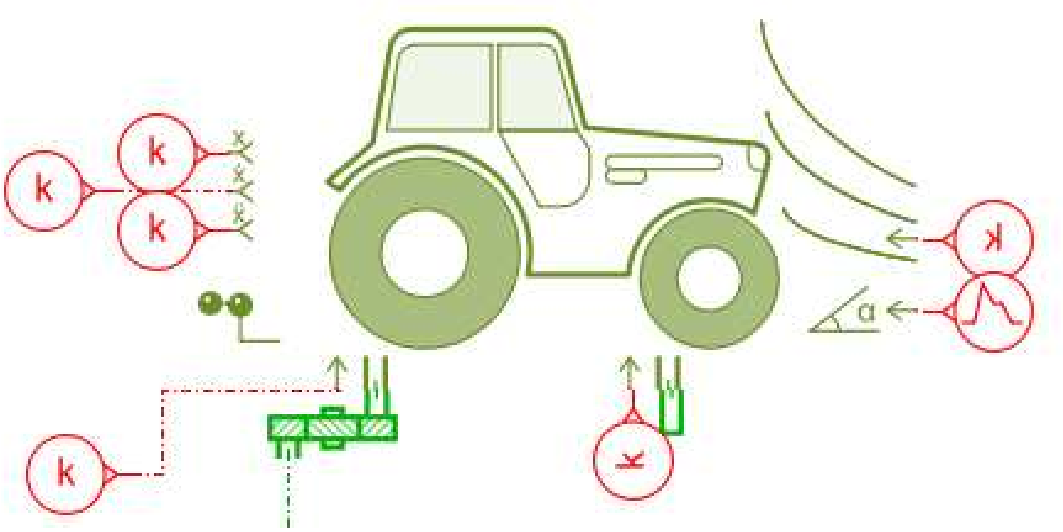

Figure 2.28: 1D vehicle model in Amesim[3]

The three signals on the left side of the picture represent from top to bottom the position, velocity, and acceleration of the vehicle. The two signals under the vehicle indicate the rear axle braking signal and the front axle braking signal, respectively, they can assume values between 0 and 1 and the sum must not exceed 1.
Focusing on the two signals in front of the vehicle, starting from the bottom to

the top, the slope of the road profile expressed in percentage and the wind speed expressed in m/s, even if with these types of vehicles the aerodynamic resistance plays a negligible role with respect to other factors.

From this figure is also visible the reducer shown in Fig.2.29, which represents an ideal rotary mechanical gear system and operates with a fixed gear ratio α, which defines the relationship between the rotary velocities of port 1 and port 2. This model assumes an efficiency equal to 1, so there are no losses due to friction, heat, or other causes.

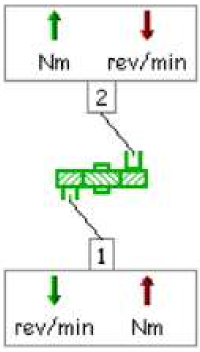

Figure 2.29: Reducer model in Amesim[3]

The equations that regulate this block are:

$$n_2 = \alpha \cdot n_1 \tag{2.2}$$

$$M_1 = \alpha \cdot M_2 \tag{2.3}$$

Where in 2.2 is clear that the output velocity is reduced by a factor equal to the gear ratio, whereas the output torque is increased by the same factor.
Another submodel used to connect the vehicle is the one in Fig.2.30, it represents a multi-disc clutch model where the input at port 2 supplies a value which is used to calculate the frictional torque.

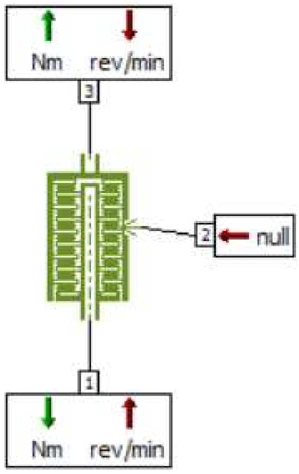

Figure 2.30: Multi-disc clutch model in Amesim[3]

The last block to analyze is the three-port gear used to transmit the torque to the off-highway vehicle model shown in Fig.2.31.

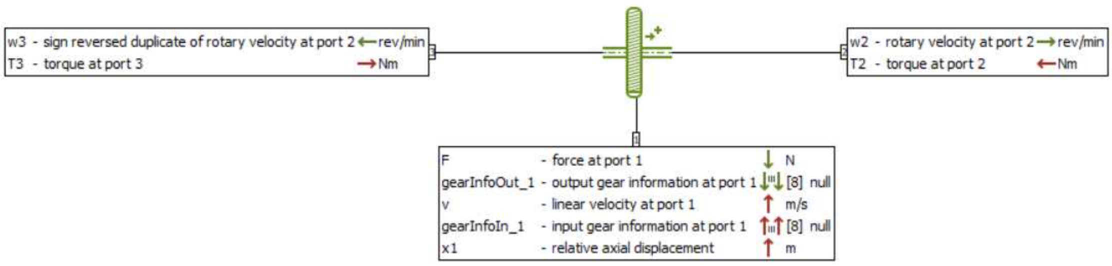

Figure 2.31: 3 port gear model in Amesim[3]

Port 2 and port 3 are rotary, while port 1 takes into account linear motion. The rotational speed at ports 2 and 3 is the same, the linear velocity at port 1 is proportional to the gear radius (2.4). Moreover, the force at port 1 is given by the relationship in eq.2.5 . No losses were considered.

$$v = R \cdot \omega_2 \tag{2.4}$$

$$M_2 = M_3 + R \cdot F \tag{2.5}$$

Where R is the gear radius, v is the tangential speed, ω is the rotational speed, M is the torque and F is the force.

Chapter 3

Simulations setup and results of hydrostatic transmission without vehicle

The first set of simulations has been performed on the hydrostatic transmission circuit without the vehicle in Fig.3.1, to understand the functioning of the model with a simulated load.
For all the simulations was assumed that mechanical and hydraulic components were ideal with the exception of the control valves described in the first section of chapter 2.
The model used for the fluid is always the same for all simulations, the main parameter are listed in Tab.3.1.

Parameter	**Value**	**Unit**
Density	850	kg/m^3
Bulk modulus	17000	bar
Absolute viscosity	51	cP
Absolute viscosity of air/gas	0.02	cP
Saturation pressure	1000	bar
Air/gas content	0.1	%
Polytropic index fo air/gas/vapor content	1.4	-

Table 3.1: Fluid properties

The first set of simulations without considering the vehicle takes into account the parameters shown in and considers the top branch as the HP side of the hydrostatic transmission.

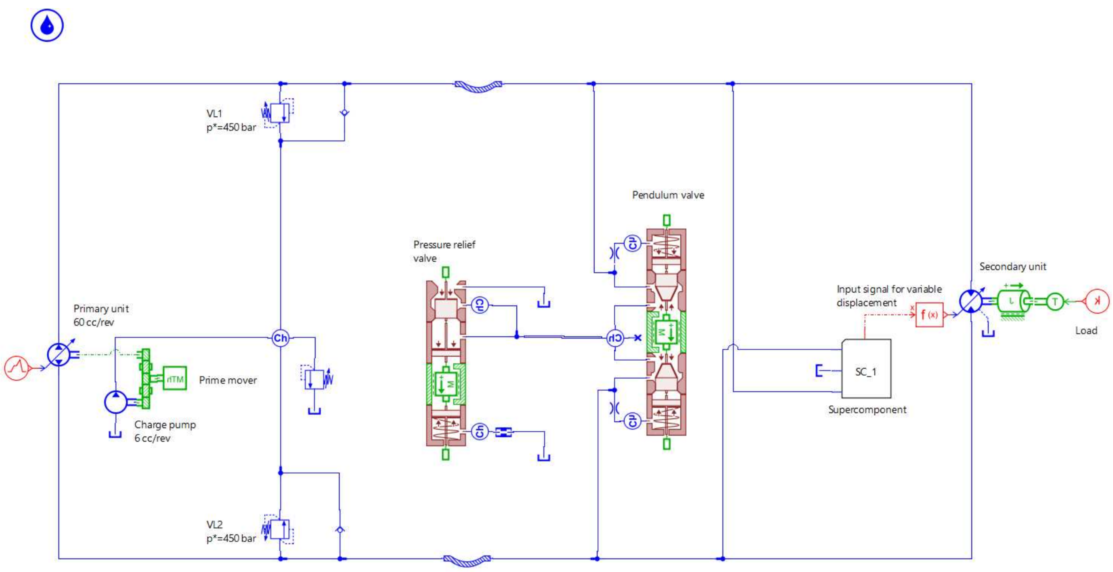

Figure 3.1: Model used for simulations

Parameter	Value	Unit
Prime mover rotational speed	2000	$[rpm]$
Max. pump displacement	60	$[cm^3/rev]$
Charge pump displacement	6	$[cm^3/rev]$
Cracking pressure VL1 and VL2	450	$[bar]$
Crack. pressure relief valve	25	$[bar]$
Max. motor displacement	60	$[cm^3/rev]$

Table 3.2: Set parameters in the circuit

The parameters shown on Tab.3.2 are kept unchanged for all simulations. The simulation time is set to 20 s.

3.1 Simulations with different constant loads

This subsection aims to present the results obtained from simulations with different values of loads maintained constant along the simulation time, highlighting the differences between resistive and overunning load and underlining the behavior of the control valves according to the simulation setup. In the first part is considered a growing resistant load, while in the second part an overunning load is taken into account.

As mentioned above, the hydraulic components are assumed to be ideal except for the control valves and thermal effects are neglected.

3.1.1 Resistant loads simulations

The first simulations have been performed with a constant resistant load, that increases through the simulations, starting from a relatively low value up to bigger value in order to see the differences in the motor's behavior and to verify if the control valves previously described works properly according to the working condition.

An important aspect to underline because affects the circuit response, is the variation of α_1 and α_2. Their variation is shown in Fig.3.2.

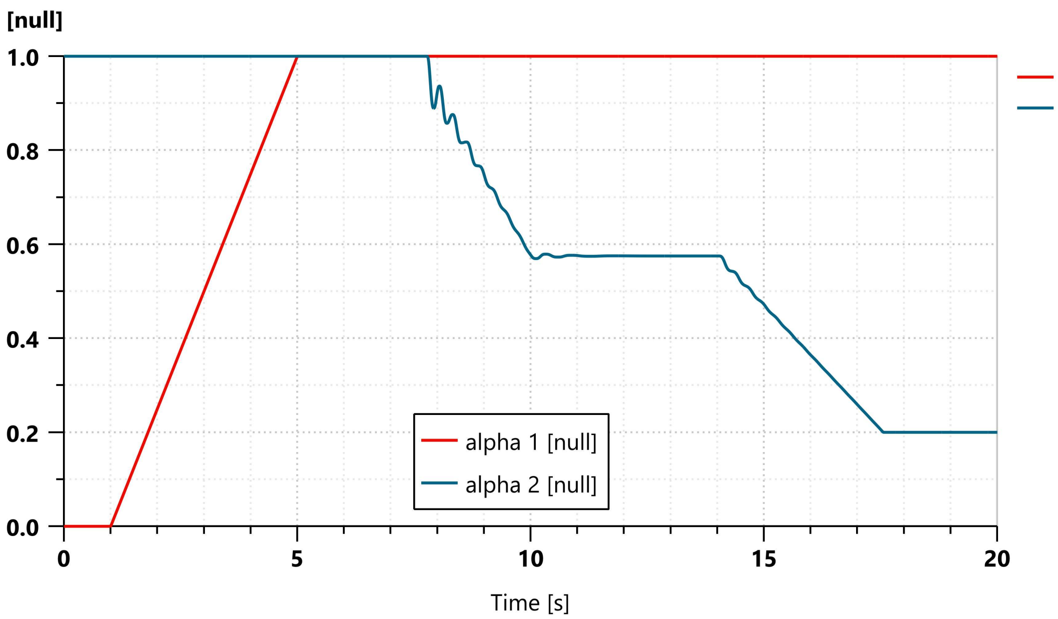

Figure 3.2: Alpha coefficients

Pump displacement starts from 0 for 1 s and then reaches its maximum value in 4 s, the motor starts from maximum displacement up to 8 s, then with the input command given to the solenoid of the proportional control valve, it achieves its minimum possible fractional displacement equal to 0.2. It must be underlined that this is the trend of α_2 without considering the control systems of the motor and the load attached to the system.

The setting of the command given to the proportional control valve is described in the following table 3.3. It acts on the linear actuator according to the model depicted in Fig.2.21.
Moreover, for all the simulations is set in the rotary load block shown in Fig.2.26, a coefficient of viscous friction equal to 0.025 $N \cdot m/(rev/min)$.

Stage	**Start value**	**End value**	**Unit**	**Duration**
1	0	0	N	6 s
2	0	40	N	4 s
3	40	40	N	4 s
4	40	65	N	4 s
5	65	65	N	2 s

Table 3.3: Given command to proportional control valve

This command ensures that the linear actuator moves along all its stroke bringing the motor to its minimum displacement, without considering the intervention of other control systems.
The values of resistant loads used for the simulations are shown in Tab.3.4.

Simulation	**Load**	**Unit**
1	25	$[N \cdot m]$
2	50	$[N \cdot m]$
3	100	$[N \cdot m]$
4	200	$[N \cdot m]$
5	300	$[N \cdot m]$

Table 3.4: Set resistant loads for simulations

First of all it is useful to focus the attention to the pressures in LP and in HP side. In Fig.3.3 the trends of high pressure for different loads are shown.
Looking at the graph it's possible to observe how the maximum pressure reached in the LP side is more or less the same (around 22 bar) independently from the load, which is an acceptable value considering the assumptions made about the non-ideality of the designed valves. Looking at the plot is almost impossible to distinguish one curve from another because the value of the pressure does not depend on the load, but it is imposed by the pressure relief valve.

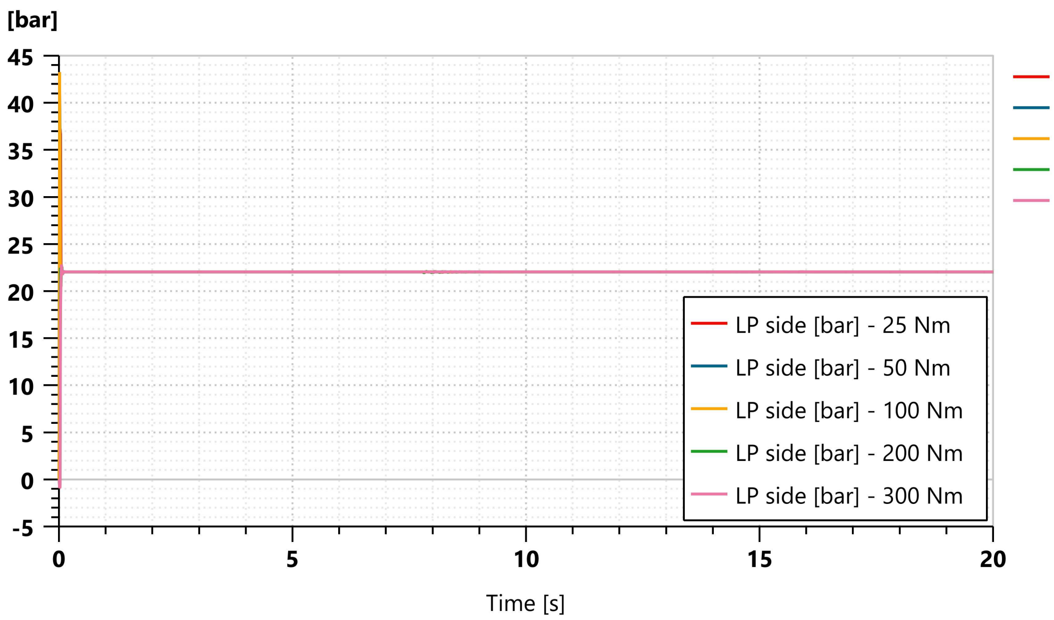

Figure 3.3: Pressure trends in LP side

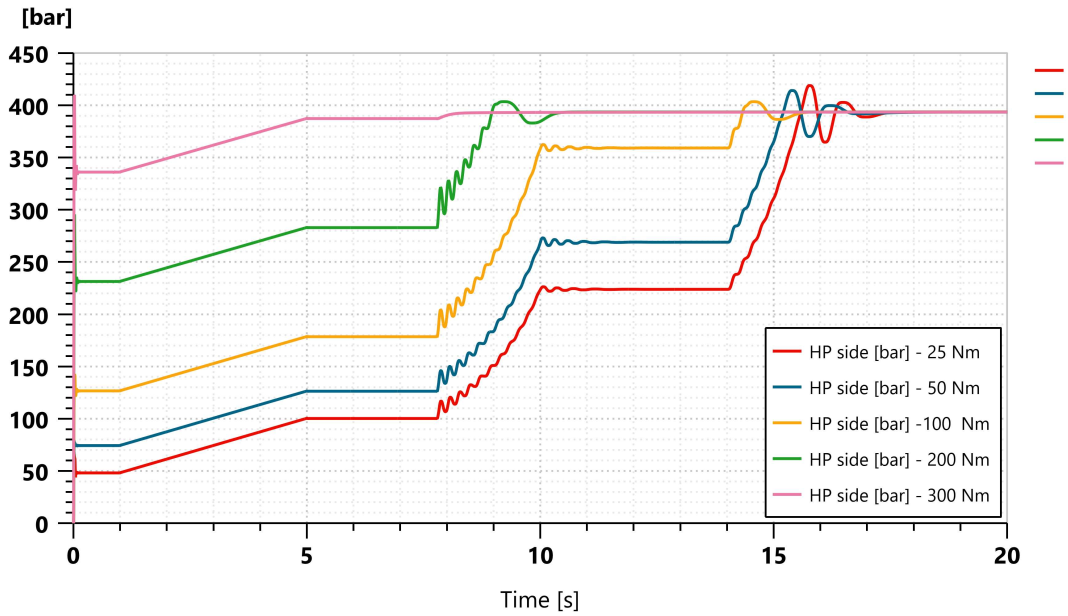

Figure 3.4: Pressures in HP side

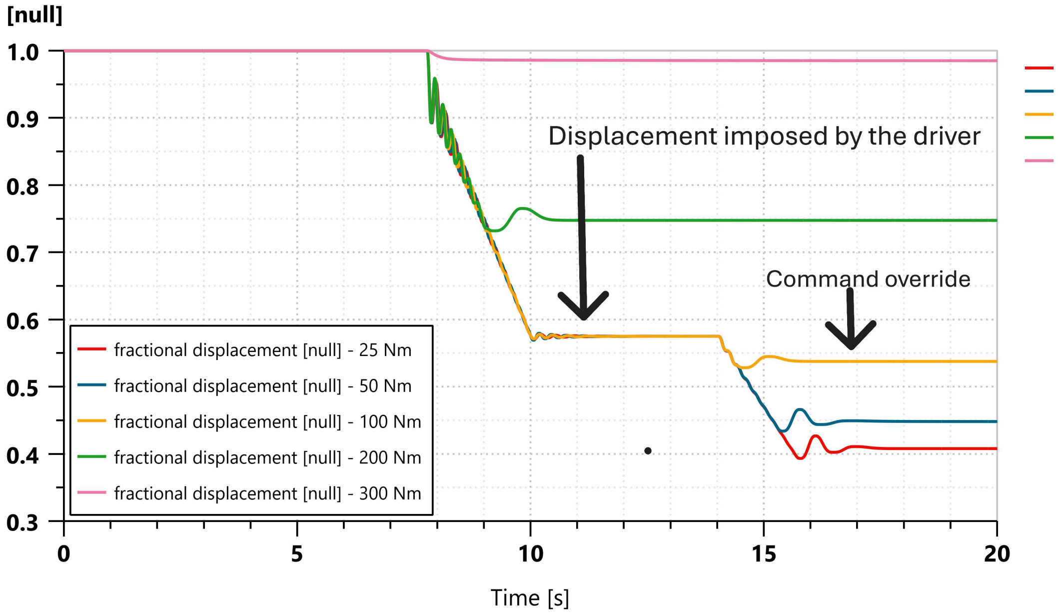

Figure 3.5: Motor fractional displacement

The Fig.3.5 depicts the fractional displacement of the motor. This plot can be analyzed in parallel with the one representing the pressure trend on the high pressure side of the circuit in Fig.3.4. Taking into account the HP side plot, it can be seen that in the first part of the simulation, the pressure trend depends only on the α_1 coefficient and so on the pump flow rate, because there is no input command on the solenoid of the proportional control valve. When the solenoid starts to move the proportional valve, the pressure rises because the motor displacement decreases and the velocity increases. The pressure related to the highest load does not increase so much because the PCOR system intervenes immediately, next it is the turn of the second curve that stops the increment and stabilizes its value around the same pressure. The other three curves increment the pressure up to a certain value, then maintain a constant value in correspondence of the constant input of the solenoid. With the last increment of the input force the pressure rises up to the maximum value when the PCOR system is active for every simulation. The lower the load, the bigger the oscillations.

