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Review and preliminary modelling of electro-hydraulic steering system for off-road vehicles

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Abstract

Nowadays, the importance of accuracy, reliability and energy efficiency is increasing in most of vehicle sectors, including off-road applications. This process is speeding the development of advanced systems, in particular this thesis work will focus on the steering systems and new prospective regarding their structure and management. Above all, electro-hydraulic steering architecture represents a potential solution leading to the integration of new electronic control functions and autonomous driving features. This thesis work started from a first review of electro-hydraulic steering system for off-road vehicles, focusing on the structure, operating principle and main components of the solutions found in scientific literature, academic research and industrial implementations. Furthermore, is provided a basic overview of the control strategies that could be used in such systems to optimize the functioning. Following the literature review, a preliminary dynamic model of some of the most interesting solutions is developed. The models are made and simulated in the Amesim workspace with the focus on the verification of the operations of these systems; the models are designed considering simplified boundaries thus reducing the computational time. The analysis will pay attention to the pressure and flow rate evolution for better understanding the behavior of these steering systems, ending with a brief look to the energy efficiency of the different solutions. Simulation results demonstrate that the electro-hydraulic steering systems analyzed can ensure high accuracy and reliability, also implementing the autonomous driving feature, enhancing the efficiency of the steering architectures. The models developed could be considered a preliminary framework for future investigation aiming at system optimization, new control strategy and evolution of electro-hydraulic steering system for off-road vehicles.

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Introduction

The growing importance of energy efficiency and safety is leading to significant developments in all the systems implemented in vehicles, including the steering system. In off-road vehicles, such as wheel loaders and agricultural machines, steering systems operate under severe working conditions while reliability and steering accuracy must be ensured at a high level; in Figure 1 an example of wheel loader is shown. This is due to the influence of the steering system on the maneuverability, stability, safety of the vehicle and the operator comfort, making the study and development of new architectures both interesting and critical.



Figure 1: Wheel Loader

The steering systems for off-road vehicle applications typically rely on mechanical-hydraulic architecture where the steering wheel is mechanically connected to the hydraulic power steering system. In Figure 2 an OSPC valve is shown, it is possible to see in front of the image the site where the mechanical link takes place. However, the introduction of advanced driver assistance systems (ADAS) is pushing

manufacturers to the development of electro-hydraulic steering systems, making the architecture more flexible. Indeed, this type of system can combine the high power density of hydraulic actuation, due to the harshness of the operating environments, with the typical controllability of electronic systems. In Figure 3 it is possible to see an Electro-Hydraulic valve, instead of the location for the mechanical connection the electric link is present.



Figure 2: OSPC valve used in hydraulic power steering system

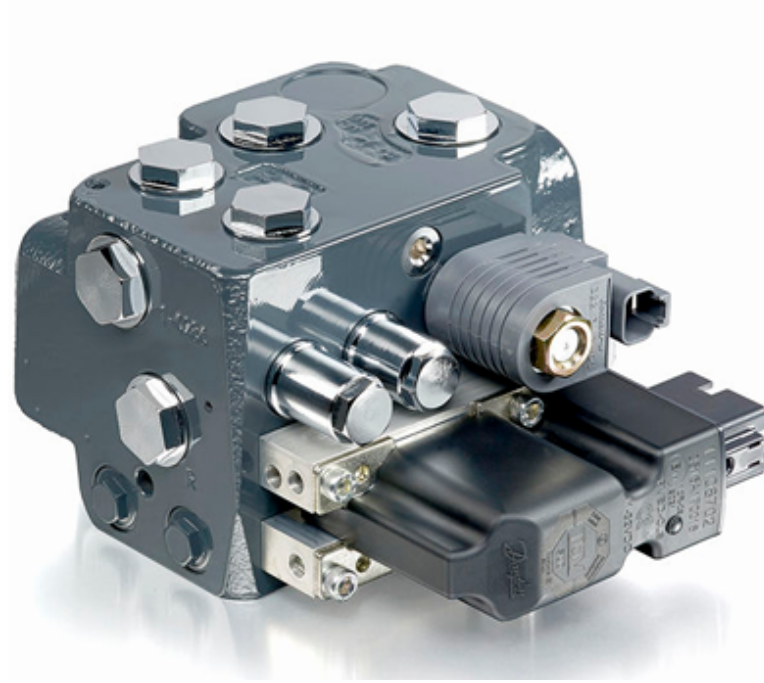


Figure 3: Electro-Hydraulic valve used in EH power steering system

In this context, the aim of this thesis is to review the main electro-hydraulic steering architectures currently developed for off-road vehicle applications and investigate the behavior of the different solutions. In addition, a preliminary study on the potential implementation of autonomous driving functionalities is carried out. For the most interesting solutions, simplified models are developed in Amesim to analyze the behavior and better understand the functioning of each component of the electro-hydraulic system. To investigate the autonomous driving capabilities the hydraulic models are supported by a PID control system that is carefully tuned to ensure the steering accuracy and system responsiveness. Moreover, besides the first model, the electro-hydraulic solutions investigated are steer-by-wire systems, the peculiarity of this type of system is the absence of any mechanical link between the steering wheel and the hydraulic circuit. An example of this type of steering system is shown in Figure 4. The architectures studied mainly differ in the way the flow rate delivered to the actuator is managed, including variable speed prime mover combined with on-off valves, variable-displacement pump systems and electrically actuated proportional valve control.

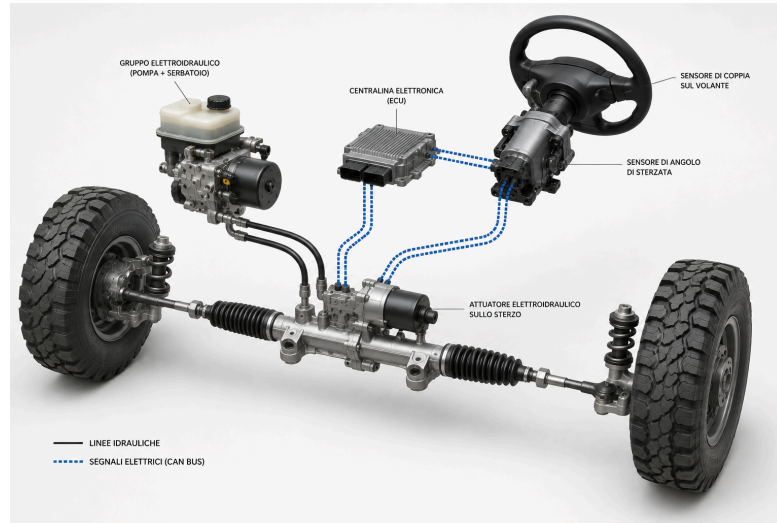


Figure 4: Steer-by-wire steering system

The analysis focuses on the monitoring of hydraulic quantities such as flow rate and pressure, allowing a preliminary evaluation of safety, redundancy and accuracy of the different electro-hydraulic steering systems. Moreover, a brief analysis of the energy efficiency of each solution is carried out, making it possible to compare the efficiency of the various architectures and underlining the main advantages and limitations.

Literature Review

A comprehensive review of electro-hydraulic steering systems can be organized into three main categories: pure steer-by-wire systems, hybrid electro-hydraulic systems with mechanical linkage, and advanced control strategies for electro-hydraulic steering systems. This classification allows for a clearer comparison of different technological approaches and highlights the evolution from fully decoupled architectures to hybrid and control-oriented solutions.

Steer-by-wire systems eliminate the mechanical connection between the steering wheel and the hydraulic actuation unit, relying entirely on electro-hydraulic components and control strategies. Within this category, the main differences concern flow generation, control architecture, and redundancy strategies.

The circuit proposed by Lee et al. [1] is composed of a fixed displacement gear pump driven by an ICE to supply constant flow to the hydraulic circuit. The system, shown in Figure 5, can manage both manual and automatic steering modes, but the two cannot be performed simultaneously. A normally open two-way directional valve is positioned in the supply line of the hydrostatic valve. To operate the automatic steering mode the directional valve is powered on, then the Electro-Hydraulic Proportional Valve (EHPV) can control the flow rate to the steering cylinder by moving the valve spool through a solenoid. The controller uses a proportional-feedforward architecture to evaluate the electric signal to be sent to the EHPV, thus reducing the influence of disturbances. In case of any failure, the manual steering mode can always be re-activated so that the redundancy of the system is ensured.