If the PCOR system were not present, the only input would be from the proportional valve, and the curves would follow the trend with "2 step" (because of the solenoid force) up to a fractional displacement value equal to 0.2.
With respect to the second graph, is visible how the PCOR system intervenes for every load. It intervenes almost immediately for the highest load. In fact the

fractional displacement remains very close to 1. The second curve decreases to a value of around 0.75 and then remains constant. The last three curves receive the second "step" from the proportional valve, but the higher the load, the sooner the control system will intervene. This moment is clearly visible for every curve when there is an oscillation and then a constant trend.

Another interesting plot to analyze is the one that shows the PCOR displacement in Fig.3.6.

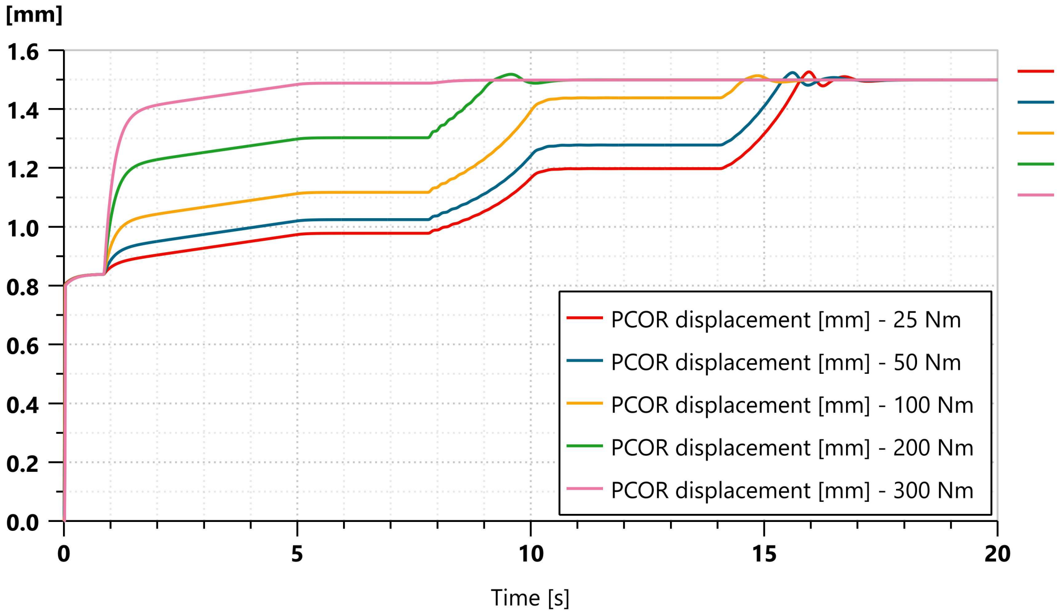

Figure 3.6: PCOR displacement

The PCOR valve starts from a preset displacement value of 0.8 mm, in fact, all curves start from that point at 0 s. After this first phase, the curve that corresponds to the highest load is the first one to achieve the steady-state value of displacement because the pressure on the HP side grows faster with respect to other loads.
On the contrary, the lower the load, the later the valve starts to regulate because the pressure increment is slower. This plot is strictly related to the one of fractional displacement, in fact there is a correspondence between the time at which the PCOR reaches the steady state value and the time at which the motor increases and then stabilizes its displacement.

Concerning the BPD valve, it is a two-position valve and connects the controls to the admission branch of the circuit, so its displacement depends only on the given input command to the solenoid. In fact all the curves follows the same trend of Fig.3.7,independently from the considered load.

The input force from the solenoid reaches a value of 220 N in 2 s and then remains constant. With this input command, the valve completes its entire stroke after 2 s of simulation, feeding the controls with the pressurized fluid.

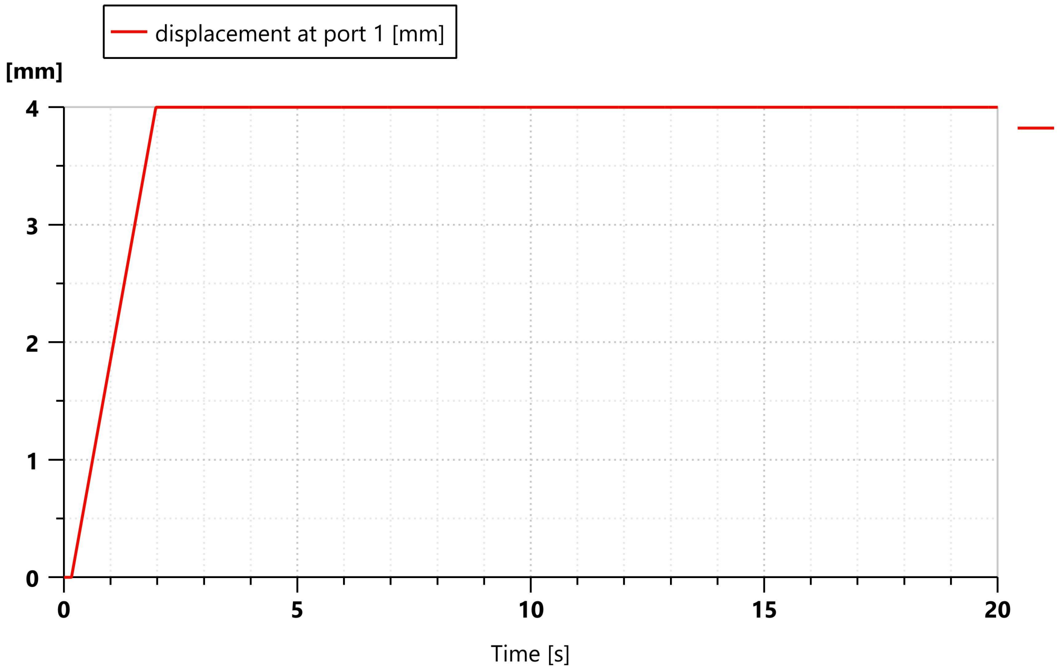

Figure 3.7: BPD displacement

Talking about the 3-way pendulum valve, its displacement is reported in Fig.3.8. It can be noticed a negligible difference between the different simulations because the high pressure is enough to move the valve along its entire stroke even with the smaller load. With respect to the pressure relief valve, its displacement follows the same path for all simulations. This behavior could already be expected looking at the plot of LP side (fig.3.3), because the pressure is imposed by the valve and depends on its design.

Focusing the attention on the motor it is possible to have a look at Fig.3.9 to analyze the rotational speed with different loads. Starting from the smallest load, its curve achieves higher rotational speed because this simulation corresponds to the one with smallest fractional displacement and so the velocity increases, looking at the loads equal to 25 $N \cdot m$, 50 $N \cdot m$, 100 $N \cdot m$, an oscillation is present when the PCOR system regulates, between 14 s and 17 s, it becomes more abrupt decreasing the load. The green curve has an almost linear behavior with some oscillations due to the PCOR valve. The pink curve, related to the highest load has a linear behavior, it reaches its steady-state value after 5 s, thanks to the incremet of pump flow rate and then remains constant along all the simulation time.

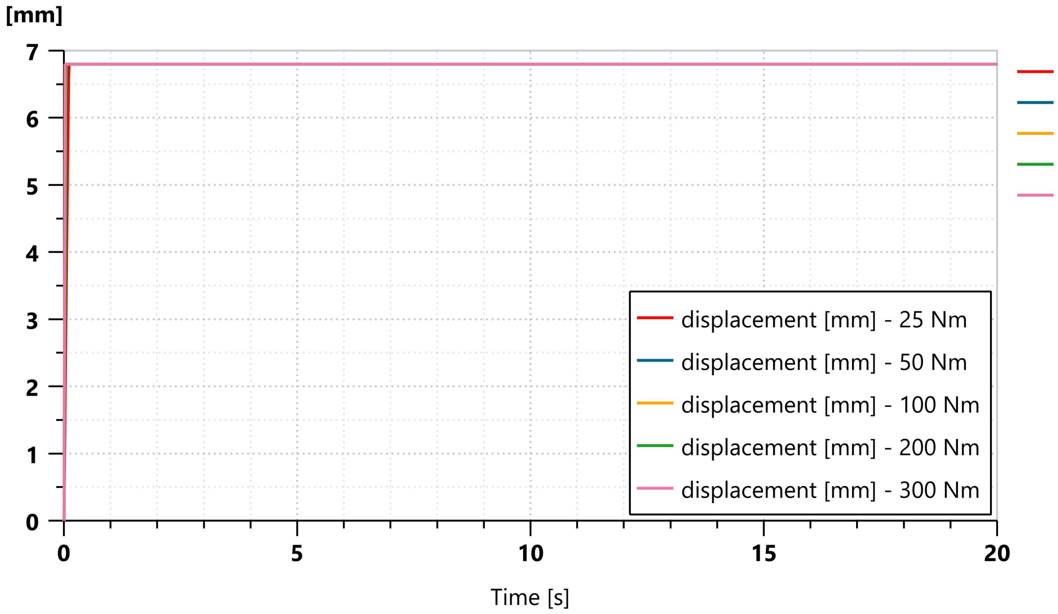

Figure 3.8: 3-way pendulum valve displacement

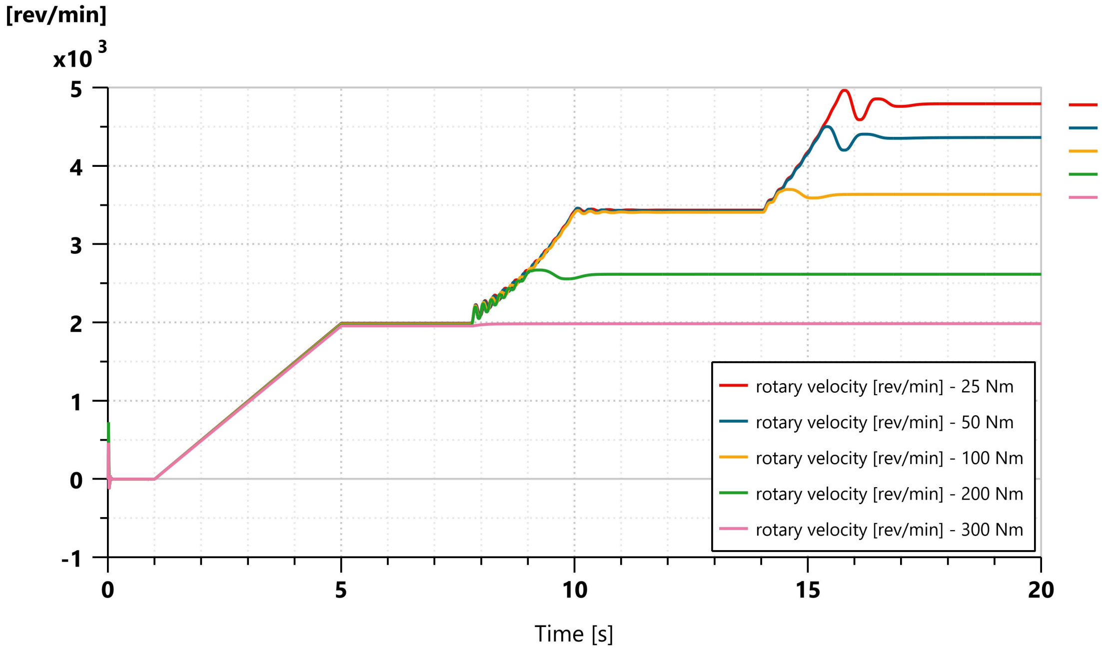

Figure 3.9: Motor rotational speed

The motor velocity has to respect the law stated by 1.3, and so $n_2 = n_1/\alpha_2$. Another important output to underline is the torque delivered by the hydraulic motor under different operational conditions. The related graph is depicted in Fig.3.10.

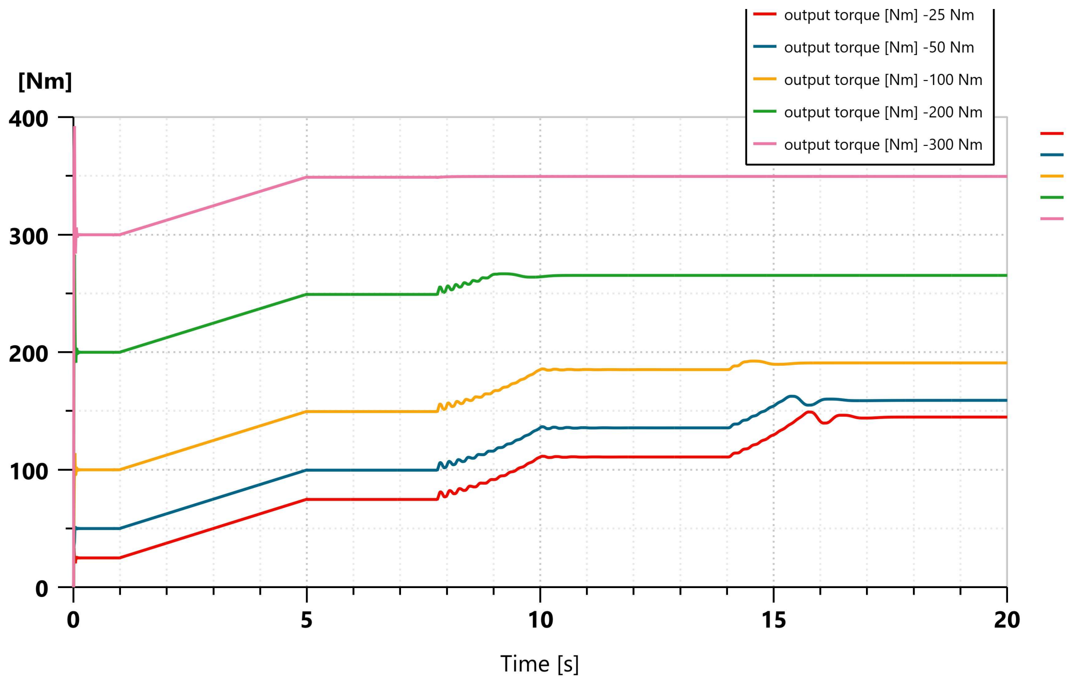

Figure 3.10: Motor's delivered torque

As for the rotational speed also for the torque, the higher the load, the more linear the behavior. The torque tends to increase during the simulation time up to 5 s because the pump displacement increases during that simulation time. For this reason a minor increment is found on the curve referred to the higher load, in fact the maximum pressure is achieved before with respect to the other conditions, and due to this the torque does not increase so much.
In particular looking at the curve referred to the load equal to 300 $N \cdot m$, the torque appears with a small increment in the first 5 s and then is constant in the second part of the simulation.
Being the torque the product between the displacement and the delta pressure this plot is in agreement with those related to fractional displacement and pressure in HP side. The output torque delivered by the motor, according to the 1.4, has to be $M_2 = M_1 \cdot \alpha_2$.

Now it is possible to analyze the performance of the system from the power point of view. In particular, in a system like this, the power of the primary unit is calculated with the 3.1 and the power of the secondary unit with the 3.2:

$$P_1 = M_1 \cdot n_1 \tag{3.1}$$

$$P_2 = M_2 \cdot n_2 \tag{3.2}$$

Where:

- M_1 and M_2 are the torques delivered by primary and secondary unit respectively;
- n_1 and n_2 are the rotational speeds of primary and secondary unit respectively;

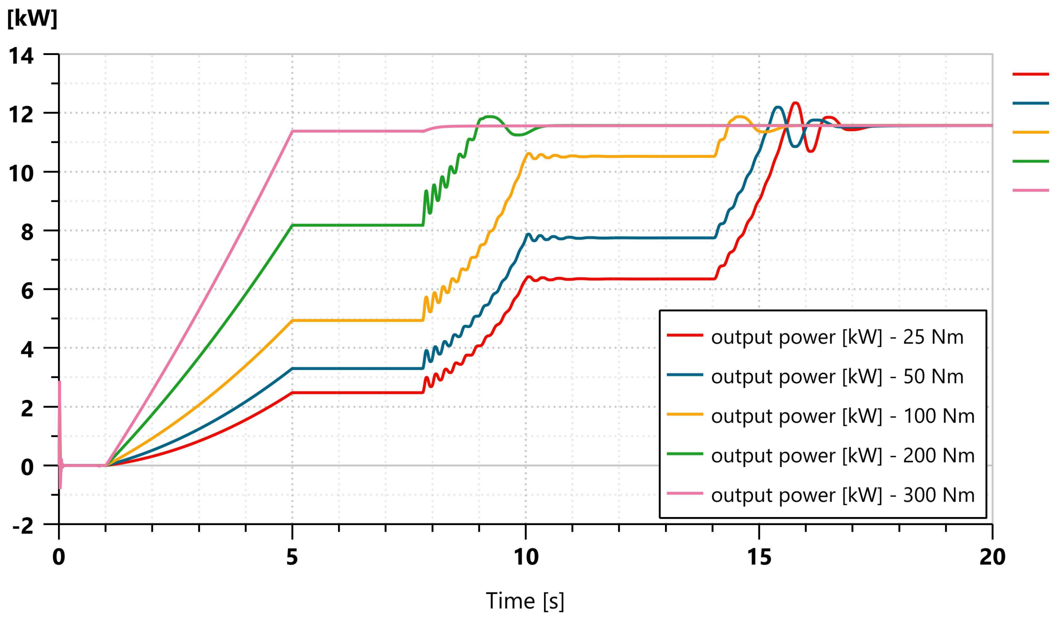

Figure 3.11: Output power

Analyzing the output power depicted in Fig.3.11, it can be observed that at the end of the simulation, the system achieves the same value for all the loads considered. The smallest one has a more oscillatory behavior when the PCOR system regulates, whereas moving the analysis towards higher loads, the trends becomes more linear. Being the output power function of the torque, of the delta pressure, of the speed and of the displacement, the results are in line with what could be expected.

3.1.2 Overunning loads simulations

This simulations have been performed simulating an overunning load attached to the secondary unit of the transmission to understand the differences with respect to the previous simulations.
The variation of α_1 is set like in the previous cases, as depicted in Fig.3.2. With respect to the previous simulations, now they have been perfomed one at a time, changing from time to time, the input command for the proportional control valve, always with reference to the supercomponent in Fig.2.21. Two different overunning loads, constant during the simulation time, have been considered and are listed in Tab.3.5.

Simulation	**Load**	**Unit**
1	100	$[N \cdot m]$
2	150	$[N \cdot m]$

Table 3.5: Set overunning loads for simulations

An overunning load means, for example, a situation where an off-highway vehicle travels downhill. In a situation like this, the driver wants to decelerate, so the simulations have been performed supposing that at a first time the driver starts to accelerate and then to decelerate in order to decrease the output velocity of the motor.
Increasing the load means to simulate an increment of the slope in a downhill road, for this reason the command input to the proportional control valve has been set time by time for each simulation. The command for the first load is the following in Tab.3.6.

Stage	**Start value**	**End value**	**Unit**	**Duration**
1	0	0	N	6 s
2	0	40	N	4 s
3	40	40	N	4 s
4	40	0	N	4 s
5	0	0	N	2 s

Table 3.6: Command to proportional control valve 1^{st} case

Also in these simulations has been taken into account a coefficient of viscous friction equal to 0.025 $N \cdot m/(rev/min)$ for the rotary load block represented in Fig.2.26.
For overunning loads is useful to start from the plot representing the pendulum

valve displacement because this valve moves in the opposite direction now and this motion determines the switch between the two sides of the circuit. This plot is the same for all the value of overunning loads chosen for these simulations and is shown in Fig.3.12.

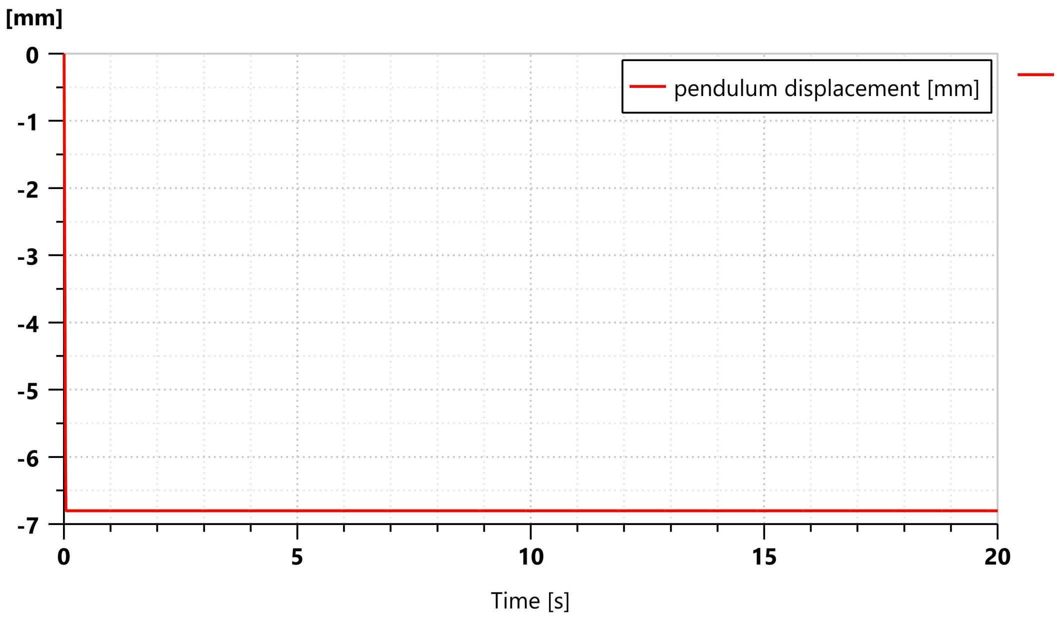

Figure 3.12: Pendulum displacement

As for the previous simulations, also in this case, analyzing the case with an overunning load of 100 $N \cdot m$, is useful to have a look to the pressures in low-pressure and in high-pressure side. In fact, with an overunning load the branches of the circuit have to switch, the one that was the HP-side, now is the LP side and viceversa.

Starting from the pressure achieved in the low-pressure, shown in Fig.3.13, it is possible to understand that the value of pressure is exactly the same with respect to the resistant loads cases, but now the LP side is the top branch in the circuit looking at the model in Fig.3.1.
This result is expected because this pressure depends only on the design of the pressure relief valve.

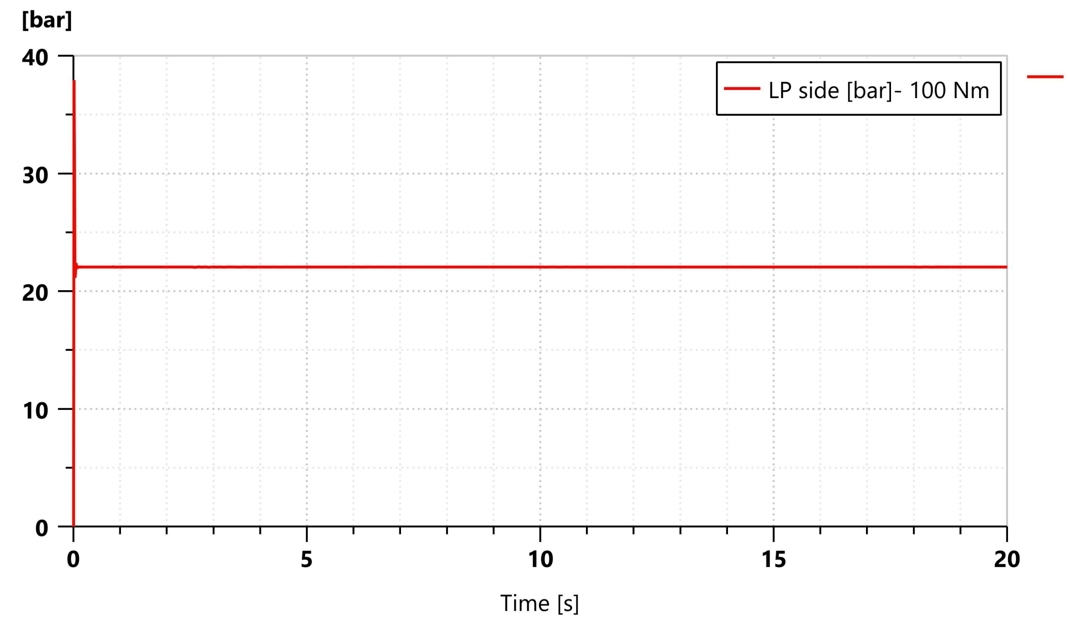

Figure 3.13: LP side overunning load 1^{st} case

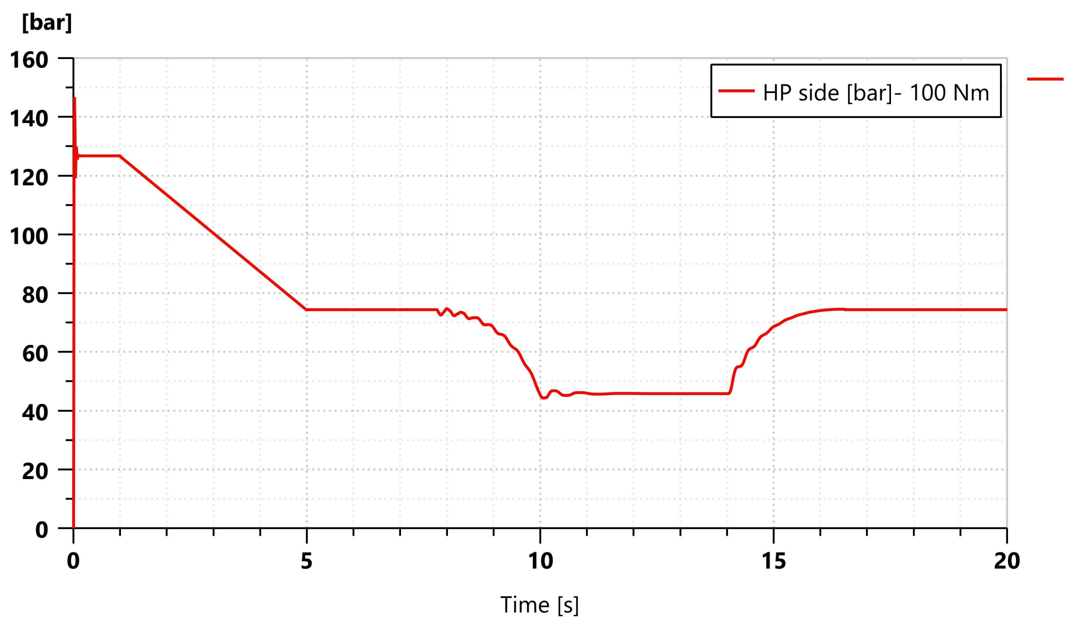

Figure 3.14: HP side overunning load 1^{st} case

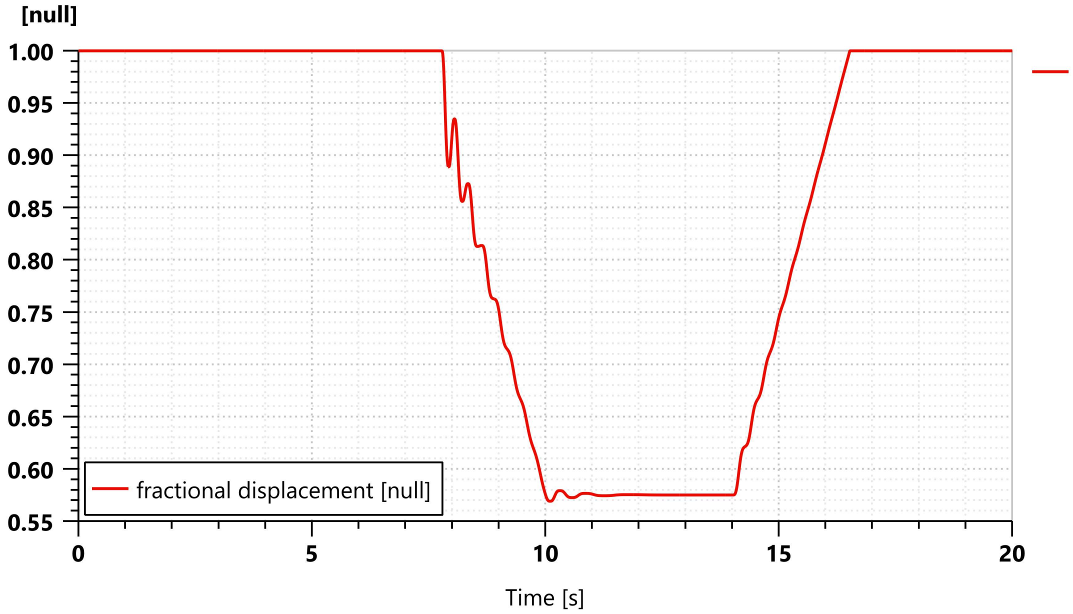

Figure 3.15: Fractional displacement - 100 $N \cdot m$

Regarding the HP side of the circuit in Fig.3.14, it is useful to analyze this graph together with the one related to the fractional displacement of the hydraulic motor in Fig.3.15. Now the fractional displacement depends only on the driver's command, in fact, up to 6 s he accelerates according to the table 3.6, and then he wants to decelerate because it is supposed to be a downhill road.

In correspondence of the acceleration, the pressure tends to decrease because the velocity increases, the torque decreases, and with a constant flow rate, the variable that has to decrease is the pressure. After the acceleration phase, the motor comes back to higher displacement and there is the opposite trend for the pressure, now it increases up to its steady-state value equal to almost 80 bar. The fractional displacement goes up to 1 following the input command previously described.

In this case the PCOR system does not regulate because the pressure reached in the HP side is lower than the set pressure of the valve, in fact the curve of the fractional displacement follows only the driver's command. Talking about the BPD valve, the command is the same as before because even if the branches are switched, the admission port remains the same.

Now it is possible to analyze the output rotary velocity of the hydraulic motor represented in Fig.3.16.

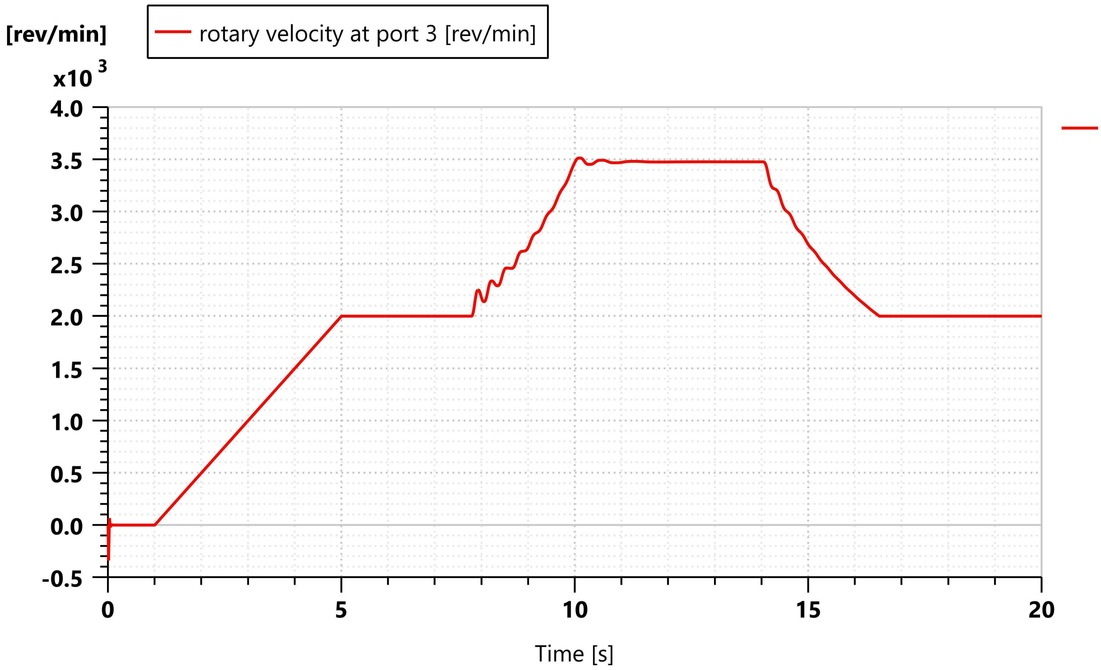

Figure 3.16: Output rotary velocity- 100 $N \cdot m$

The velocity starts from zero because of the setting of α_1, then increases up to a value of around 3500 rev/min in the acceleration phase and maintains this value when the fractional displacement is constant. Then, in the second phase, the driver wants to decelerate, the displacement increases, and the velocity decreases until, with a fractional displacement equal to 1, it reaches the minimum possible value of $2000 rev/min$, that is the rotational speed of the prime mover and of the pump.

After that, it is time to analyze the plot related to the torque delivered to the output shaft. Being the torque the product between the flow rate and the pressure drop, the trend can be expected to be similar to that of pressure since after the first 5 s the flow rate is constant.

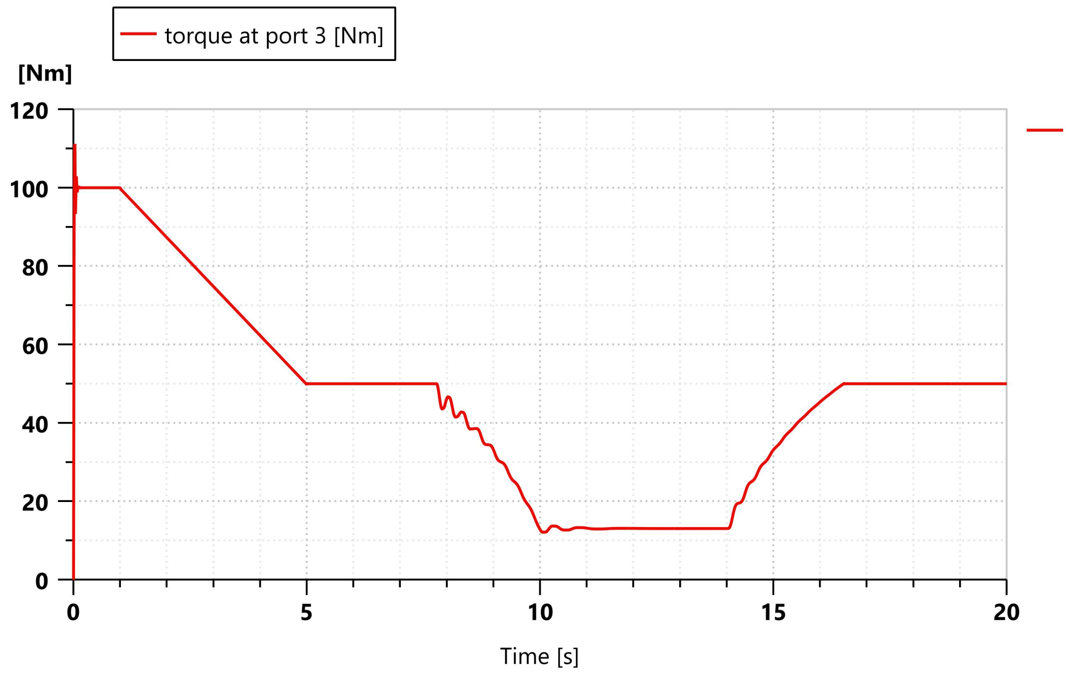

Figure 3.17: Output torque- 100 $N \cdot m$

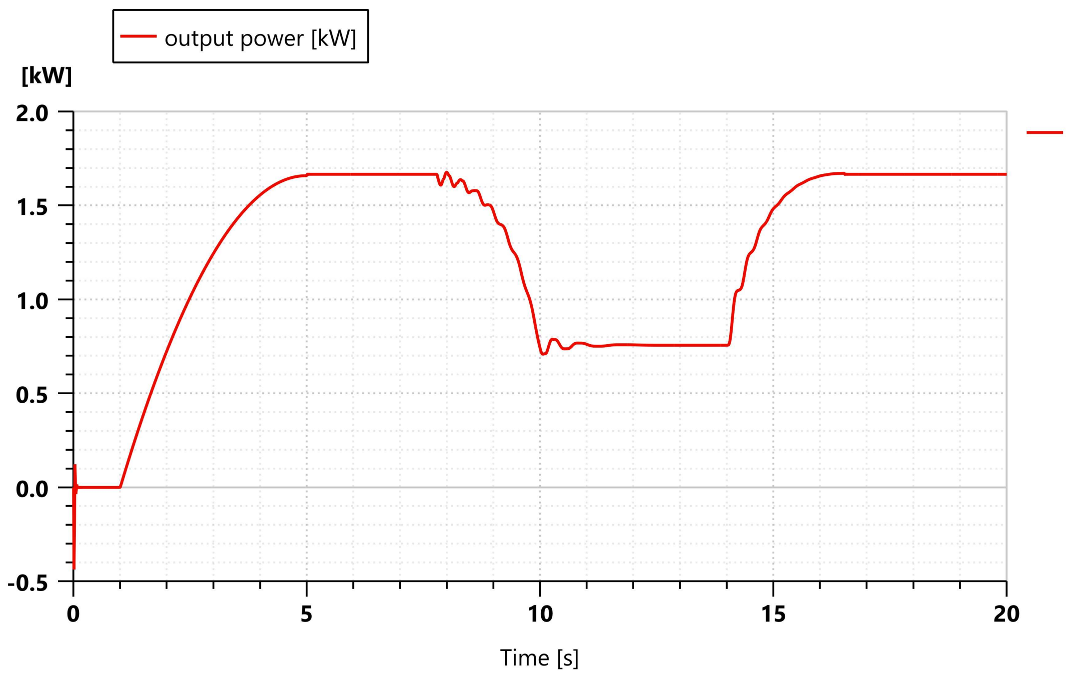

Figure 3.18: Output power- 100 $N \cdot m$

As it is clearly visible from the plot in Fig.3.17, the output torque starts from a value of $100 N \cdot m$, that is the set value for the overunning load, it remains constant for 1 s because there is no flow rate at the start of the simulation.
After this first phase it decreases, when the driver accelerates and the fractional displacement decreases, up to a value a little bit higher than 10 $N \cdot m$. Then, in the deceleration phase the torque grows up to a value equal to 50 $N \cdot m$ when the motor reaches its maximum displacement and the velocity is as low as possible.

The last plot to comment is the one that shows the output power in Fig.3.18, delivered by the secondary unit. The delivered power is calculated as the product between the delivered torque multiplied by the output rotational speed.

Taking into account this information, it is possible to see that the power achieves a peak at around 4 s of simulation time because even if the torque decreases, the rotational speed is going to reach its maximum value for this simulation. After this peak the power is affected more by the torque that is at its minimum value and decreases to a constant value up to 10 s. This first part corresponds to the acceleration phase, then when there is the deceleration phase, the torque starts to increase and the velocity decreases, but the power grows reaching a steady-state value, till the end of the simulation, almost equal to the peak reached in the first phase.

Now it is time to analyze the second test that has been performed with an overunning load of 150 $N \cdot m$ and with a different input command from the driver and so to the proportional control valve, described in the following table 3.7. For this case has been set a lower input command because it simulates a downhill road with higher slope, so the driver wants to decelerate more with respect to the previous case.

Stage	**Start value**	**End value**	**Unit**	**Duration**
1	0	0	N	6 s
2	0	25	N	4 s
3	25	25	N	4 s
4	25	0	N	4 s
5	0	0	N	2 s

Table 3.7: Command to proportional control valve 2^{nd} case

Looking at the low-pressure side graph in Fig.3.19, it is almost exactly the same as that of the first case, except for the peak reached right at the beginning of the simulation. In this case, it is much lower than before and just slightly exceeds the value at which the pressure stabilizes during the simulation. After the pressure stabilizes, it is exactly at the same value as the previous simulation, demonstrating that it depends exclusively on the design of the pressure relief valve.