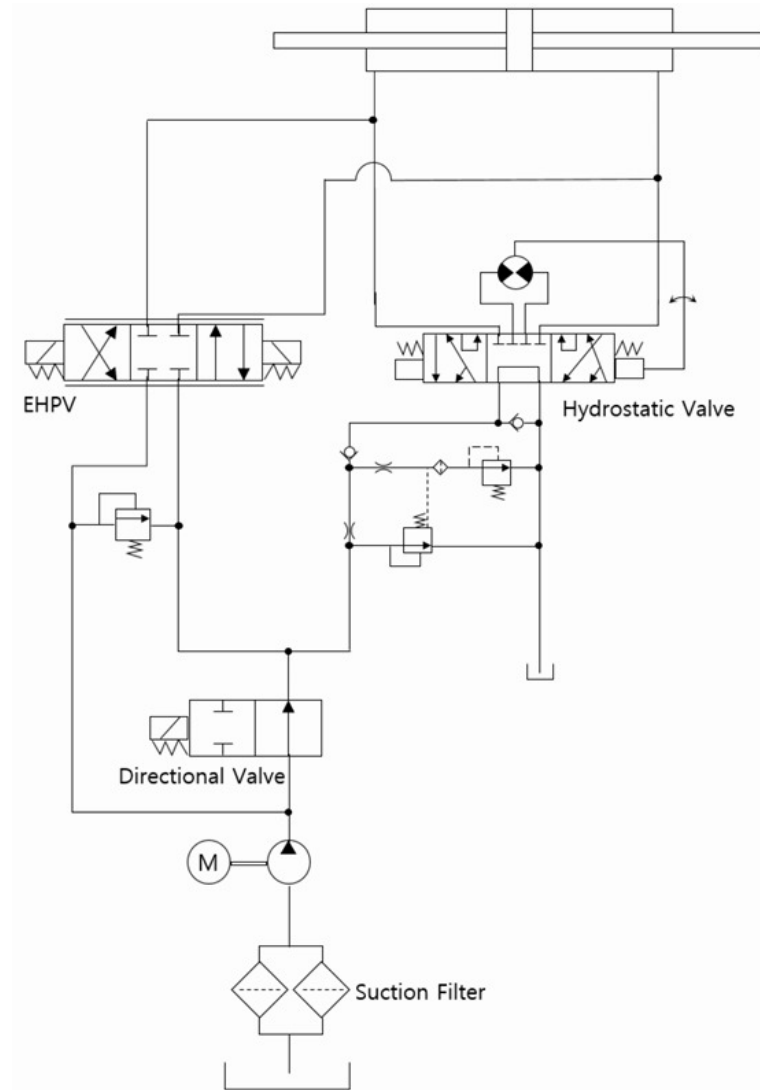


Figure 5: Schematic view of electro-hydraulic steering system taken by Lee et al.[1]

Safarzadeh et al. [2] introduced a more radical configuration, in which the steering wheel is completely removed and the driver interacts through two push-button-switch. The electric signal is transmitted to a solenoid valve by means of an electric circuit thus avoiding any mechanical link in the steering unit. The hydraulic scheme shown in Figure 6 is composed of a fixed displacement pump driven by a motor, and the flow to the actuator is decided by the solenoid valve. An auxiliary circuit is also present to make the steer possible in situations where the main is not working; there are two electrically connected shut-off valves used to select which

flow generation unit must feed the actuator. This auxiliary system is made of a fixed displacement pump, electric motor and a secondary reservoir. Moreover, for safety reasons both in the main and the auxiliary system a relief valve is present. Since the double-acting cylinder is asymmetric, another flow control valve is used to manage the flow rate to the smaller chamber; this way it is possible to have the same actuator velocity for both the ingoing and outgoing phases. In this study the aim was to reduce the effort and increase the comfort of the driver by deleting the steering wheel. The scheme represented below is not so qualitative and easy to understand, in fact the shut-off valves that are normally closed would not allow any passage of flow. The images are taken from the respective articles trying to simplify the comprehension of the systems explanation.

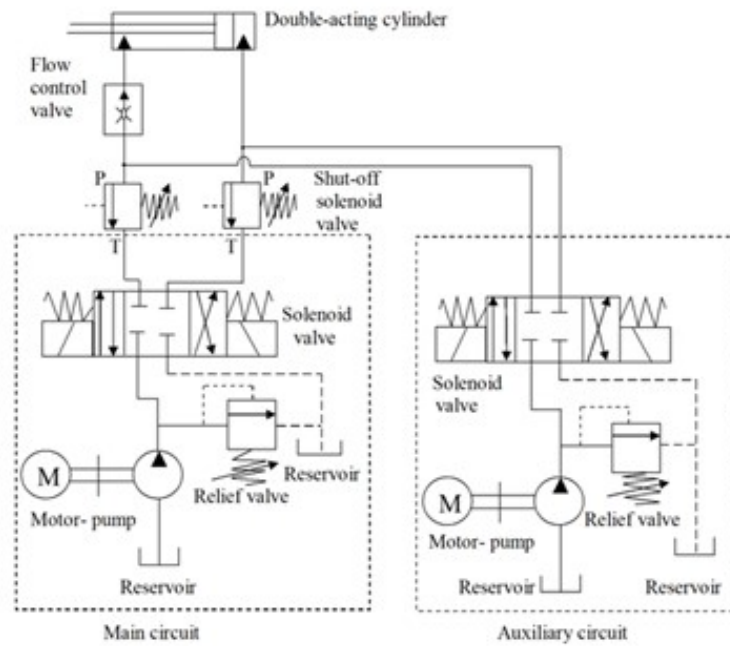


Figure 6: Hydraulic scheme for the steer-by-switch method taken by Safarzadeh et al.[2]

Industrial implementations, such as those developed by Danfoss [3], provide modular and scalable architectures. Among them, there is a version for pure steer-by-wire usage that is composed of a priority valve, a solenoid control valve (SOL) (2) to control the EH cut off valve (COV) (3), the EH directional valve (4), a pilot reduction valve (PVR) (9), a solenoid valve bridge (7) and relief (14), shock (18) and suction valve (17) for safety. The scheme of this type of steering system is shown in Figure 7. Before it is possible to steer the SOL (2) must receive power, this signal is operated by the PEVD CLS (5) through an input from a Man Machine

Interface (MMI). Once activated the SOL (2), the connection between the PVR (9) and one side of the COV (3) is established, the cut-off valve moves left and opens the passage from the EH directional valve (4) to the steering cylinder. When a steering signal is transmitted to the PVE (5) the solenoid valve bridge (7) is activated and the EH directional valve (4) is moved. The Load Sensing (LS) (15) line in the EH valve spool returns the maximum load pressure to the priority valve spool that maintains the requested delta pressure to its ends. In case the PVED CLS (5) registers any unintended movement of the EH directional valve (4), it will switch off the power to the SOL (2). In this way, the COV (3) will close the passage to the steering cylinder and EH steering will be deactivated.

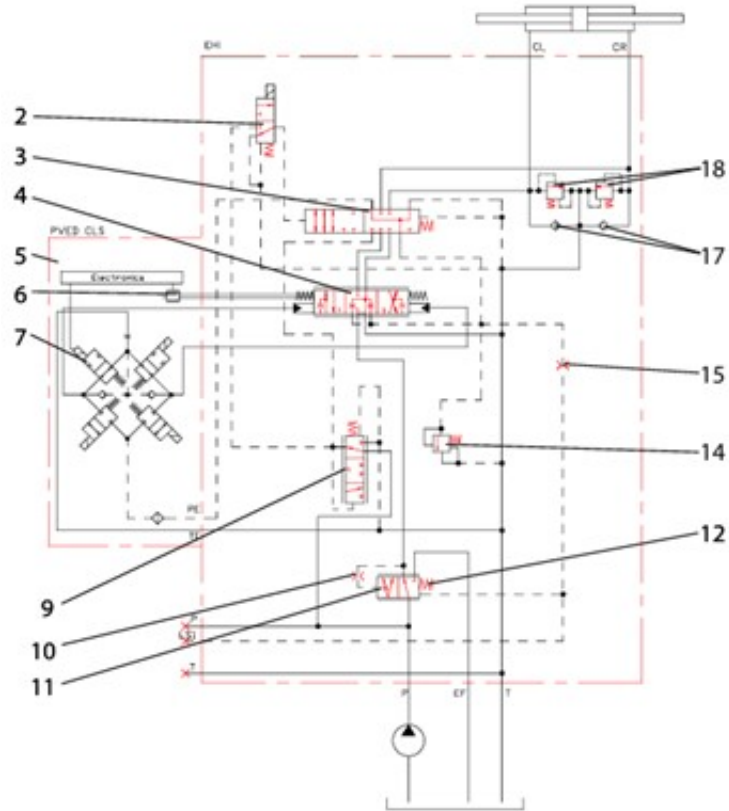


Figure 7: Hydraulic scheme of steer-by-wire system taken by Danfoss [3]