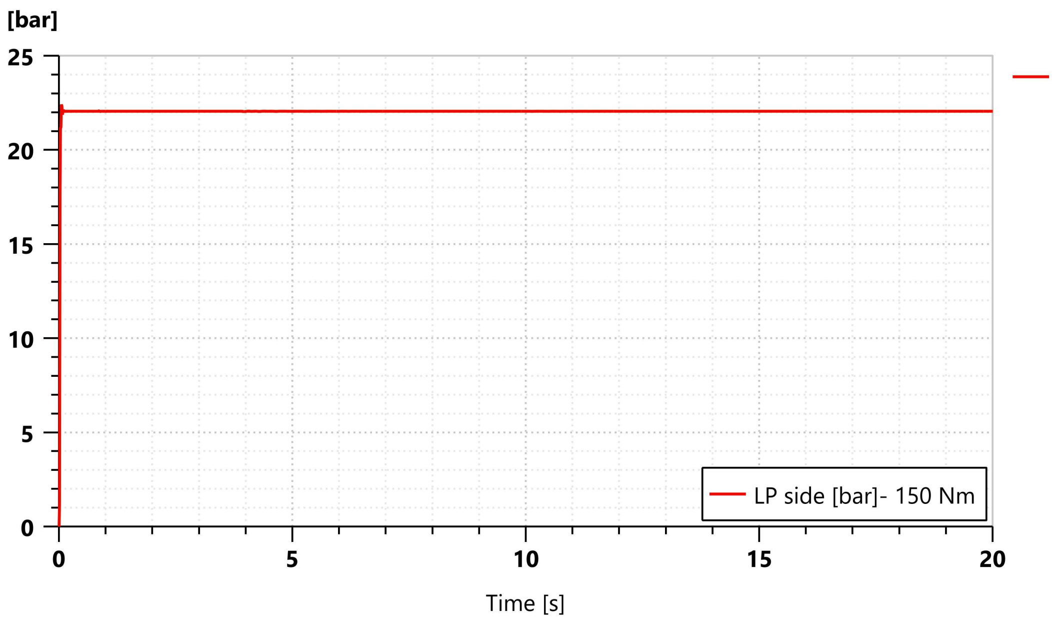

Figure 3.19: LP side overunning load 2^{nd} case

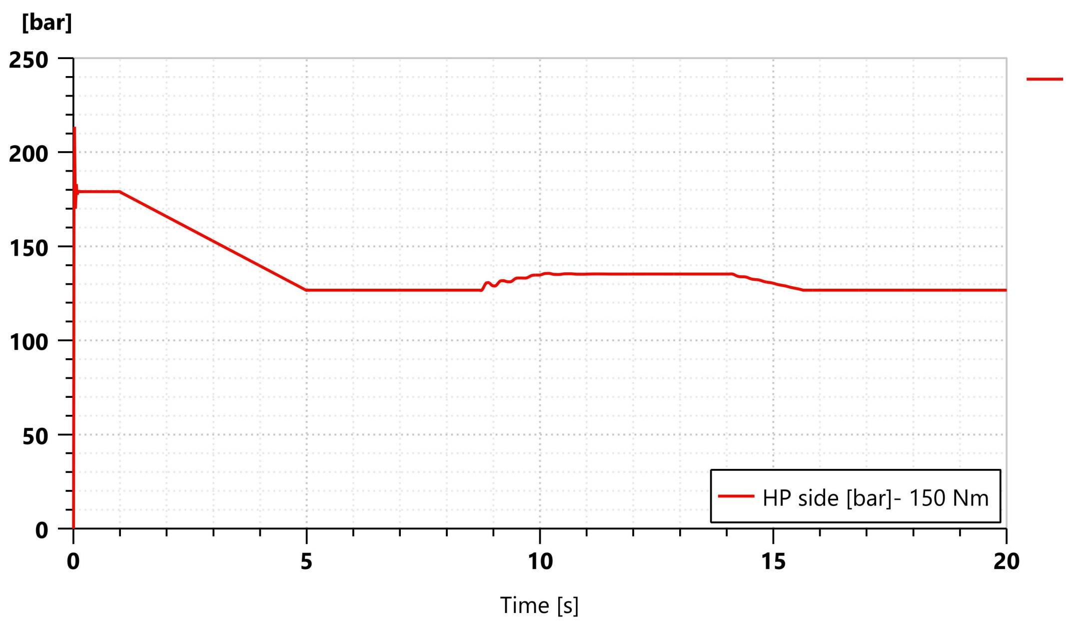

Figure 3.20: HP side overunning load 2^{nd} case

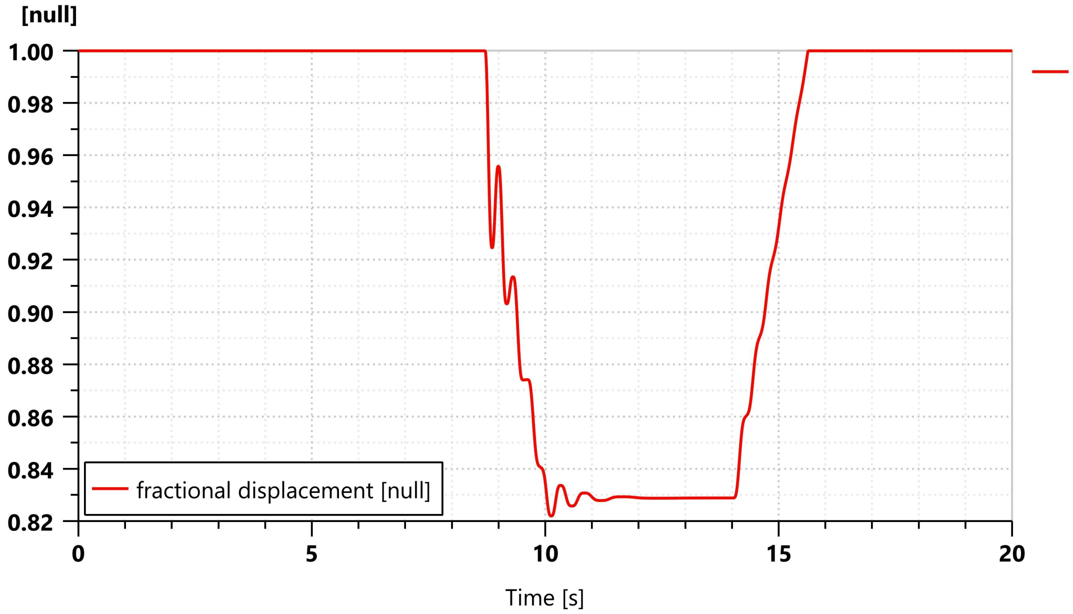

Figure 3.21: Fractional displacement - 150 $N \cdot m$

As in the previous case, it is also useful to analyze the high-pressure side graph (Fig. 3.20) in parallel with that of the motor's fractional displacement (Fig. 3.21). In this case, the pressure starts from a higher value due to the higher load, then remains constant for one second, when the pump volume is still zero. After that, it begins to decrease as the speed increases and the motor's displacement remains constant, until the pressure reaches a constant value corresponding to the constant command to the proportional control valve. In the second part of the simulation, when the engine displacement decreases due to the input command, the pressure slightly increases and then decreases when the fractional displacement comes back to 1 stabilizing at a value that is a little lower with respect to that achieved during the acceleration phase (fractional displacement decrement).

The next step is to observe the plot related to the output rotary velocity of the hydraulic motor shown in Fig.3.22.
Also this plot is strictly related to that of the fractional displacement. In fact, as can be seen from the figure, the velocity is equal to 0 during the first second of the simulation time, due to the setting of α_1. It then increases until it reaches the maximum pump speed when α_1 is equal to 1. After this initial phase, it remains constant at that value without receiving any command, until the solenoid of the proportional control valve is activated. The solenoid decreases the fractional volume and thus increases the rotational speed of the hydraulic motor. The speed reaches

a higher value until the driver wants to decelerate, at that point the fractional volume returns to 1, and the speed returns to the minimum possible value under these conditions, which is the same speed of the pump at its maximum fractional volume equal to 2000 rev/min.

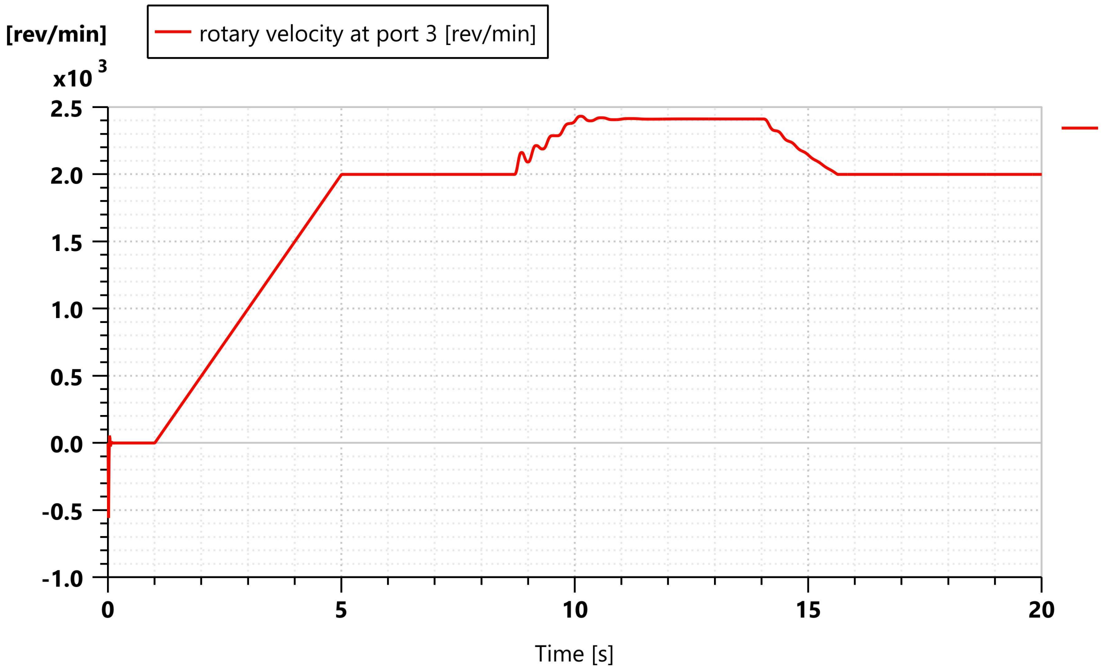

Figure 3.22: Output rotary velocity- 150 $N \cdot m$

It is now possible to analyze the trend of the torque produced by the hydraulic motor, visible in the following image in Fig.3.23. Being the moment, the product of the volume and pressure drop, its trend can be understood by looking at the variation of these two quantities. In this case, the moment starts from a constant value dependent on the applied overunning load at output shaft, remains constant for one second, and then decreases in correspondence with the increase in the pump's displacement. From that point on, it maintains a constant value for the entire duration of the simulation, except when the driver gives the command to accelerate first and then decelerate. In correspondence with that signal, the moment experiences a slight decrease due to the reduction in the motor's displacement, even though the pressure increases slightly. This means that, in this case, the variation in displacement has a greater influence on the moment compared to the difference in pressure.

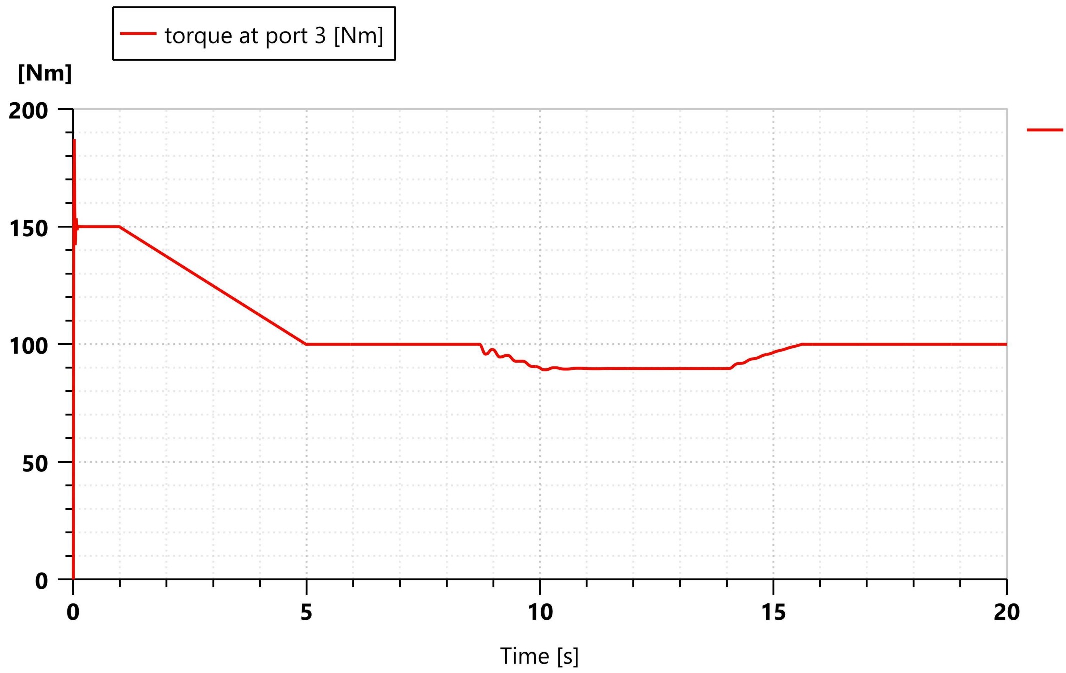

Figure 3.23: Output torque- 150 $N \cdot m$

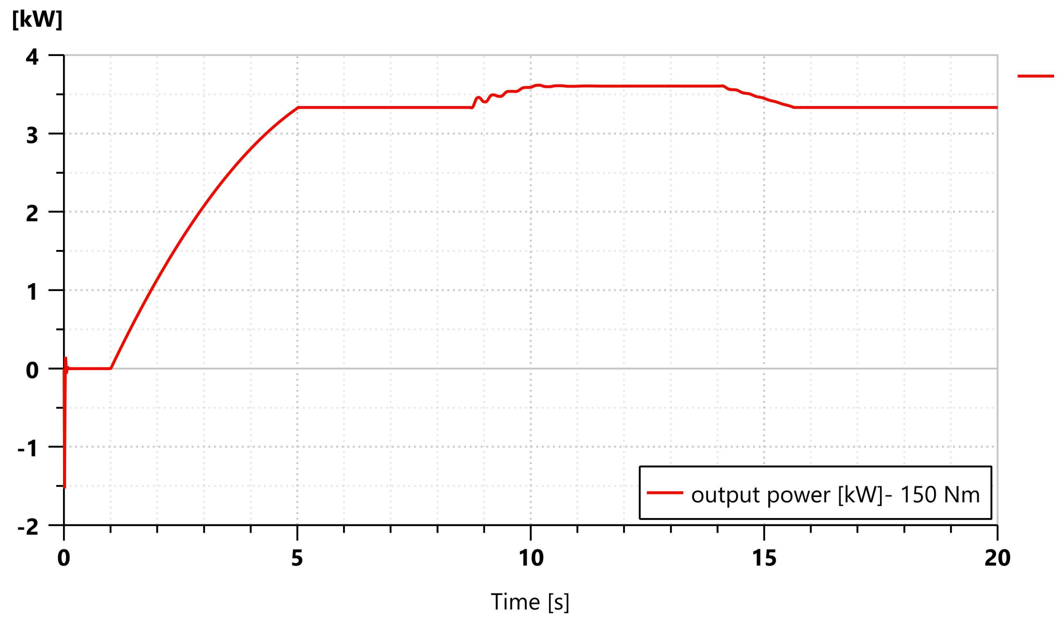

Figure 3.24: Output power- 150 $N \cdot m$

The last graph to consider for this simulation is the one related to output power depicted in Fig.3.24.

The output power is given by the product between the output torque of the motor multiplied by the output rotational speed of the secondary unit.

Considering this aspect, it can be seen how the power increases in correspondence with the increase of α_1, but not linearly as it does for the torque and for the speed. After reaching a constant value, the power remains steady until the solenoid of the proportional control valve is activated, at that point there is a slight increase in power due to the increase in speed, since the moment decreases during this phase. When the command is deactivated, the power returns to the value it had before the acceleration phase.

Chapter 4

Simulations setup and results of hydrostatic transmission with vehicle

The second part of simulations has been performed on the same model used before, but with the 1D vehicle model attached to the circuit of hydrostatic transmission. The model used is the one depicted in Fig.4.1. In particular, the set parameters in the circuit are the same as before, the input commands of the control valves are set according to the specific simulation.

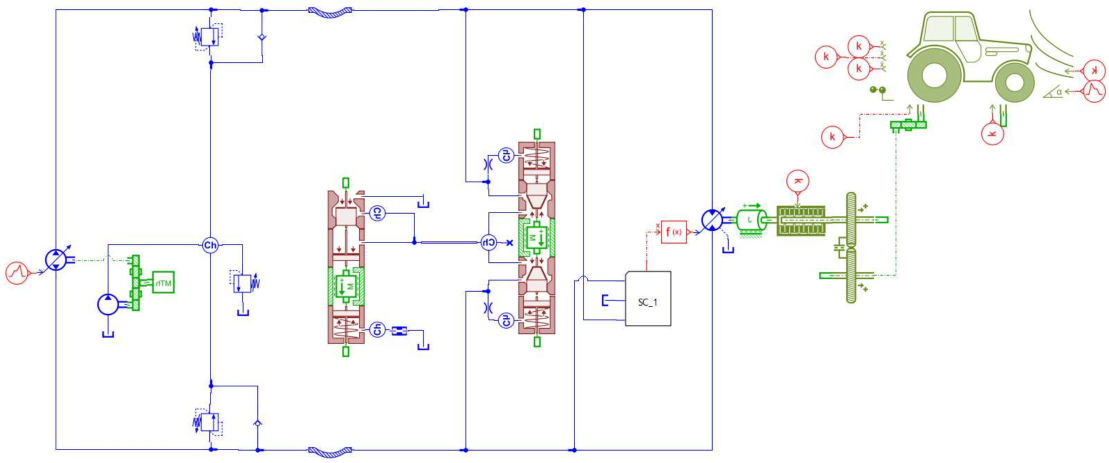

Figure 4.1: Model with vehicle used for simulations

In particular the set parameters for the 1D vehicle are listed in the following Tab.4.1.

Parameter	**Value**	**Unit**
Vehicle mass	9000	$[kg]$
Coulomb friction coefficient	0.02	$[-]$
Viscous friction coefficient	0	$[1/(m/s)]$
Windage coefficient	0	$[1/(m/s)^2]$
Air density	1.226	$[kg/m^3]$
C_x-Drag coefficient	0.9	$[-]$
Frontal area	4	$[m^2]$
Stiction coefficient	1.2	$[-]$
Front wheel radius	0.9144	$[m]$
Rear wheel radius	1.3208	$[m]$

Table 4.1: Vehicle parameters

The set parameters of the clutch are reported here in Tab.4.2.

Parameter	**Value**	**Unit**
Number of clutch contact faces	1	$[-]$
Maximum Coulomb friction torque for one contact face	500	$[N \cdot m]$
Rotary stick velocity threshold	1	$[rev/min]$

Table 4.2: Clutch parameters

The data of the transmission gears are reported below. In particular, the first gear, the one connected to the clutch, has the following set parameters (Tab.4.3).

For the other gear, the one connected to the reducer, the following parameters have been set (Tab.4.4).

The reducer that was described in Fig.2.29 has been set with a gear ratio equal to 7. For all the simulations has been set an ambient wind velocity equal to 0 m/s because for this kind of vehicles the aerodynamic forces are not so relevant.
This set of simulations has been divided into two types:

- vehicle on a flat ground;
- vehicle on slope;

Parameter	**Value**	**Unit**
Working pitch radius	10	$[mm]$
Working transverse pressure angle	25	$[degree]$
Helix angle	0	$[degree]$
Constant gear efficiency	0.99	$[-]$

Table 4.3: Gear 1 parameters

Parameter	**Value**	**Unit**
Working pitch radius	100	$[mm]$
Tip radius	17.2	$[mm]$

Table 4.4: Gear 2 parameters

The first set of simulations aims to verify the correct behavior of the hydrostatic transmission, taking into account that a vehicle on a flat ground assumes the same behavior of a resistant load for the circuit.
The second set of simulations wants to analyze the different behavior of the model with a positive slope (uphill road), so simulating a resistant load, and with a negative slope (downhill road), simulating an overunning load.

4.1 Simulation of the vehicle on a flat ground

This simulation has been set with the same variation of α_1 and α_2 depicted in Fig.4.2.

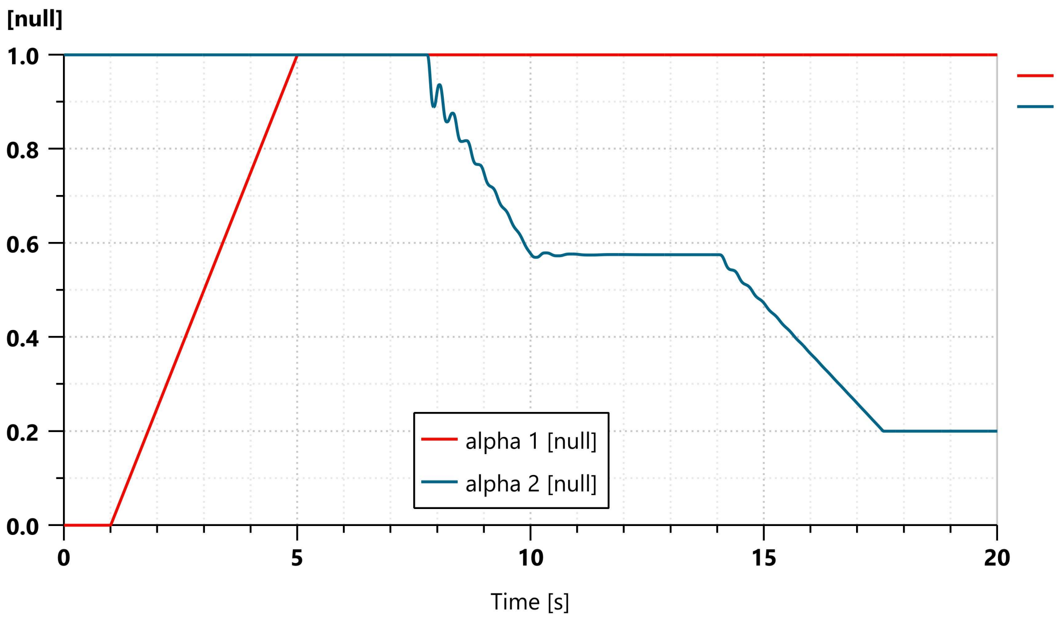

Figure 4.2: Alpha coefficients - vehicle

And the input command to the proportional control valve is described in the following table 4.5. This means that the primary unit achieves its maximum

Stage	Start value	End value	Unit	Duration
1	0	0	N	6 s
2	0	40	N	4 s
3	40	40	N	4 s
4	40	65	N	4 s
5	65	65	N	2 s

Table 4.5: Given command to proportional control valve

displacement, and thus its maximum flow rate, in 5 s and that the driver during the first 6 s does not give any input to the valve to increase the velocity.

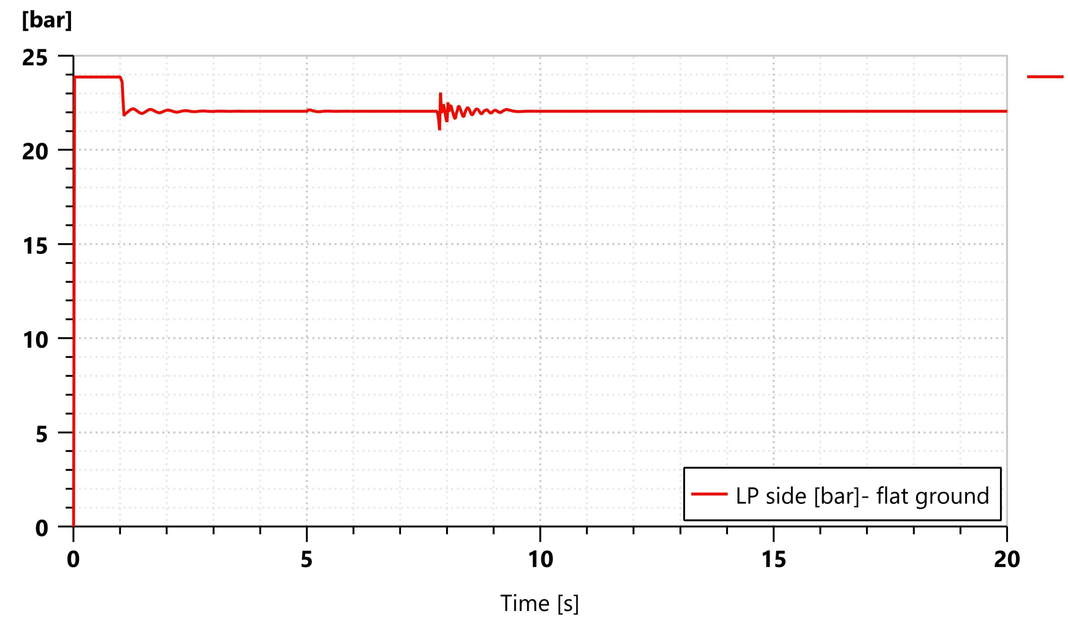

Figure 4.3: LP side in simulation with vehicle in flat ground

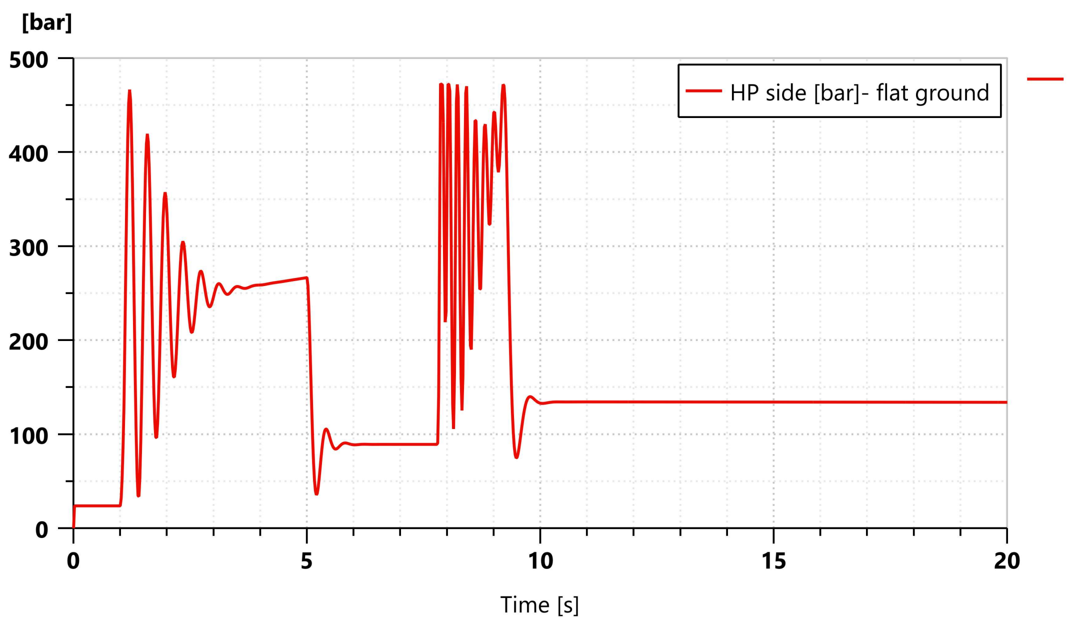

Figure 4.4: HP side in simulation with vehicle in flat ground

The trend for the low-pressure side is almost the same as that of the model without the vehicle, as can be seen in Fig.4.3, except for the first second of the simulation, because without oil flow rate, it's as if there is no load attached to the hydrostatic transmission, unlike the previous simulations. After that, the pressure reaches its steady-state value, which remains the same as before, confirming that it depends exclusively on the pressure relief valve design.

As for the low-pressure side, also for the high-pressure side (Fig.4.4) for the first second of the simulation, the pressure is constant and has the same value as the low-pressure side, while it shows various instabilities corresponding to the variation of α_1, which brings the pump displacement to the maximum, and in correspondence with the command to vary the displacement of the hydraulic motor. In the second half of the simulation the value of the pressure stabilizes at an acceptable value and to understand the reason of this behavior is useful to give a look at the plots related to fractional displacement and to PCOR displacement.

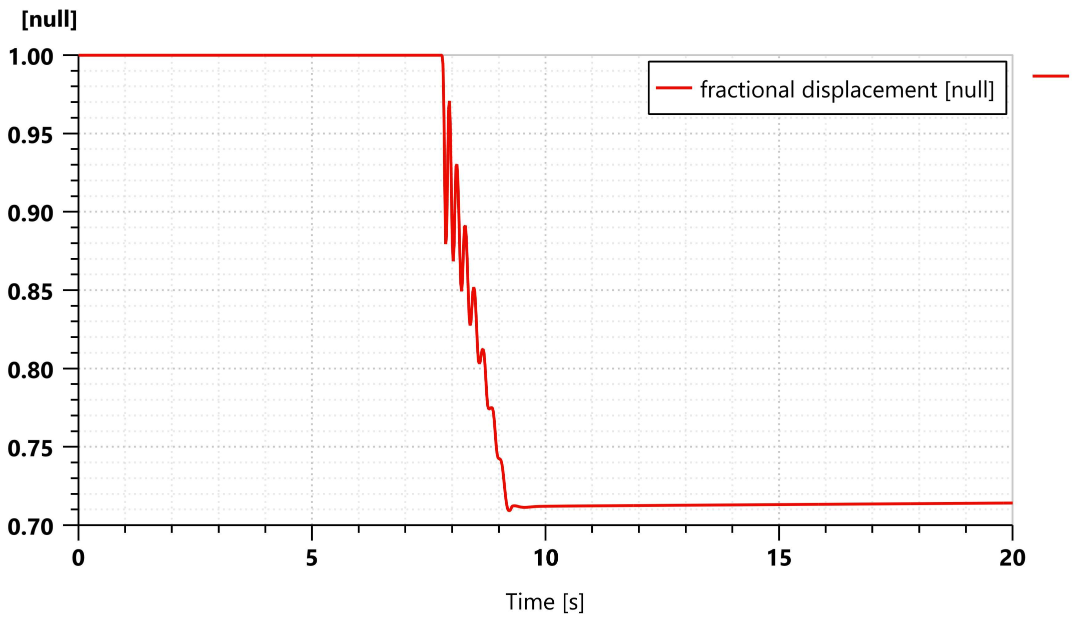

Figure 4.5: Fractional displacement - vehicle on a flat ground

The fractional displacement is depicted in Fig.4.5, and clearly shows that in this working condition the PCOR system intervenes to limit the pressure on the high pressure side, in fact the motor starts to decrease its displacement thanks to the driver's command, but after more or less 9 s of the simulation time, it stops at a value of about 0.71 and then slightly increases until the end of the simulation.
The fractional displacement plot is explained by the one that represents the PCOR

displacement in Fig.4.6.

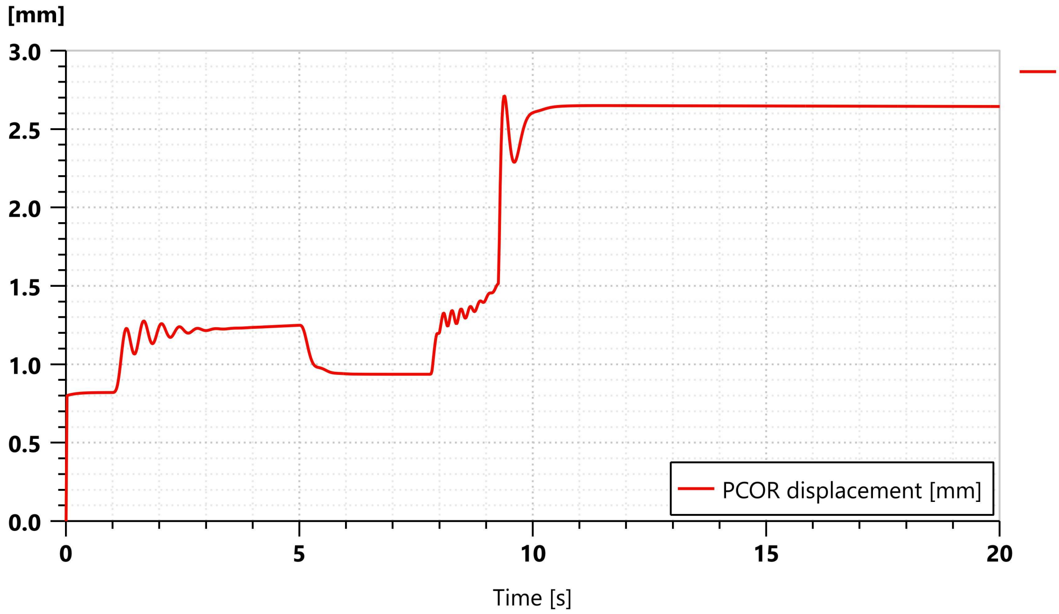

Figure 4.6: PCOR displacement - vehicle on a flat ground

Here, it is clearly visible that when the motor decreases its displacement, the valve moves to regulate the pressure at 9 *s* of the simulation time and then stabilizes its motion at the same value until the end of the test, in fact the displacement remains constant.

Now it is possible to analyze the outputs of the transmission to study the performance of the system and to understand if the vehicle moves consistently with what is expected.
The analysis starts from the output torque delivered by the hydraulic motor to the mechanical transmission that connects the vehicle to the system. The output torque is depicted in Fig.4.7. At a first sight the torque seems to have more or less the same trend of the pressure in the high-pressure side, but it is not identical because the torque is affected also by the motor displacement. In fact, in the first phase, the one with growing pump displacement and constant motor displacement, it has higher peaks with respect to the second phase of the simulation, differently from what happens in the HP side plot. After that the torque achieves its stability there are other oscillations with peaks that achieves lower values, in correspondence of the driver's command. At the end, the torque has a constant value when the motor displacement is maintained constant by the PCOR valve.

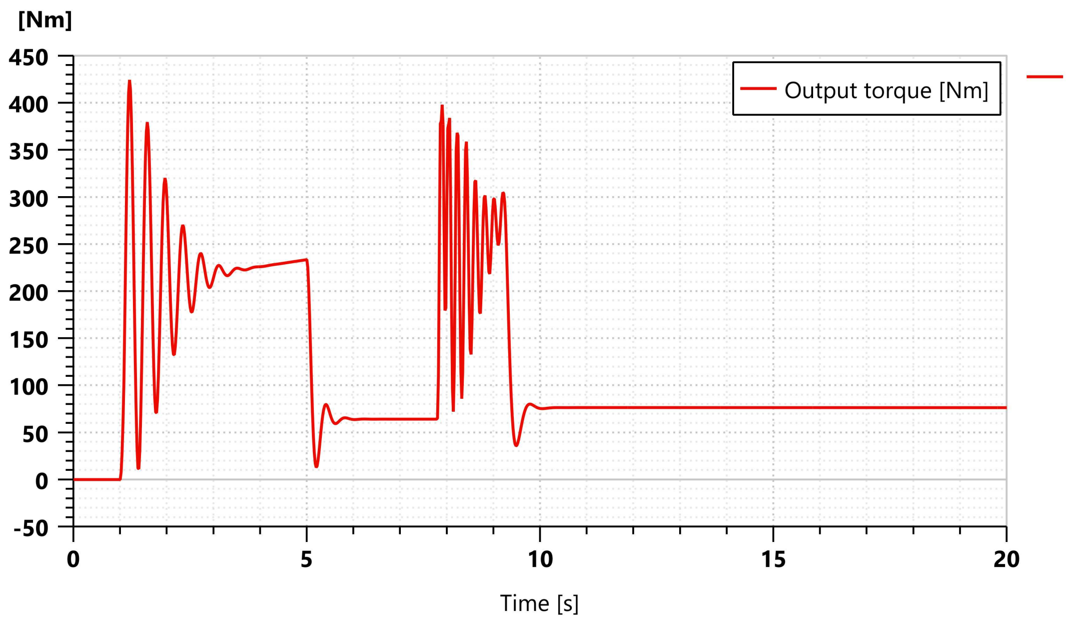

Figure 4.7: Output transmission torque - vehicle on a flat ground

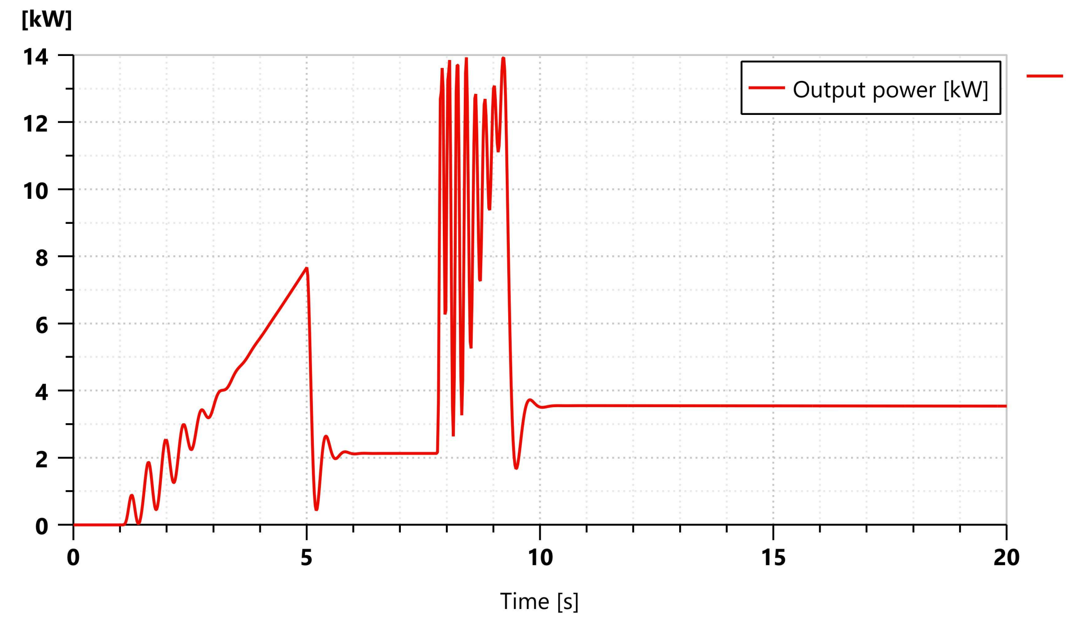

Figure 4.8: Output power of the hydrostatic transmission

Another output to take into account is the power delivered by the hydrostatic transmission to the vehicle. It is possible to see it in Fig.4.8, being the power the product between the delivered torque multiplied by the output rotational speed, it is easy to understand that at the start of the simulation the power is affected a lot by the low rotational speed, whereas it has the higher peaks during the acceleration phase, when both the speed and the torque have high values. In the second half of the simulation, the power maintains a constant value because torque and speed are also constant.

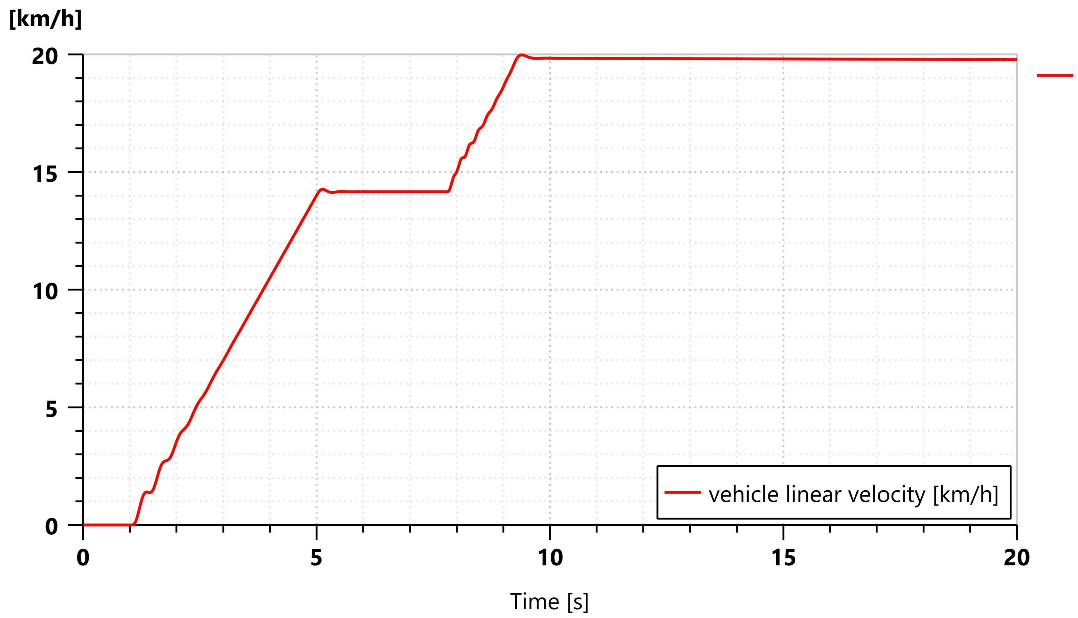

Figure 4.9: Vehicle linear velocity

Regarding the linear speed of the vehicle, we can look at the graph shown in Fig.4.9.
From the graph, it can be seen that the speed is obviously zero when there is no oil flow at the beginning of the simulation, then it increases linearly due to the variation of the coefficient α_1, reaching a value of about 14 km/h when the pump has reached its maximum volume. After this phase, the command given to the proportional control valve comes into play to increase the vehicle's speed, which will reach a maximum of about 20 km/h due to the intervention of the PCOR system that prevents the hydraulic motor from decreasing beyond its displacement in order to maintain a pressure under the threshold limit in HP side.