2.	SOL: Solenoid control valve for EH cut off valve	10.	PP damping orifice
3.	COV: EH cut off valve	11.	Priority valve spool
4.	EH directional valve	12.	Priority valve spring
5.	PVE control unit	14.	Pilot pressure relief valve
6.	Spool position sensor LVDT	15.	LS orifice
7.	Solenoid valve bridge	17.	Suction valve
9.	Pilot reduction valve, 13 bar, PE-T	18.	Shock valves

Table 1: Components

Other studies focus on increasing efficiency by acting on the flow generation unit and control logic. Singh et al.[4] proposed a steer-by-wire system in which the flow generation unit is composed of a fixed displacement pump rotated by a variable speed electric motor. As shown in Figure 8, there are four locking valves that are normally closed, these valves are arranged in pairs on the two sides of the pump and each couple allows the flow in both directions. When a steering command is given, the primary controller processes it and, through an electric circuit, opens two valves allowing only one flow path: from the pump to the high-pressure side of the actuators and from the low-pressure side of the cylinders back to the pump. In case of any failure a secondary controller will be activated, this controller will regulate a proportional steering valve with load sensing port which is fed by a separated hydrostatic pump through priority valve. The low-pressure side is connected to the existing hydraulic architecture to maintain minimum pressure and compensate for leakages. The aim of this study was to enhance the energy efficiency of the steering system by reducing hydraulic throttling losses.

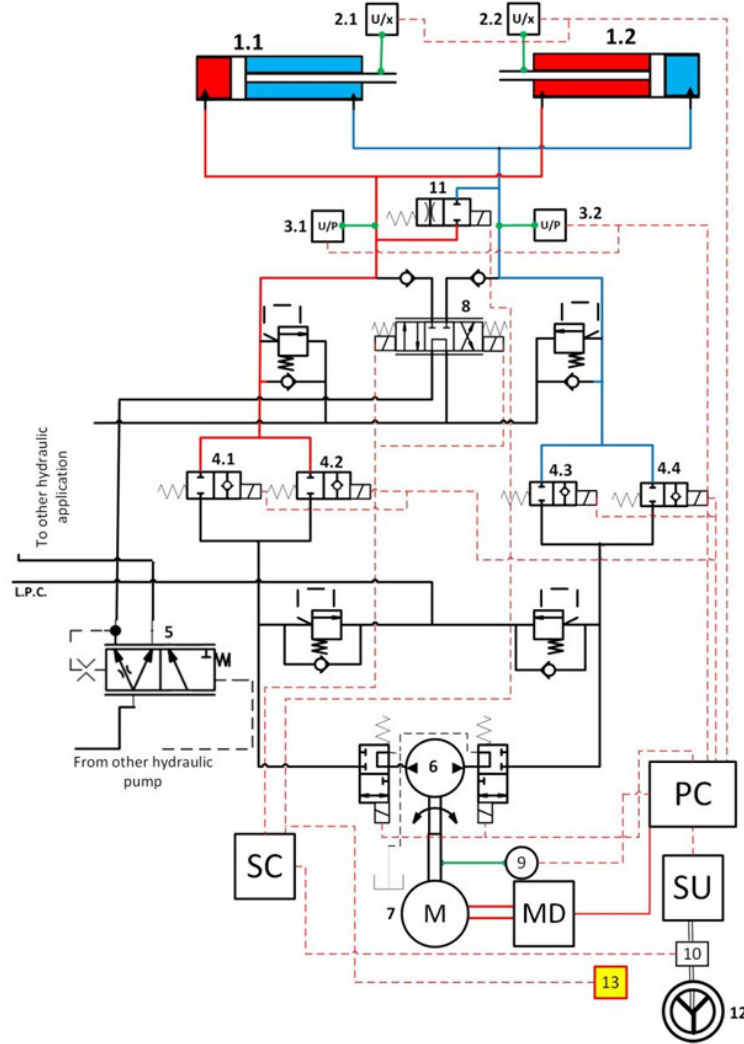


Figure 8: Scheme of electro-hydraulic articulated steer taken by Singh et al.[4]

The same authors further investigated a similar configuration, with slight variations in the control strategy and system layout. The steer-by-wire system proposed by Singh et al.[5] is composed of a fixed displacement pump driven by a variable speed electric motor and four normally closed control valves to manage the flow direction (clockwise or anticlockwise). When steering, the primary controller receives input from the steering unit and evaluates the signal to be sent to the solenoid control valves and to the motor. In case of any failure, a secondary controller will be activated and will control a conventional proportional steering valve with solenoid, making the system able to complete the steering maneuver. As it is possible to see

in Figure 9, the low-pressure side of the circuit is kept connected with the existing hydraulic architecture, thus a minimum pressure is ensured and leakage losses can be compensated. This steering system can achieve good results in terms of energy efficiency by reducing conventional hydraulic losses, maintaining high reliability and redundancy of the system.

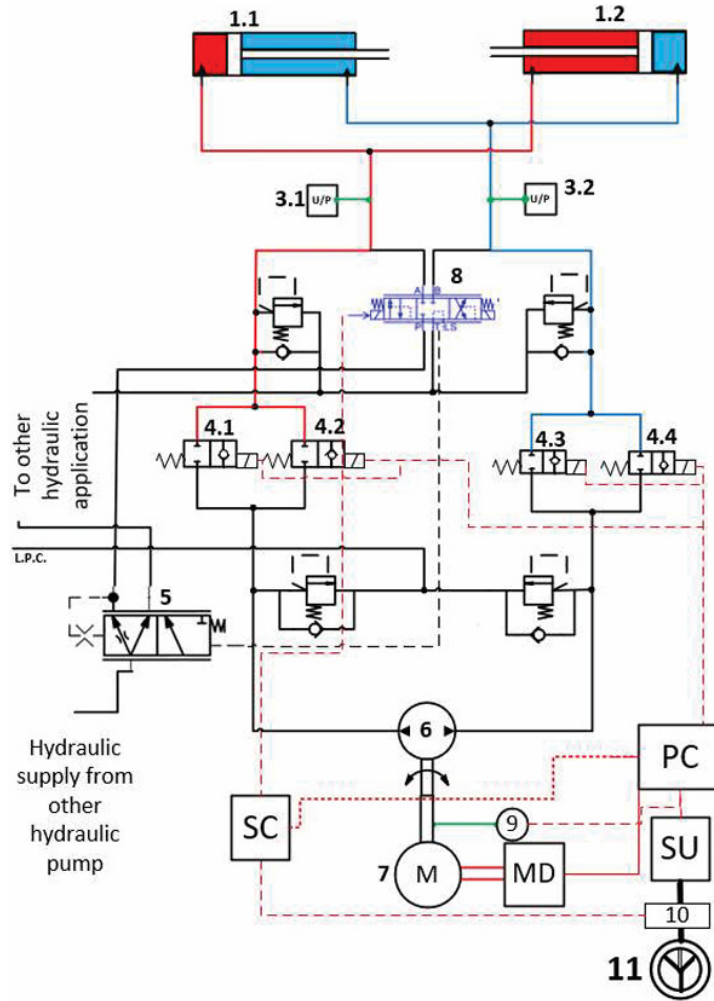


Figure 9: Hydraulic scheme for redundant electro-hydraulic articulated steering taken by Singh et al.[5]

An alternative strategy, explored by Daher et al.[6], relies on controlling the speed and/or displacement of a variable displacement pump (2). As is possible to see in Figure 10, the actuator (8) is asymmetric thus compensation is needed: pilot-operated check valves (6) are present to prevent unwanted piston movements and

block the load. This second flow generation unit is made of a fixed displacement pump (4) connected to the same prime mover (1) and provides continuous flow to the low pressure side of the circuit. The system is also equipped with pressure relief valves (7) to limit maximum pressure, for safety reasons. The control of the pump (speed and displacement) is made through an ECU: the controller acts on a double-rod actuator which is mechanically connected to the pump swash plate, indirectly managing the flow rate provided. Controlling pump displacement is possible to enhance the energy efficiency of the system, reducing hydraulic throttle losses.

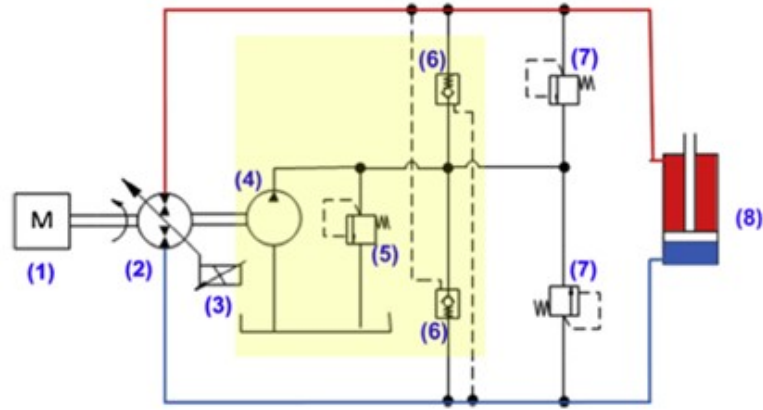


Figure 10: Displacement control steering hydraulic schematic taken by Daher et al.[6]

1.	Prime mover	5.	Pressure relief valve
2.	Variable displacement pump	6.	Pilot-operated check valves
3.	Pump control system	7.	Pressure relief valves
4.	Fixed displacement pump	8.	Actuator

Table 2: Components

Further developments exploit load-sensing principles to regulate pump operation. Haggag et al.[7] proposed a steer-by-wire system in which the flow generation unit is composed of a variable displacement axial piston pump controlled by load sensing line and moved by the ICE. As it is possible to see in Figure 11, there are two actuators used to control pump displacement: in the smaller one acts the pump pressure and in the area of the larger one acts the regulated pressure. Typically, the swash plate is pre-loaded with a spring in order to have the maximum displacement of the pump as default, the regulation of the swash plate is made so that the

pressure difference between the load and the pump output is kept constant. A two-stage proportional electro-hydraulic valve called Hydrac valve is used to control the passage area to the actuator lines, the passage area along with the pressure drop across the valve ends determine the flow rate to the actuator. The valve receives current as input from the controller, thus varying this signal the position of the valve is regulated. The inputs of the steering control algorithm are the steering wheel angular velocity, the vehicle speed and the articulation angle between front and rear frame in case the system is implemented on an articulated vehicle.

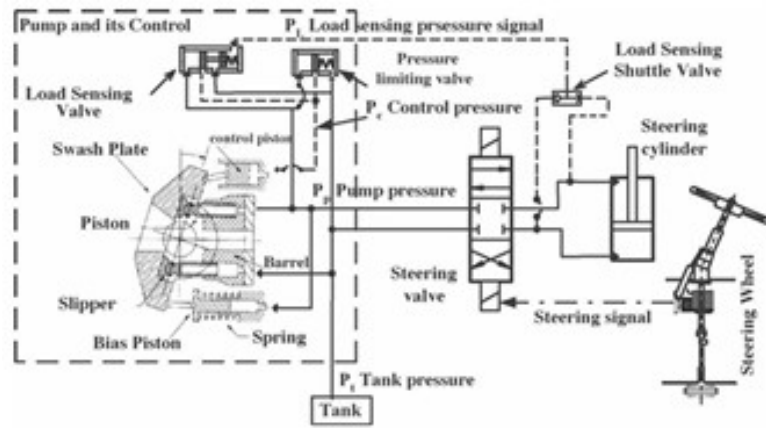


Figure 11: Variable-displacement pump operating principle and load-sensing control circuit taken by Haggag et al.[7]

From single-axis systems, research has also expanded to multi-axle configurations. The electro-hydraulic system provided by Zhang et al.[8] is used in multi-axle vehicle and is considered as a five-bar mechanism. The Two DOF Electro-Hydraulic Servo Steering System (TDEHSSS) system has two servo solenoid valves: the one on the right is used to regulate the flow rate to the steering cylinder, the other is used to feed the tie rod which has variable length in this solution. In Figure 12 is shown the hydraulic circuit that is not complete, for sake of simplicity two assumptions are made: the flow generation unit is a constant pressure source and the connection lines are short and thick, preventing from high pressure drop. The system has two DOF: the left steering angle α and the right steering angle β . The aim of this study is to find a solution that makes possible the Ackermann steering and to reduce the abnormal wear of the tire.

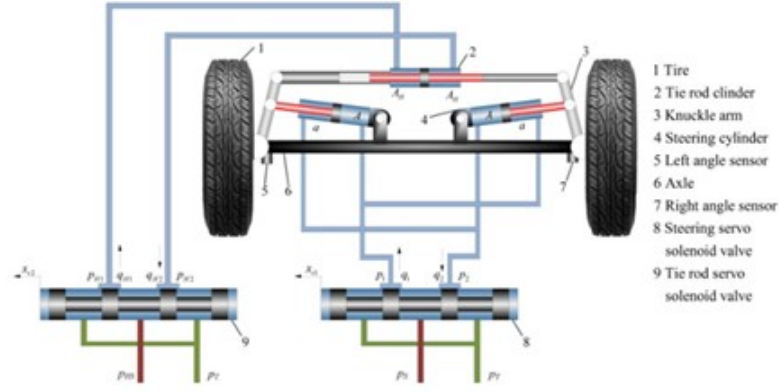


Figure 12: Structural diagram of heavy vehicle TDEHSSS taken by Zhang et al.[8]

Similarly, Du et al.[9] implemented an Electro-Hydraulic Servo Steering System (EHSSS) in a multi-axles vehicle to improve maneuverability and handling. Typically, such system is mounted only in the rear axles, in the front ones is kept the mechanical-hydraulic steering system thus ensuring vehicle safety. The hydraulic scheme represented in Figure 13 is composed of a servo solenoid valve (1) with a 4-way directional valve with solenoid in parallel (2), two pilot operated check valves (4-5) that connect the servo valve with the two actuators and two relief valves (6-7) for safety reasons, for sake of simplicity the flow generation unit is considered ideal. The two pilot check valves receive the pilot flow through a 3-way directional poppet valve with solenoid (3), during electro-hydraulic servo steering maneuver this valve is in the normal position thus allowing bidirectional operation of the check valves. If there is necessity to lock the wheels in a certain position in steering mode, the poppet valve is energized thus closing the check valves that block the oil from the cylinders. In case of any malfunctioning of the servo valve (1), the directional valve (2) is used to move the wheels by manual operation thus ensuring system reliability.

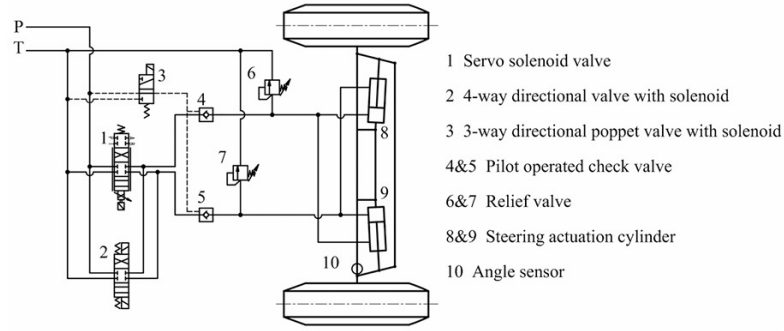


Figure 13: Electro-hydraulic scheme of servo steering system for a single axle taken by Du et al.[9]

Building on this, stability and dynamic response improvements are addressed in the following work. The system proposed by Du et al.[10] is a variable speed pump controlled steering unit with the addition of the back pressure control (BCVSPCSS) to enhance the system stability. This solution is implemented in multi-axle vehicles and is composed of a servo motor connected to the pump, a directional valve with solenoid, two actuators and a servo proportional valve with solenoid, as is possible to see in Figure 14. During steering operation, the controller sets the speed of the motor and the opening of directional and proportional valves according to the control signals. The solenoid directional valve is used for controlling the direction of steering, instead the servo proportional valve function is to control the back pressure of the circuit low pressure side by controlling the valve opening. The controllable speed pump is implemented in order to reduce the throttling losses: at each time the controller imposes the motor speed such that the pump delivers only the amount of flow that is required by the steering maneuver. The system has great advantages in terms of energy efficiency, and the dynamic characteristic is improved thanks to the back pressure control.