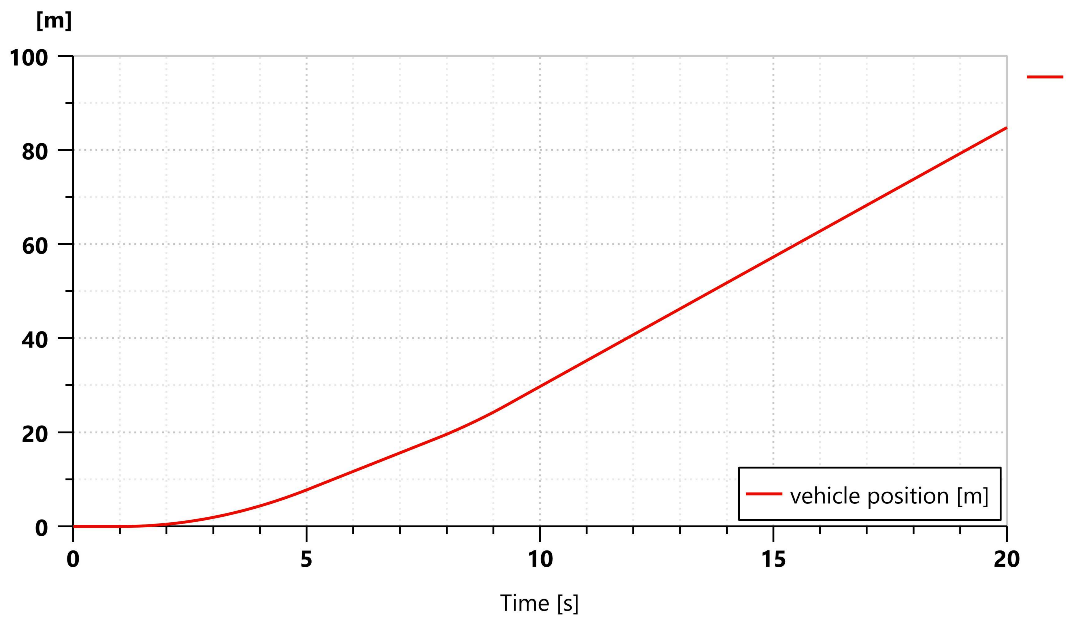

Figure 4.10: Vehicle position

Regarding the vehicle position during the simulation time, depicted in Fig.4.10, it is easy to understand that it doesn't move during the first second of simulation because there is no oil flow and so, no velocity. After that it starts to move and it is possible to notice a slight variation in the curve slope in correspondence of the acceleration phase, in fact the curve increases its slope between 8 s and 9 s and then proceeds linearly until the end.

4.2 Simulation of the vehicle on slope

In the second part of this chapter has been simulated exactly the same model, but supposing the vehicle on a road with a slope.

Both cases have been considered, with positive and negative slope, and also a third case with variable slope along the simulation.

4.2.1 Vehicle on a positive slope

A positive slope means a vehicle traveling along an uphill road. In particular, in the software the slope is expressed in percentage and taking into account that a percentage of 100% means an inclination equal to 45°, for this simulations has been set a constant value of 10%.

The coefficient α_1 and the input command to the proportional control valve have been set in the same way as before. The trend of the pressure in LP side is the same of the previous simulations, so the plot is not shown in this section.

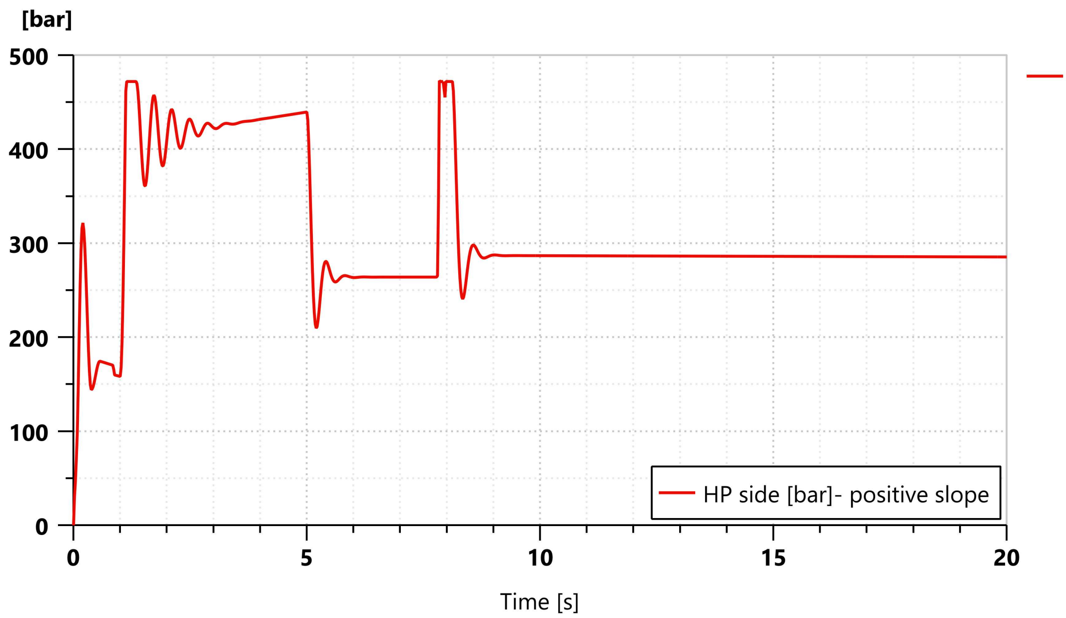

Figure 4.11: HP side - vehicle on a positive slope

The analysis starts from the HP side represented in Fig.4.11, in this case the pressure achieves high values already in the first second of simulation because the vehicle is in an uphill ground with no power. When the pump reaches its maximum displacement and when the driver gives the command to accelerate, the pressure peaks arrive to the maximum allowable value. After the last peak, the PCOR system intervenes and the pressure maintains a constant value until the end.

The pressure trend is affected by the fractional displacement that is the next plot to show in Fig.4.12. In this case is clear how the PCOR system plays a fundamental role because the achieved pressure is very high, in fact despite the driver's command, the fractional displacement never goes below a value of 0.89 and then tends to increase thanks to the control system.

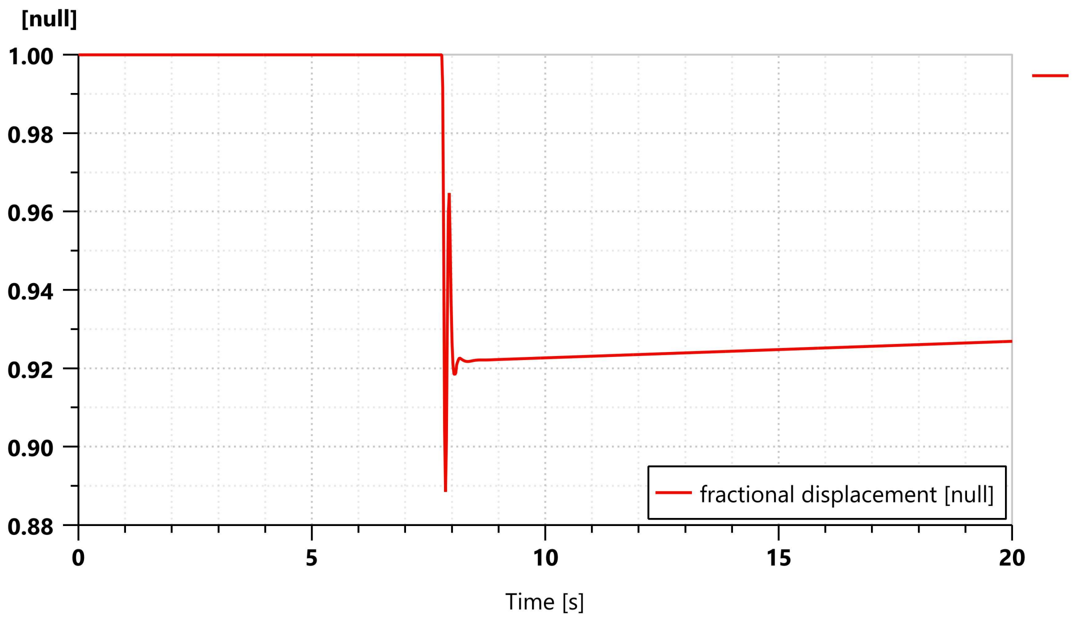

Figure 4.12: Fractional displacement - vehicle on a positive slope

The fractional volume graph is clearly closely related to that of the displacement of the PCOR valve, which will indeed be the next plot to analyze.

The PCOR displacement is depicted in Fig.4.13. It is possible to observe how the valve already regulates in the first phase, when there is the variation of α_1 and it reaches a displacement of around 1.5 mm. When the pump reaches its maximum displacement and the input command is not still able to move the proportional control valve, the PCOR valve comes back in rest position until the driver's command decreases the motor displacement. When this happens the PCOR valve moves along its entire stroke and regulates until the end of the simulation.

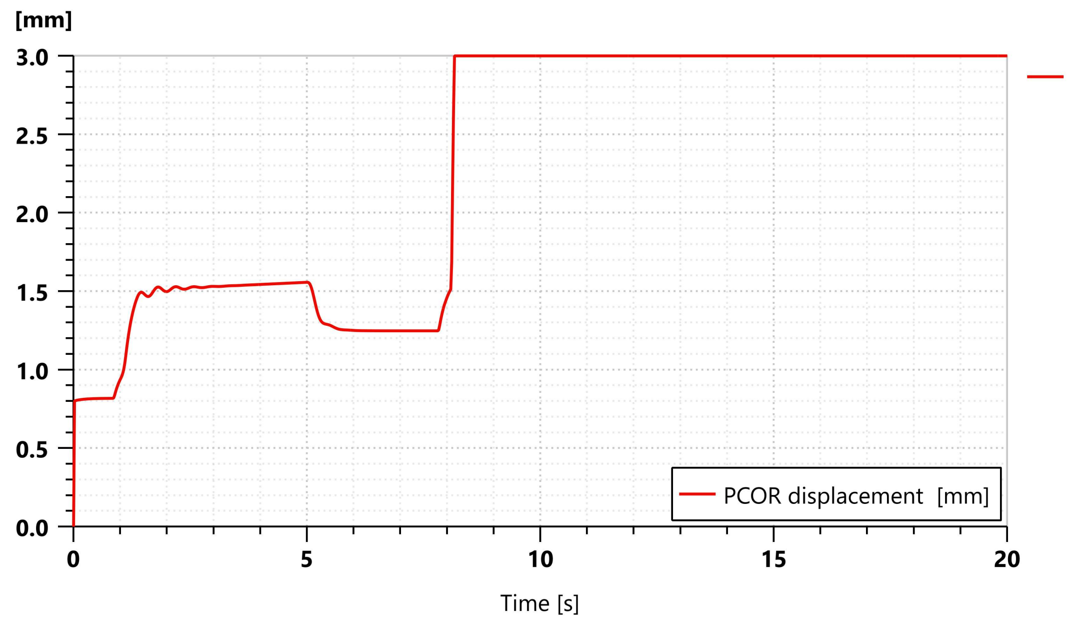

Figure 4.13: PCOR displacement - vehicle on a positive slope

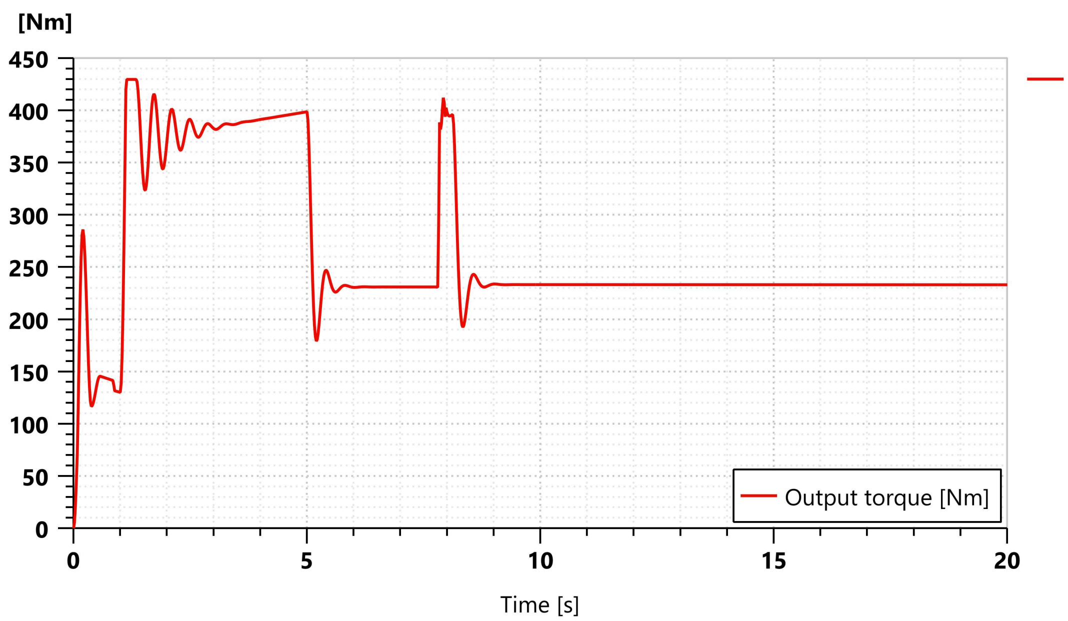

Figure 4.14: Output transmission torque - vehicle on positive slope

Regarding the output torque delivered by the hydraulic motor, in Fig.4.14, it almost follows the same trend of the pressure in HP side, with a small influence, in this case, of the pump and motor displacements. The reason is that the pressure reaches very high values, so it affects more the torque with respect to the displacement variation of both the hydraulic units.

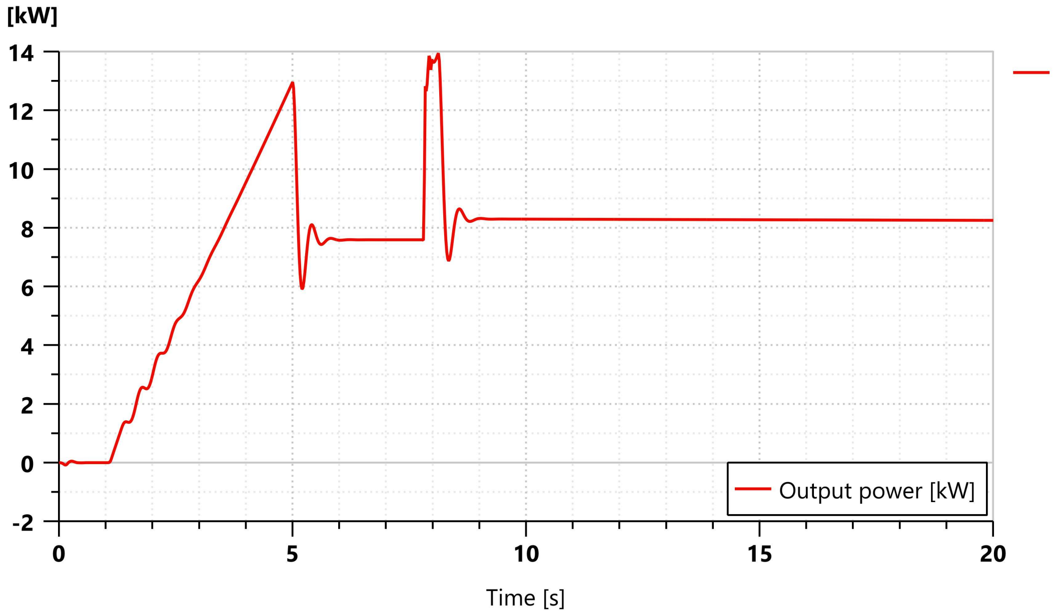

Figure 4.15: Output transmission power - vehicle on positive slope

The output power of the transmission in Fig.4.15, shows a linear increment in correspondence of the variation of α_1 up to 5 s, then it diminishes because the rotational speed reaches a constant value and the torque decreases. When the hydraulic motor decreases its displacement, the power achieves the highest peak in the simulation at around 14 kW and then stabilizes when the PCOR valve is in regulation position.

Regarding the linear velocity and the vehicle position represented in Fig.4.16 and in Fig.4.17.
The linear velocity has the same path up to 5 s, when it reaches the correspondent value to the maximum value of pump displacement and then there is a lower increase with respect to the previous case because of the slope.
Talking about the motion of the vehicle there are no significant differences, expect for the lower traveled distance due to the lower velocity.

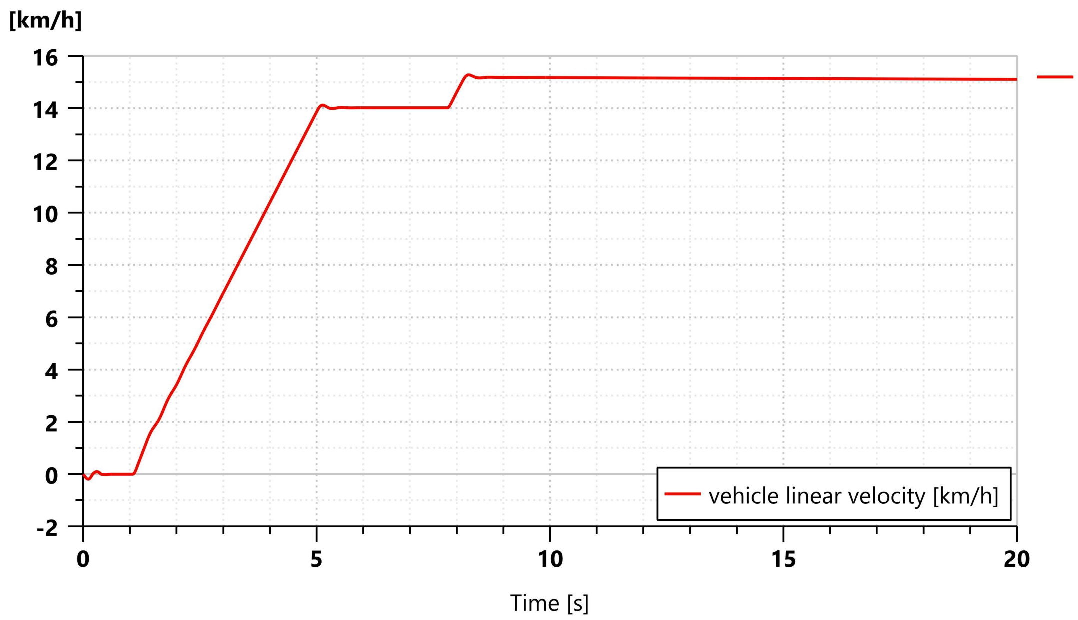

Figure 4.16: Vehicle linear velocity - positive slope

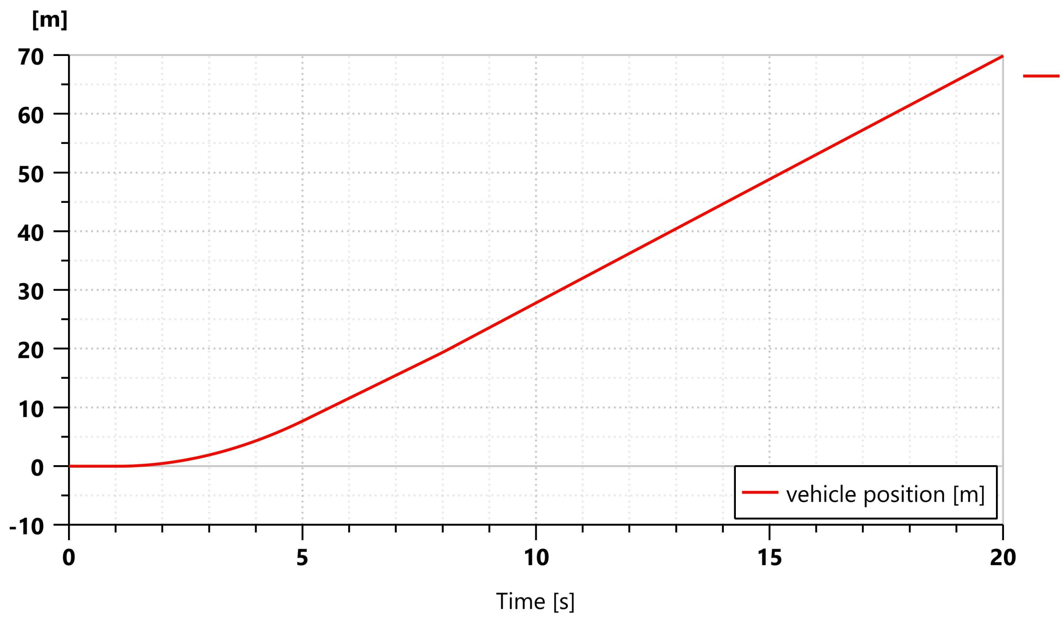

Figure 4.17: Vehicle position - positive slope

4.2.2 Vehicle on a negative slope

A negative slope means a vehicle traveling along a downhill road. For this simulations has been set a constant value of -8%. Regarding the input command to the proportional control valve, in this case has been set to 0 N, because in a situation like this, the driver of an off-highway vehicle does not want to accelerate.