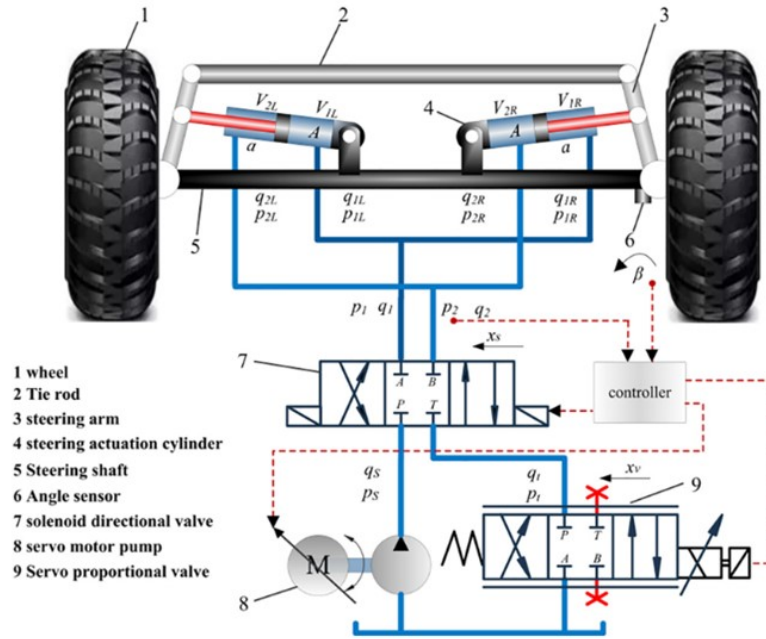


Figure 14: Basic principle diagram of BCVSPCSS taken by Du et al.[10]

While the previous systems completely remove the mechanical connection, the following solutions retain it while integrating electro-hydraulic functionalities to improve performance and reduce driver effort.

The solution suggested by Wang et al.[11] is a dual-mode steering system: it is present the classic steering wheel and a second controller is implemented in order to reduce the effort of the driver during sequential operations. To reach this purpose, a pilot control valve with a joystick, solenoid directional valves, and a controller are added to the traditional SWC steering system, as is possible to see in Figure 15. The flow generation unit is composed of a variable displacement pump connected to the prime mover, and feed both the flow amplifier valve and the pilot lines. Both the pilot lines coming from the pilot control valves (joystick control) and the redirector (steering wheel control) are used to manage the flow amplifier spool displacement, the oil supply valve is the one in charge to feed both circuits. The two systems cannot operate simultaneously thus two solenoid directional valves are used for the control mode conversion. Both the solenoid directional valves and the pilot valves are electrically connected with the controller, while the redirector is mechanically linked with the steering wheel. The aim of this study was to improve the maneuverability of the wheel loader by implementing a dual-mode steering system.

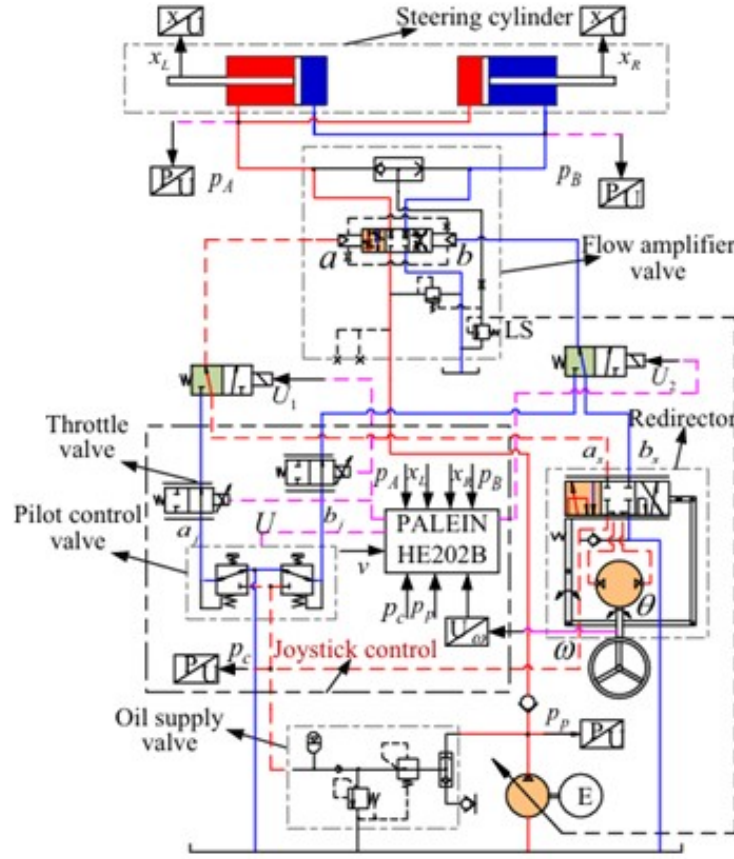


Figure 15: Schematic diagram of the dual-mode steering system taken by Wang et al.[11]

To further improve efficiency, Wang et al.[12] proposed an Electro-Hydraulic Flow Matching (EHFM) system composed of a servo motor that drive a fixed displacement hydraulic pump, which is used to supply the demanded oil flow for the steering operations. As is possible to see in Figure 16, the EHFM is connected in parallel to the original load sensing steering system through a solenoid valve. The controller receives as input the pressure of the steering cylinder, the steering wheel velocity and the angle and speed of the servo motor; instead, as output gives two control signals: the target speed of the servo motor and the on/off signal for the solenoid valve. When the EHFM feed the controlled flow to the redirector the original flow generation unit can feed the auxiliary actuators through a priority valve. In case the controller senses any failure during steering operation it immediately sent a signal to the solenoid valve thus the LS steering system can complete the maneuver, thus improving system reliability.

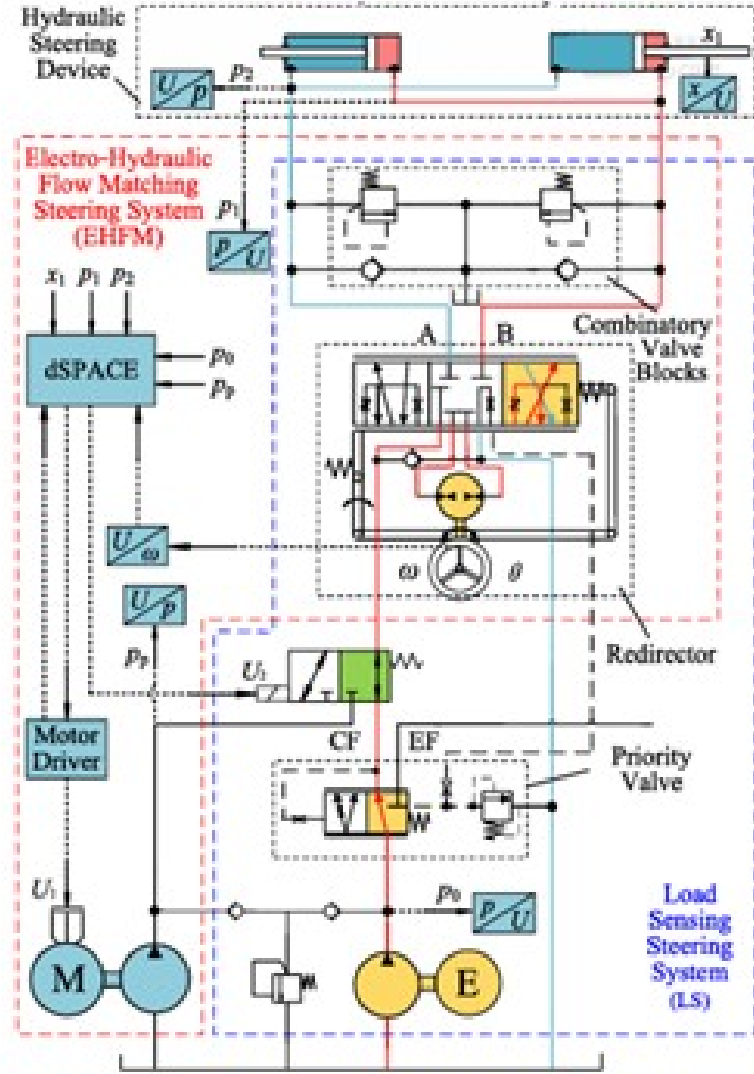


Figure 16: Schematic diagram of the steering system taken by Wang et al.[12]

Another variation maintaining mechanical linkage is investigated by Singh et al.[13], in this system the mechanical connection between the steering wheel and the hydraulic circuit is present: a modified orbital steering valve is connected through a mechanical shaft to the steering wheel. In Figure 17 is represented the test bench of the proposed hydraulic system, where the flow generating unit is composed of a fixed displacement hydrostatic pump moved by an electric motor. The speed of the prime mover is regulated by the controller through steering angle and speed data, collected by SASA sensor; a minimum speed of the electric motor is needed to avoid cavitation problems. The OSPD valve has two gearsets connected through

a shift valve of on/off configuration, this valve is normally closed by a spring and a minimum of 5 bar in the pump line is needed to overcome the spring force. During normal operation of the system the pressure is higher than 5 bar and thus the two gearsets are connected. The peculiarity of this modified orbital valve is a shortcut present at the ends of the secondary gearset that reduce the resistance of the oil path; this allows to lower the effort of the flow generation unit increasing the energy efficiency. In case of any failure the flow generation unit will be disconnected by the safety valve, the pressure in the feeding line will drop to 0 bar closing the shift valve. Thus the operator will have to move only the primary gearset involving a lower torque for the hand pumping. This study had the objective of increasing the energy efficiency of the system by generating only the required hydraulic flow.

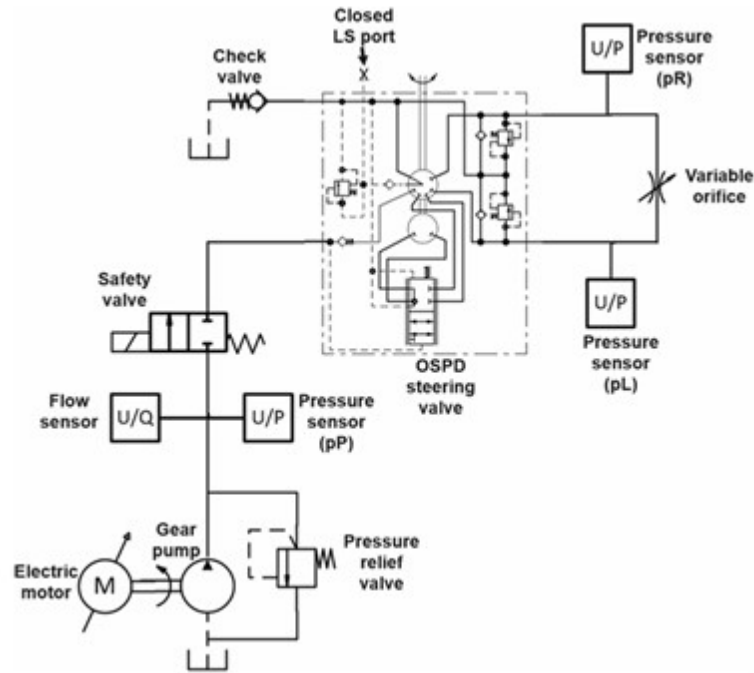


Figure 17: Hydraulic system schematic of steering test bench taken by Singh et al.[13]

A different hybrid approach combining electric and hydraulic assistance is proposed by Zhao et al.[14]. The system represented in Figure 18, is made of two power-assisting units: one acts directly on the steering column, and the other is the hydraulic circuit. On the base of the signal received by the steering wheel the Electro-Hydraulic Compound Steering (EHCS) system decides the electro-hydraulic assist power ratio: at low speed they work together, the hydraulic unit provides large steering assist power and the electric unit can reduce energy consumption. As

the vehicle speed increases this ratio is regulated to achieve good road feeling for the driver: both the electric and hydraulic unit work with smaller power output, in these conditions also the energy consumed by the system is lowered. In case of any failure the two systems can work separately thus ensuring the steering ability of the vehicle. The mechanical link between pinion and rack is present, thus it is not a steer-by-wire system; however, the system achieves good results in terms of steering performance and energy consumption.

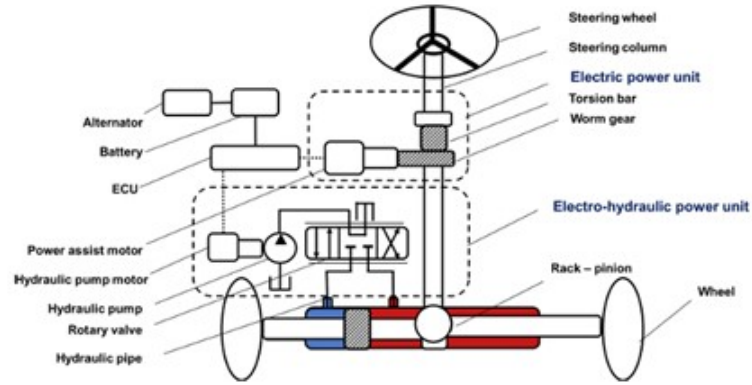


Figure 18: Electro-hydraulic compound steering system taken by Zhao et al.[14]

Finally, the Electro-Hydraulic Power-Steering (EHPS) system developed by Kemmetmuller et al.[15] try to increase the energy efficiency introducing a new configuration of the power supply. The hydraulic circuit shown in Figure 19 is composed of a fixed displacement pump, driven by the ICE, which charge a hydraulic accumulator; two closed-center electromagnetic steering valves are used to control the flow rate to the actuator. The ECU receives the pressure signal (ps) from the accumulator and if it is under a certain minimum level the pump is activated, recharging the accumulator up to a maximum pressure. Then the pump is disconnected through the activation of the electronically controlled bypass valve. During steering operation, the ECU receives the pressure signals (pa-pb) from the two chambers of the assistance cylinder and through a feedforward closed loop regulate the openings of the steering valves. To these information are added also the input steering torque and the steering wheel angle speed, from a sensor mounted on the steering column, to evaluate the ratio of assist torque to give to the driver. The advantages of this type of power supply are that the energy efficiency is increased thanks to the bypass valve, the pump can be designed smaller while the peaks in volume flows demand can be fulfilled by the hydraulic accumulator.

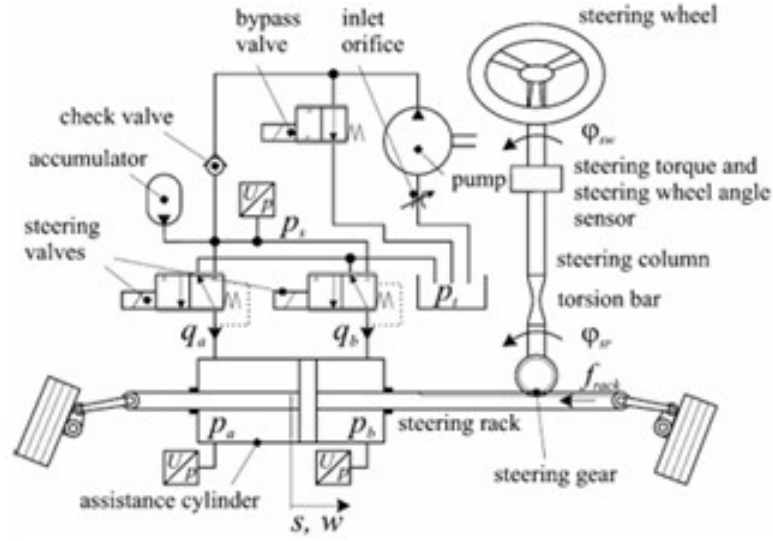


Figure 19: Schematic of electro-hydraulic steering system taken by Kemmetmuller et al.[15]

Beyond hardware configurations, several studies focus on advanced control strategies to handle system nonlinearities, improve steering feel, and optimize energy consumption.

The Electronically Controlled Hydraulic Power Steering (ECHPS) system represented in Figure 20 has the purpose to realize an active control of steering system and to enhance the driver's road feeling. ECHPS is based on servo proportional electromagnetic valve so the controller can control the opening of the servo proportional valve in order to reach the pressure target, computed through given curves of power characteristics. To do that Yuanzhe et al.[16] used a fuzzy adaptive Proportional-Integrative-Derivative (PID) controller, its inputs are the difference between the target and measured pressure and the rate of change of that error. The fuzzy controller receives also vehicle's dynamics data, so to adapt the characteristics of the steering wheel to the vehicle speed thus improving the driver's steering feel. Moreover, a returnable control strategy based on the angle of the steering wheel can be implemented, with a Proportional-Derivative (PD) controller, to make the steering wheel return to the middle position at the end of the steering maneuver.