A slight modification has been done also to α_1, in order to reach the maximum flow rate of the circuit faster. The variation of the coefficient has been set like depicted in Fig.4.18.

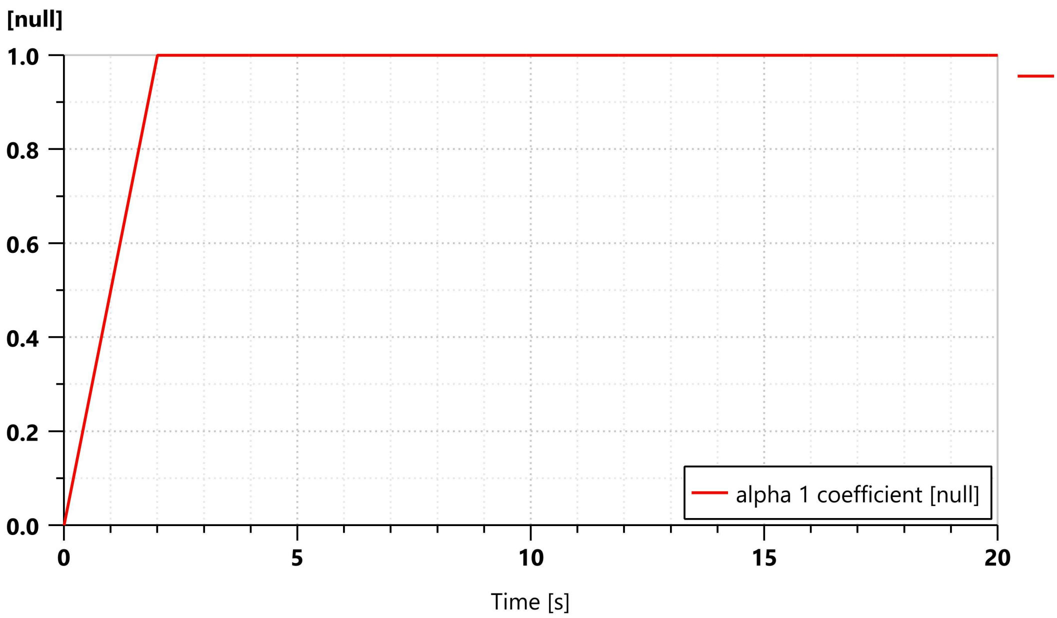

Figure 4.18: α_1 coefficient - vehicle on a negative slope

Unlike the case of the previous chapter where the overunning load did not affect the circuit behavior until there was a certain amount of fluid, now the vehicle in a downhill road always affects the system, for this reason has been chosen for this modification.

The LP side trend pressure represented in Fig.4.19 shows that with a low amount of flow, the circuit does not behave like if it has an overunning load for the first 2 s, because the direction of rotation of the pump remains the same and due to the friction torque of the rotary load. Once the maximum flow rate it is reached, the pendulum valve moves to the opposite direction and the circuit assumes the configuration with an overunning load and there is the switch of the branches. The achieved value in steady-state conditions is always the same.

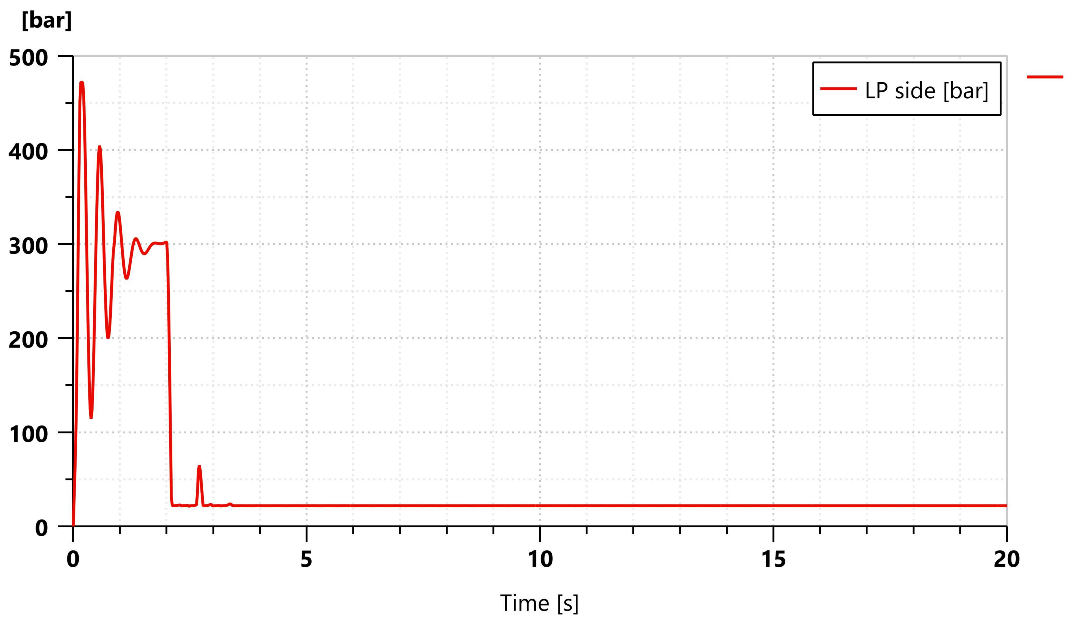

Figure 4.19: LP side - vehicle on a negative slope

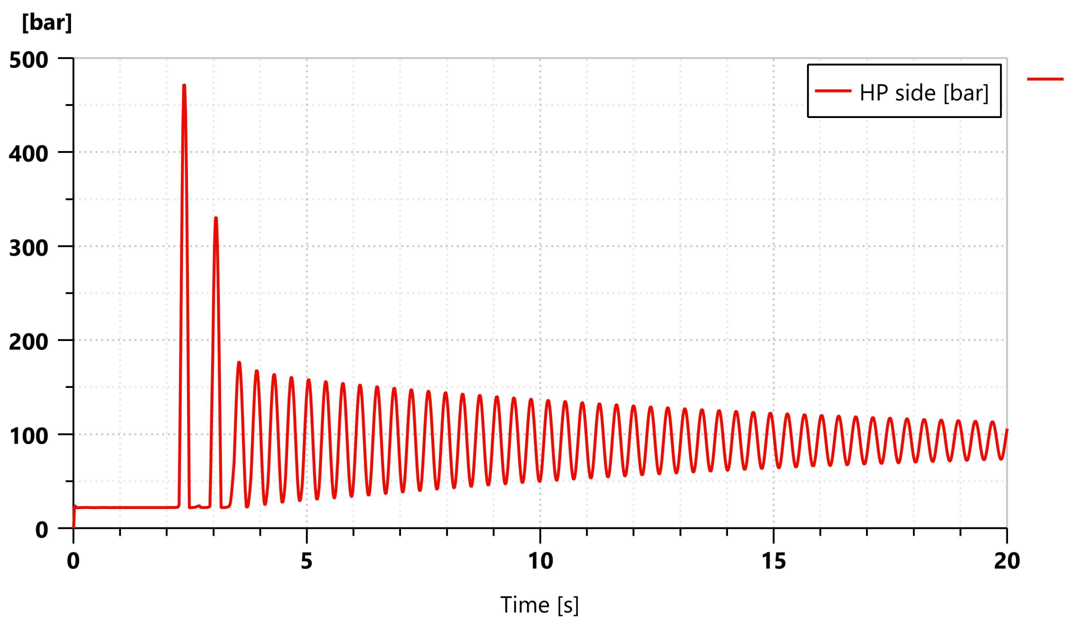

Figure 4.20: HP side - vehicle on a negative slope

Regarding the HP side plot in Fig.4.20, also here is clearly visible the switch of the circuit branches after 2 *s* of the simulation time. After this first phase, the curve assumes an oscillatory behavior decreasing the amplitude along the simulation. Regarding the plot related to the fractional displacement it is not useful because without input command it maintains a constant value equal to 1, so the maximum motor displacement.
For the same reason the PCOR valve never regulates in this case, so the plot representing its motion is not shown for this simulation.

Now it is possible to give a look at the plot of the output torque delivered by the hydrostatic transmission in this case. It is represented in Fig.4.21. Also in this graph is clearly visible the switch between the two branches of the circuit after more or less 2 *s*, because the torque changes sign from drive torque to resistant torque. After that it has the same trend of the high-pressure, in fact, the other parameter affecting the torque, the motor displacement, is constant.

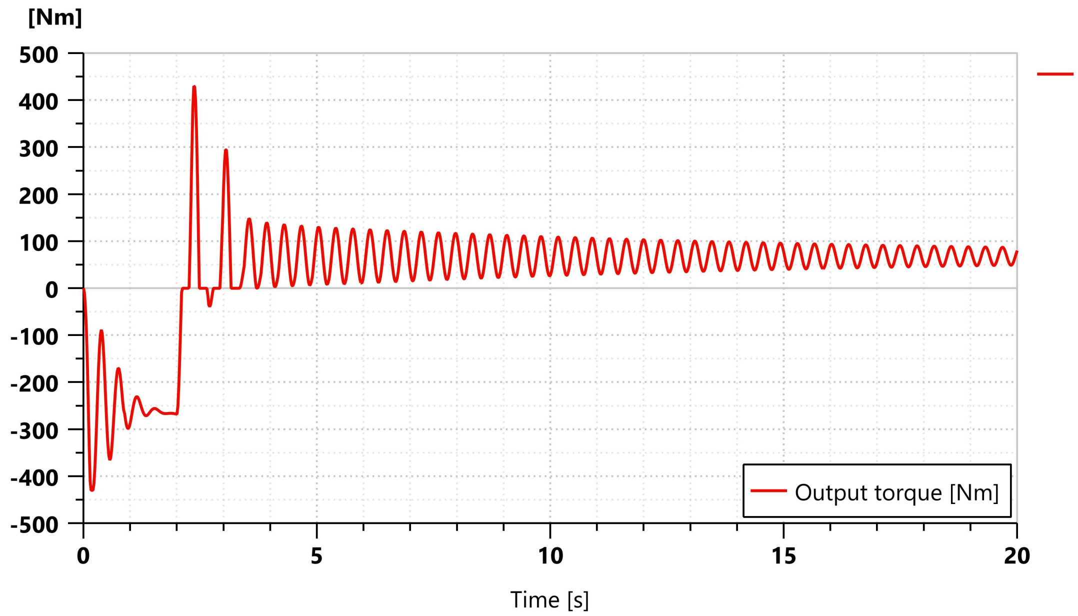

Figure 4.21: Output transmission torque - vehicle on negative slope

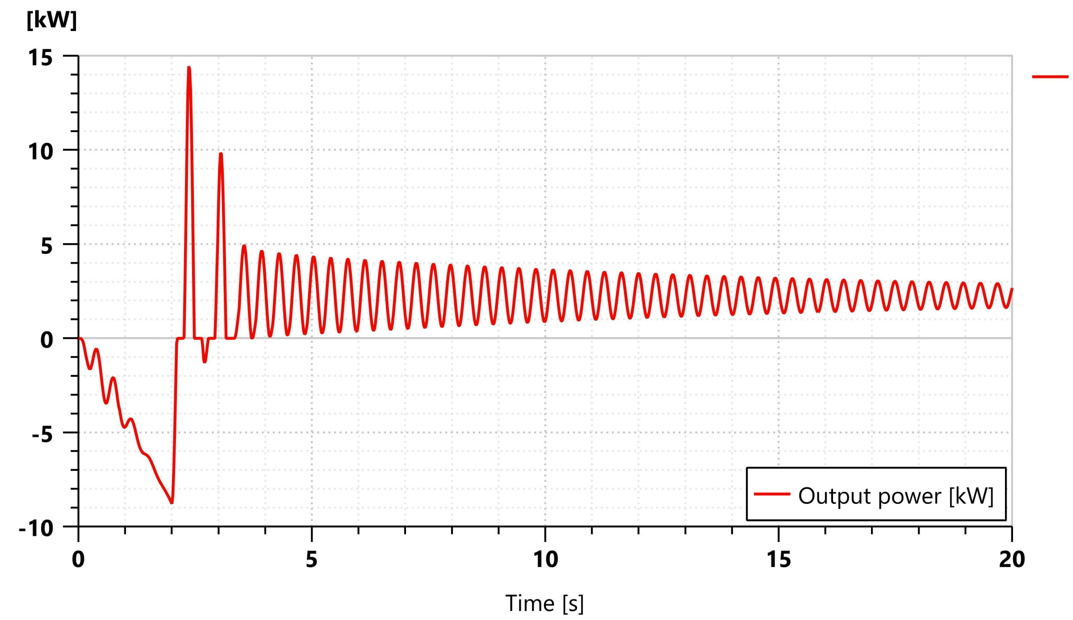

Figure 4.22: Output transmission power - vehicle on negative slope

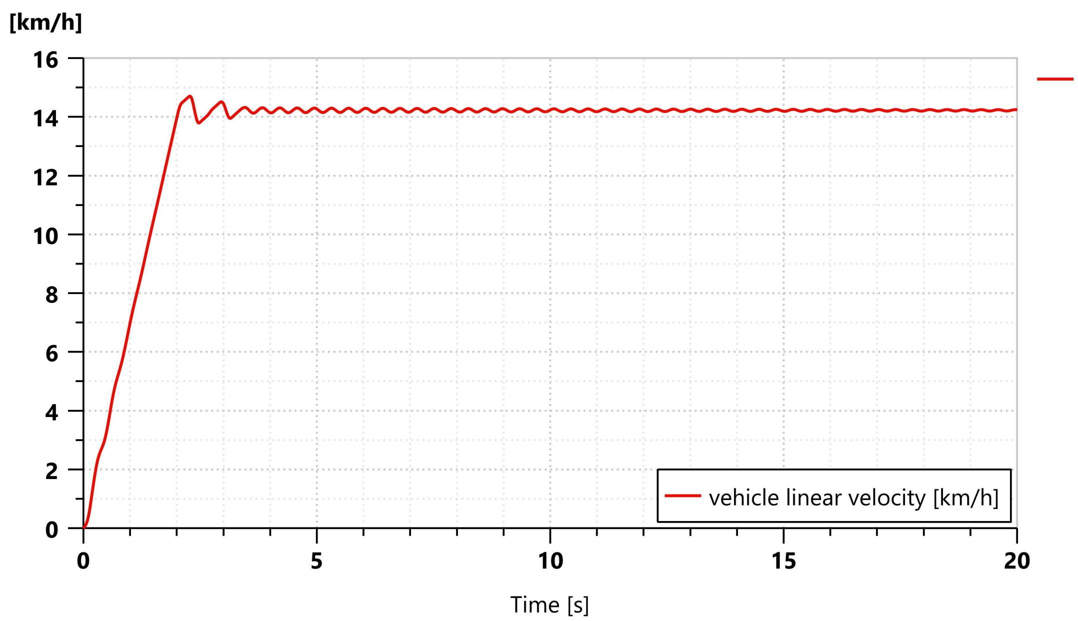

Figure 4.23: Vehicle linear velocity - negative slope

Analyzing the plot related to the output power in Fig.4.22, it has the same trend shown before by the torque and the pressure, with slight modifications due to the output rotary velocity of the motor. The velocity of the vehicle in Fig.4.23, increases linearly until the pump reaches its maximum displacement and then remains constant because there is no input to the proportional control valve to decrease the motor displacement.

The profile of the vehicle motion during the simulation, in Fig.4.24, always shows the same trend with the slope of the curve that depends on the vehicle's velocity, in this case the vehicle travels more meters with respect to the previous case.

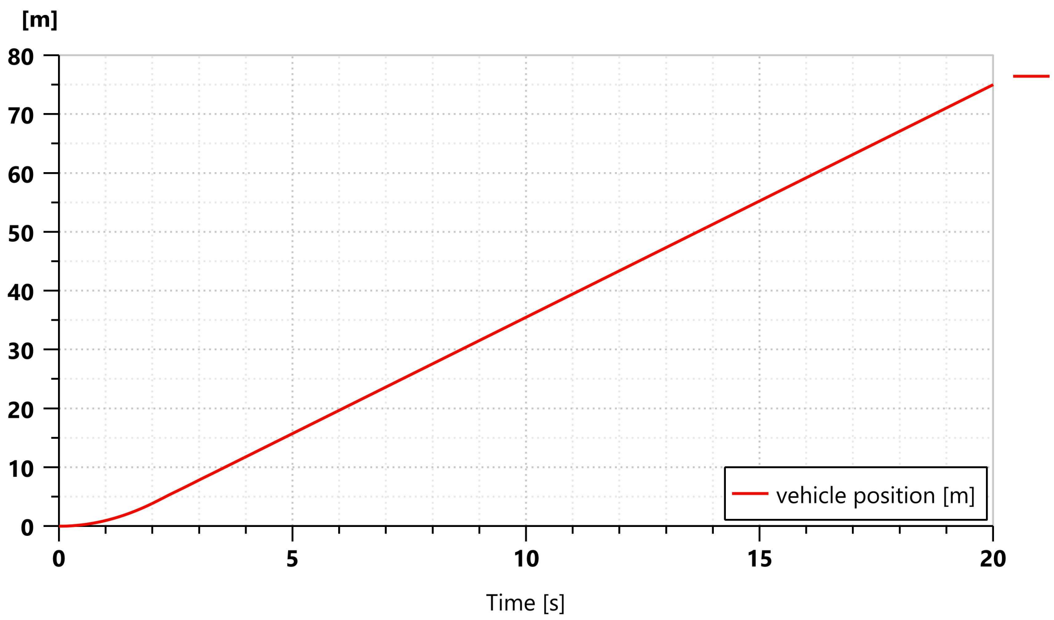

Figure 4.24: Vehicle position - negative slope

4.2.3 Vehicle on a variable slope

For this simulation has been set a variable slope to understand the behavior of the vehicle in this condition. In particular, the variation of α_1 is the same of the previous case depicted in Fig.4.18. The variation of the slope has been set according to the following table 4.6. Regarding the input command to the proportional control valve, it has been set like it's described in Tab.4.7.
For the first 2 s there is no input because of the variation of α_1.

Stage	Start value	End value	Unit	Duration
1	0	0	%	4 s
2	0	6	%	6 s
3	6	-4	%	10 s

Table 4.6: Slope variation

Stage	Start value	End value	Unit	Duration
1	0	40	N	2 s
2	40	60	N	4 s
3	60	60	N	6 s
4	60	20	N	2 s
5	20	0	N	6 s

Table 4.7: Given command to proportional control valve - variable slope

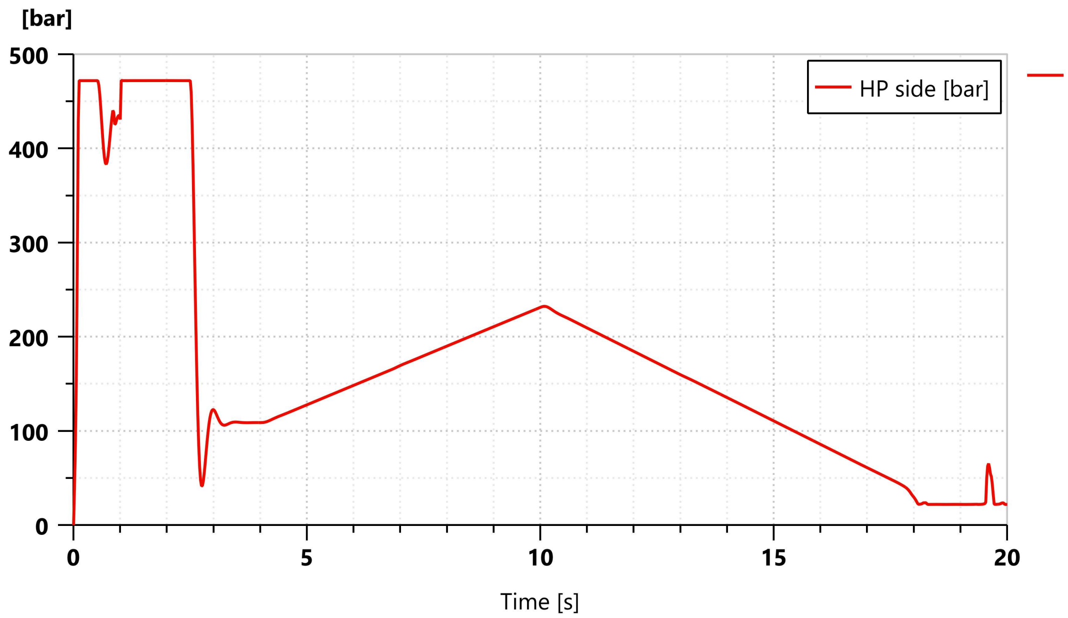

Figure 4.25: HP side - variable slope

Starting the analysis from the pressure sides, in Fig.4.25 is depicted the HP side that at the end of the simulation becomes the LP side because of the negative slope.