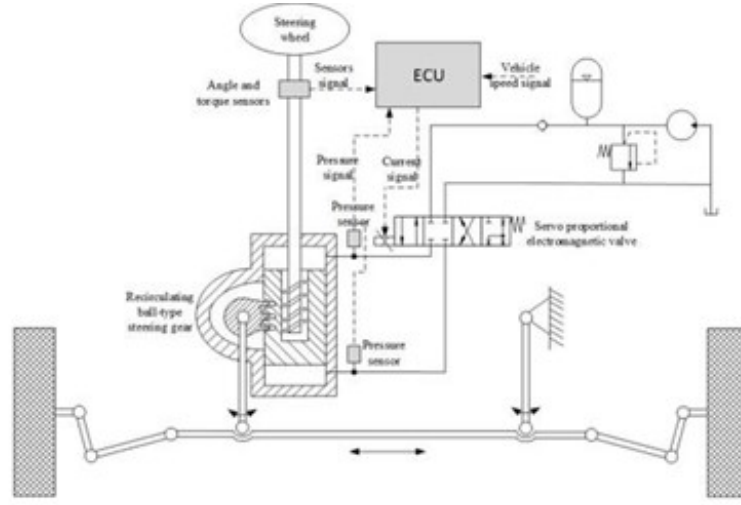


Figure 20: Structure of ECHPS system taken by Yuanzhe et al.[16]

To address more complex system Cheng et al.[17] proposed a self-tuning neural network generalized predictive controller. The Electro-Hydraulic Composite Steering (EHCS) system shown in Figure 21 is composed of two different actuators: an Electric Power Steering (EPS) system and an Electro-Hydraulic Power Steering (EHPS). To achieve effective track assistance, reduce the fluctuations of the steering torque and reduce the energy used by the system the self-tuning neural network approach is used. The aim of reducing the energy usage is taken over a double-swarm division and cooperation fruit fly optimization algorithm (DDCFOA) by optimizing the assist torque ratio between the EPS and EHPS. Instead, a neural network implicit stair-like generalized predictive control (NNISGPC) is used to manage the nonlinearity and hysteresis of the two systems, thus reducing the assist torque fluctuations. To further improve NNISGPC performance, DDCFOA can be used to make the neural network parameters self-tuning. This results in a increase of the responsiveness and precision of the system.

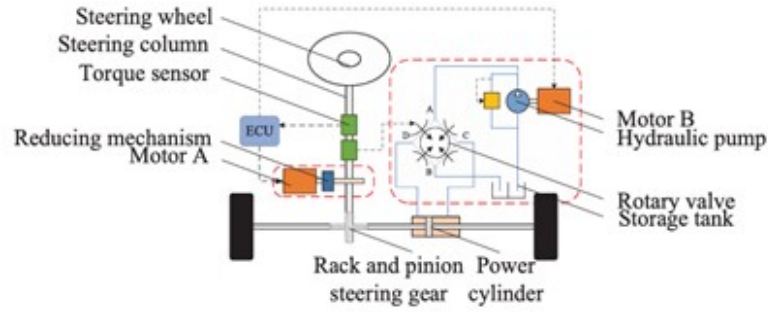


Figure 21: Structure of ECHS system taken by Cheng et al.[17]

Similarly, Guo et al.[18] focused on optimizing torque distribution using intelligent control techniques. The control of new type of Electro-Hydraulic compound steering (EHCS) system is challenging due to the non-linearity of dual actuators. The main objective of the control strategy proposed in this work is to optimize the assist torque distribution between the electric and hydraulic systems. Combining Genetic Algorithm (GA) and Elman Neural Network Predictive Controller (NNPC) allows to operate in the high-efficiency area both the motors that drive the two systems. The controller inputs are the driver steering torque and vehicle speed, and it gives back two control voltages for the two motors. This way part of the losses of the steering system are avoided, improving system efficiency. The structure of the EHCS system is schematized in Figure 22.

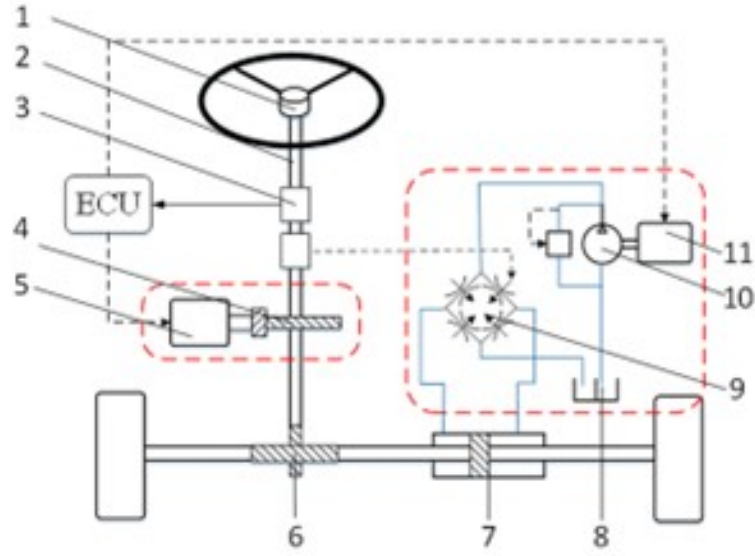


Figure 22: Structure of electro-hydraulic compound steering system taken by Guo et al.[18]

In addition to neural-based methods, classical control approaches are still widely investigated and compared. The steering system of an articulated steered vehicle can integrate different valve solutions like the Proportional Directional Control Valve (PDCV) and the Stepper Motor-Driven Orbitrol Valve (SMDOV). In the SMDOV both the orbitrol valve and the stepper motor are mechanically connected to the steering wheel, and the regulation of flow is made by feeding the motor with PWM signals at different frequencies. Instead, the PDCV can control the variation of the flow depending on the voltage received for solenoid actuation. As shown in Figure 23, the Proportional-Integral-Derivative (PID) controller receives as input the position error, evaluated as the difference between the commanded and the actual cylinder position, and by regulating the steering valve's flow the controller can minimize the position error. Using PDCV results in more accurate position tracking and higher steering response in comparison to the orbitrol valve. In terms of required energy, the study of Rowduru et al.[19] points out that the SMDOV is able to save approximately 50% of the energy consumed by the PDCV, thus improving energy efficiency of the system.

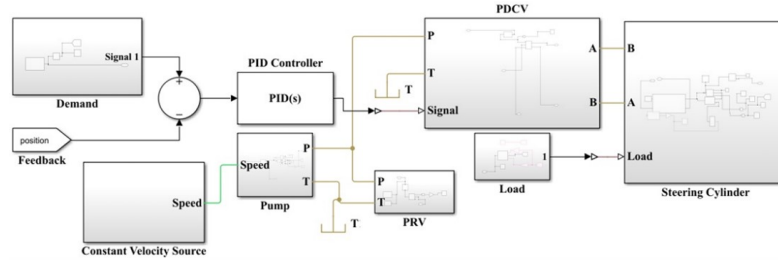


Figure 23: Position control of the PDCV/operated in articulated steering system taken by Rowduru et al.[19]

Finally, high-performance servo-valve-based system requires precise pressure control strategies. The steering system represented in Figure 24 is composed of electronically controlled high-performance servo valves with closed center, that can individually control the pressure of each chamber of the assistance cylinder. The supply to those valves is a constant pressure system and the controller is based on the driver steering torque measurement, thus avoiding mechanical connection with the torsion bar. Dell'Amico et al.[20] proposed a Proportional-Integral (PI) controller with second-order dynamics to manage valve openings and to have more reliable pressure measurements. The input to the controller is the error between the reference and measured pressure, but due to system's non-linearity the controller should be adaptive. In this type of system, an additional lag is present due to the absence of any mechanical connection with the torsion bar, this might result in negative impact on the steering feel. In order to overcome this issue, a pre-filter can be added to the control loop to control the response of the pressure and using fast valves enhances steering performance. In the figure is shown also the solution of an open-center system to make a comparison between the two.