During pump displacement variation, the pressure reaches a very high value. When the displacement is maximum, the trend is affected by proportional control valve, in fact, the pressure decreases when the solenoid force is still not able to move the valve until the valve moves and the pressure rises again up to the peak reached at 10 s due to the command of the driver. After that, the valve has reached its maximum stroke, the pressure tends to decrease, due to the slope variation and due to the driver that now wants to decelerate with a downhill road. The pressure decreases until it reaches the set value of the pressure relief valve and, due to this, it is possible the switch of the branches because at the end of the simulation the circuit behaves like it has an overunning load at the output shaft.
Looking at the other branch in Fig.4.26, it behaves like the LP side until the vehicle approaches the negative slope, then the pressure has some peaks due to the command that decrease from 60 N to 0 N and at the end without command and with negative slope, the pressure stabilizes at a value around 180 bar.

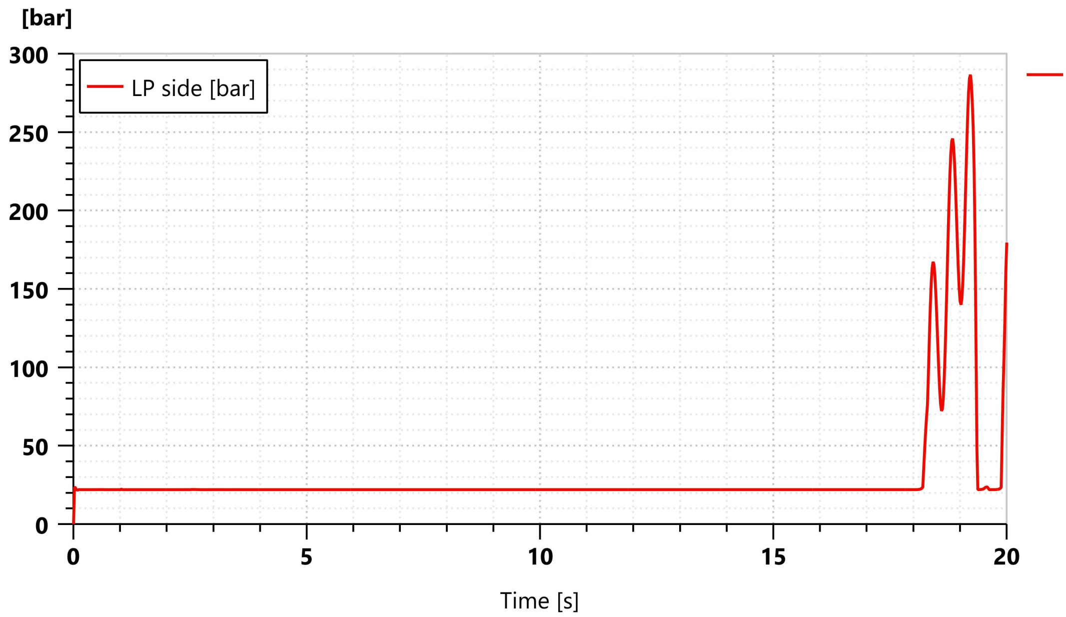

Figure 4.26: LP side - variable slope

Now it is useful to analyze the plots related to the fractional displacement of the hydraulic motor and that related to the PCOR valve displacement. The fractional displacement is shown in Fig.4.27 and is visible how the motor maintains high displacement even with the command due to the PCOR regulation.

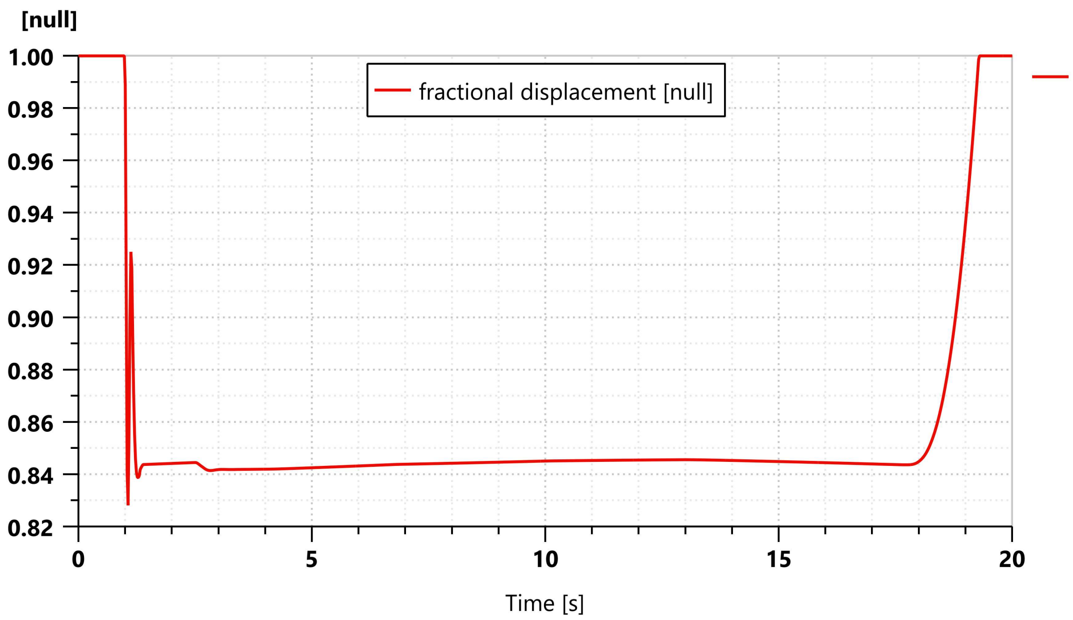

Figure 4.27: Fractional displacement - variable slope

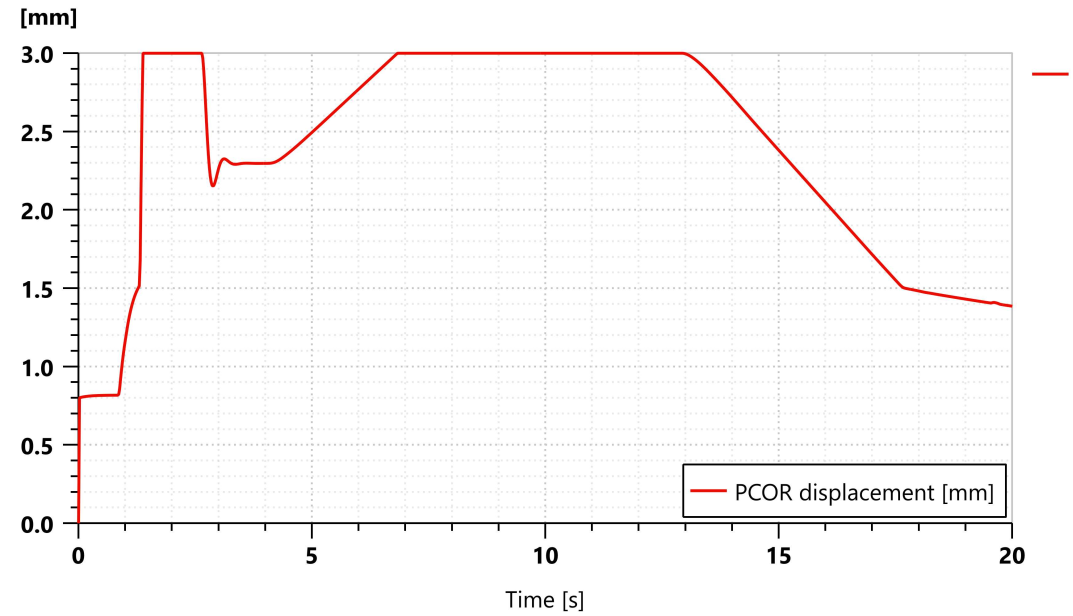

Figure 4.28: PCOR displacement - variable slope

In fact, looking at its displacement, in Fig.4.28, the valve regulates for the entire duration of the uphill road, reaching its maximum displacement, and then returns to its rest position when the slope becomes negative and due to the fact that the command goes to 0 N.

The output torque delivered by the hydraulic motor is represented in Fig.4.29. It is clear that it assumes both positive and negative values because in this simulation the vehicle acts like a resistant load and as an overunning load. It switches from a drive torque to a resistant torque in the last seconds of the simulation time.

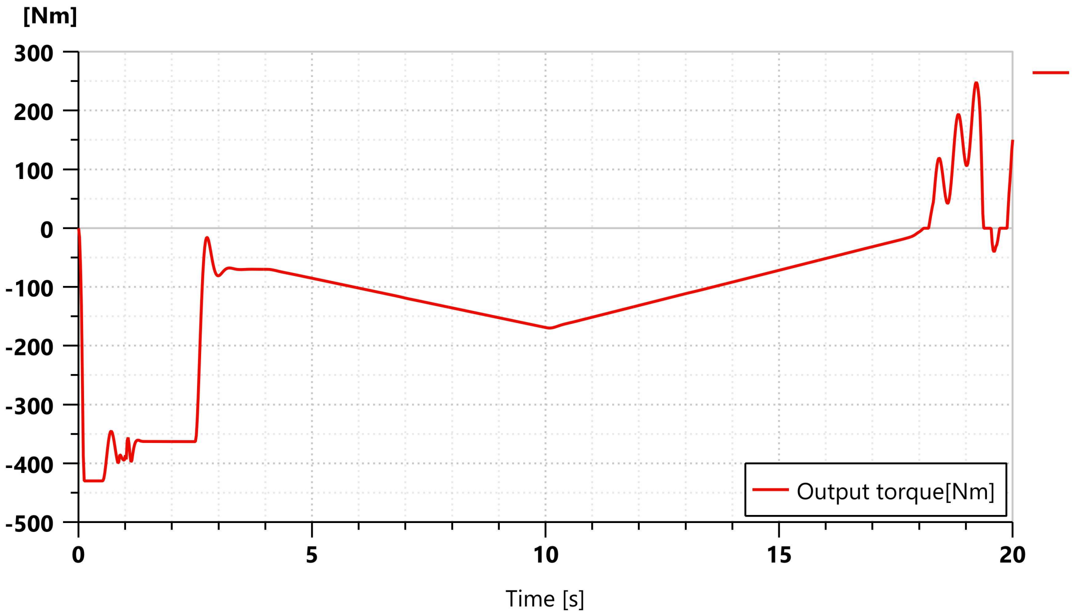

Figure 4.29: Output transmission torque - variable slope

The torque follows more or less the same trend of the pressure in the HP side with a minor influence of the displacements of the hydraulic units.

Regarding the output power of the transmission depicted in Fig.4.30, it is evident how it is influenced by the rotational speed of the engine, which in turn is affected by that of the pump, especially in the first few seconds of the simulation time, in fact having a torque almost constant the trend is linear like that related to the velocity due to the variation of α_1. In the final part of the simulation, when the transmission changes working modes, the power assumes a positive sign.

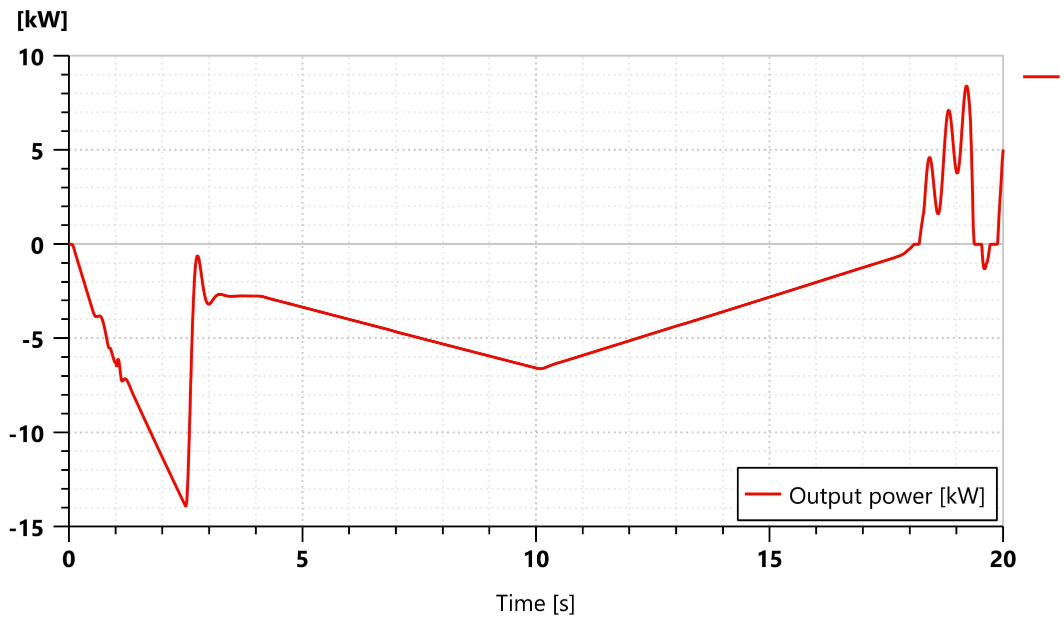

Figure 4.30: Output transmission power - variable slope

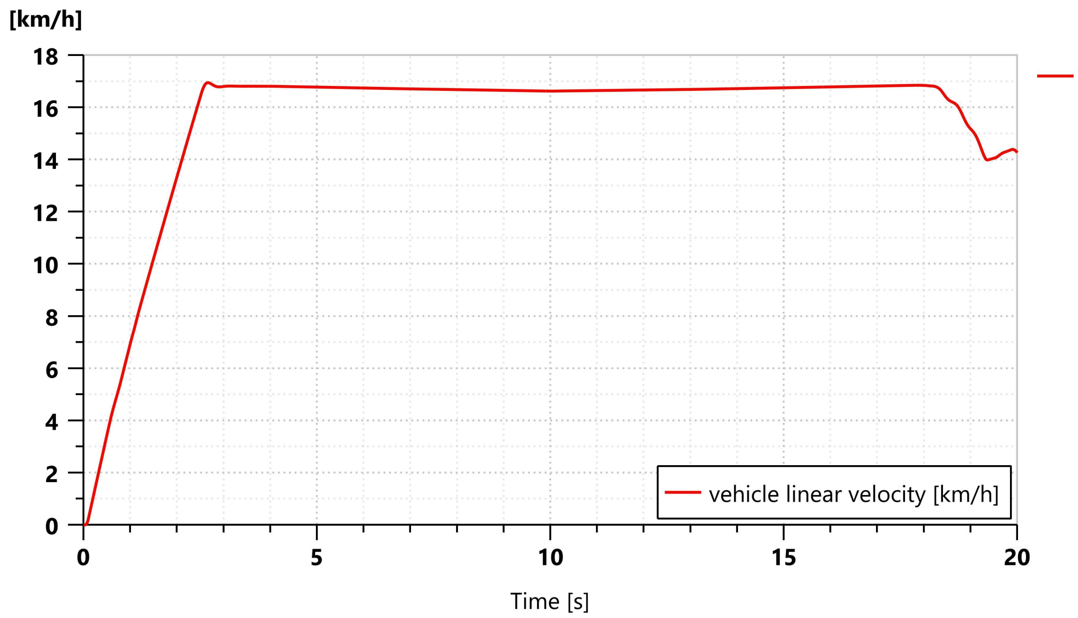

Figure 4.31: Vehicle linear velocity - variable slope

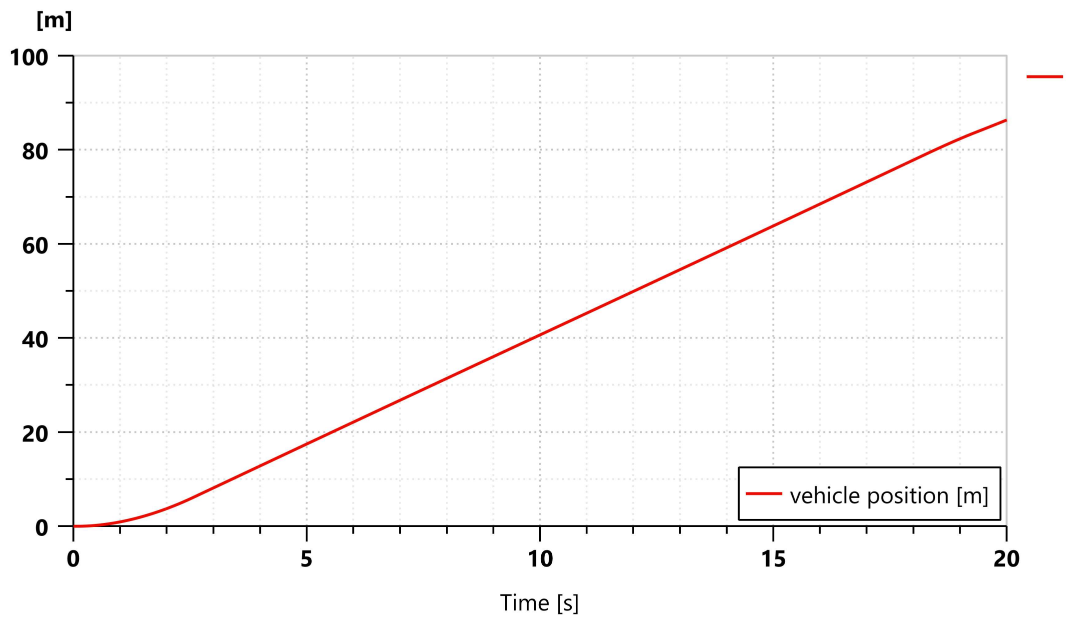

Figure 4.32: Vehicle position - variable slope

Regarding the vehicle linear velocity in Fig.4.31, it is visible that it increases up to the value determined by maximum pump displacement (around 14 km/h), and then there is still an increment due to the driver's command, in fact the vehicle reaches 17 km/h. At the end of this phase, the velocity maintains a constant value along the positive slope. When the vehicle approaches the negative solpe and the driver does not give any command, the velocity decreases to the minimum value set by the displacement of the hydraulic units (14 km/h).
The last plot in Fig.4.32 shows the vehicle motion and can be observed a slight change in the slope of the curve when there is the deceleration, between 18 s and 20 s. There are no other variations because the velocity increases linearly in the first part of the simulation and maintains constant value up to the deceleration phase.

Chapter 5

Conclusion

In conclusion, this thesis has provided a comprehensive analysis and simulation of a hydraulic motor within a hydrostatic transmission, emphasizing its critical role in enhancing the efficiency and versatility of industrial and mobile applications. Through the use of advanced simulation software, Simcenter Amesim, we have successfully modeled the key components of the transmission and conducted a thorough investigation into the motor control systems.

The findings from our simulations, both in isolated circuit conditions and in conjunction with vehicle dynamics, have demonstrated the operational effectiveness of the hydrostatic transmission under various load scenarios. By examining the system on flat ground and inclines, we have gained valuable insights into its performance across different working conditions.

While this research highlights the numerous advantages of hydrostatic transmissions, like the precise control, the compactness, the energy efficiency and the reliability, it also identifies certain drawbacks and potential areas for improvement. The main drawbacks are the cost, the maintenance, the fluid leakages and the temperature sensitivity, whereas some areas for future improvements could be the development of advanced materials for more durable and sustainable hydraulic fluids or the integration of artificial intelligence technologies to improve control and efficiency. This dual focus not only contributes to a deeper understanding of hydraulic motor dynamics but also paves the way for future innovations in the field. Ultimately, the insights gained from this study can inform the design and optimization of more efficient hydrostatic transmission systems, benefiting a wide range of applications in the industry.

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[2] *H1 Bent Axis Variable Displacement Motors*. Sauer Danfoss. 2012 (cit. on pp. 8, 13, 14, 19, 22, 26).

[3] *Simcenter Amesim Help*. 2024: Siemens (cit. on pp. 35–38, 42, 46–51).