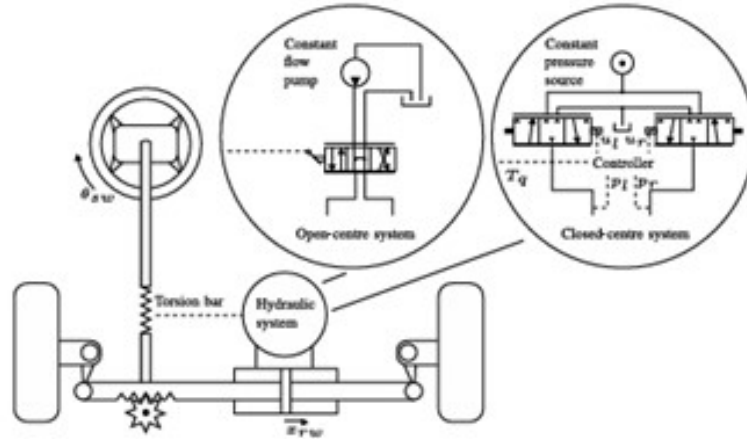


Figure 24: Schematic of steering system showing both the open/center and closed/center hydraulic systems taken by Dell'Amico et al.[20]

Capitolo 1

Autonomous steer using electro-hydraulic proportional valve

1.1 Introduction

The first system is illustrated in Figure 1.1, it can operate in both autonomous and manual steering mode by switching from an Electro-Hydraulic Proportional Valve (EHPV) to a servo hydraulic proportional valve for flow control to the steering cylinder. This configuration is the basic idea beyond the work developed by Lee et al.[1], the hydraulic scheme represented in Figure 1.2 can be used to understand how it works. The analyzed system is built from a hydraulic power steering system with a Danfoss OSPC Load Sensing valve mounted. This type of steering system can be used in off-road applications where the vehicle speed does not reach a high value, i.e. lower than 40/50 km/h.

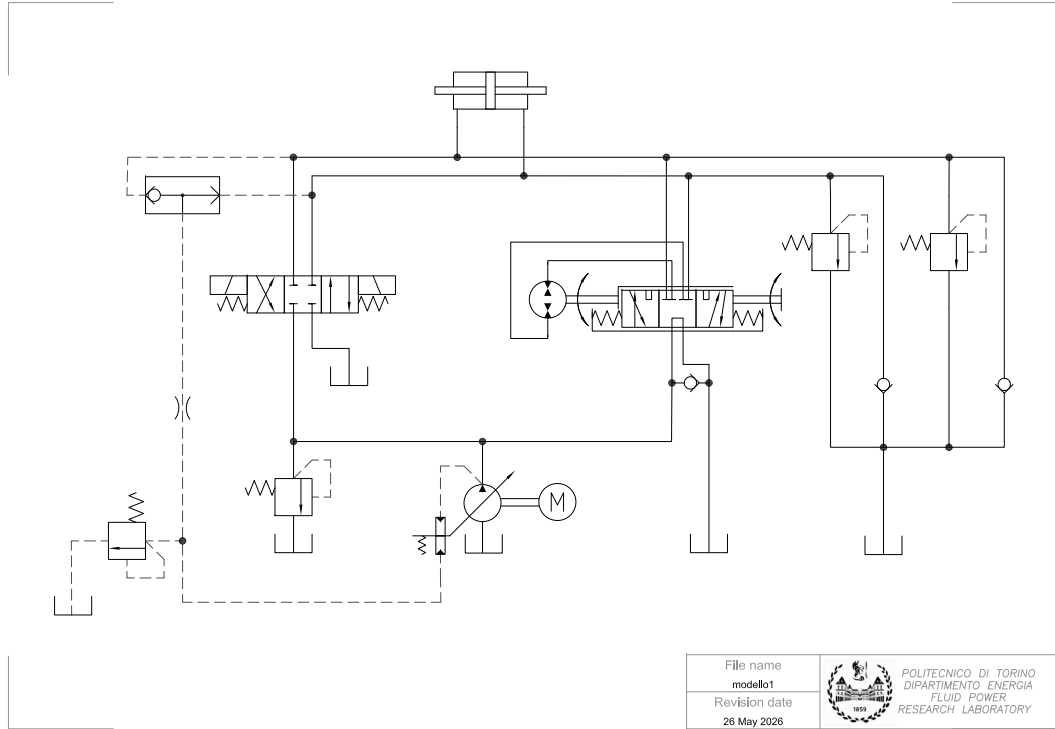


Figure 1.2: Hydraulic scheme of the first model

1.2 OSPD valve

The OSPC valve is composed of a continuous position rotary distributor modeled as a three-position six-port valve. As can be seen in the Scheme 5, in addition to the four connections with the power supply and actuator, there are two extra ports present to connect the valve to an orbit motor. This volume meter is mechanically connected with the sleeve, thus while steering the sleeve follows the movement of the spool trying to return to the equilibrium conditions. Moreover, the orbit motor is connected in series with the actuator, so it is possible to assert that the steering cylinder velocity is proportional to the speed of the volume meter.

When the driver starts the steering maneuver, a torque must be provided to the steering wheel to overcome the force of the spring that keeps the valve at rest. This torque is then transmitted to the spool, giving origin to a relative rotation between the sleeve and spool. Therefore, a certain flow is allowed to pass from the power supply through the valve and orbit motor to the actuator, making the wheel turn. Once the driver stops to rotate the steering wheel, the sleeve manages to reach the spool position, thus closing the oil passages to the orbit motor and the actuator.

This valve configuration is called closed-center non-reactive: in neutral position the ports of the valve are closed, and the steering wheel will not sense any external torque applied to the wheels.

The load sensing feature, shown in Figure 1.3, is implemented by adding a pilot line that returns the maximum pressure of the system to the load sensing controller of the variable displacement pump. In this way, the pump is always kept at a slightly higher level of pressure with respect to the load. Considering a more general hydraulic scheme, where the power supply must also feed other actuators, a priority valve can be added, and the pilot load pressure of the steering actuator should be used in feedback to give priority to the steering cylinder. This way, during normal operation of the system, the flow rate requested by the steering actuator can always be satisfied. In that case, the pressure used for the control of the variable displacement pump is the maximum between the one from the steering cylinder and the one from the auxiliaries.

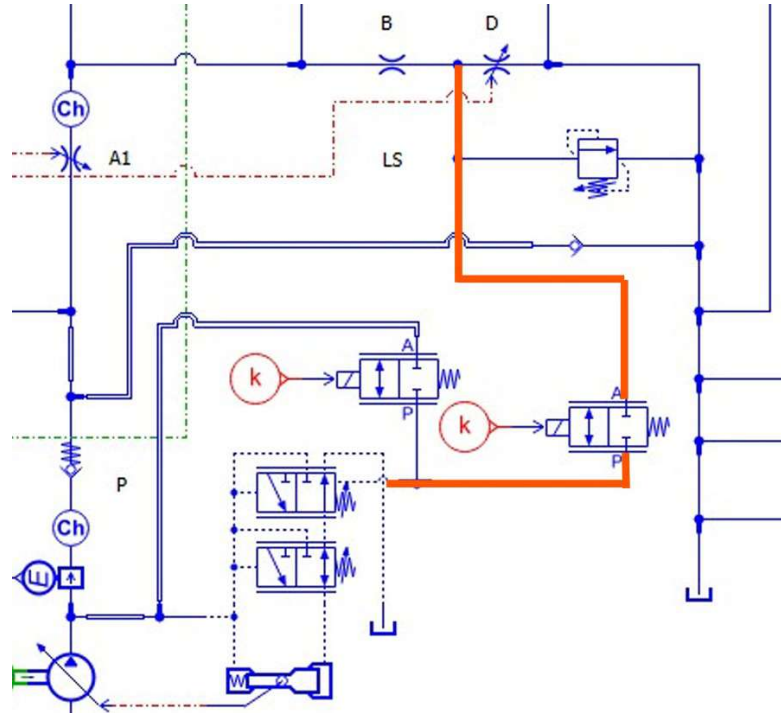


Figure 1.3: Load Sensing circuit

The Amesim model of this hydraulic circuit, represented in Figure 1.4 was developed by Shalabi [21] in his thesis project. The OSPC valve is modeled by multiple variable orifice valves, where the functions of the opening areas are designed based on the geometry and relative positions between the spool and sleeve.

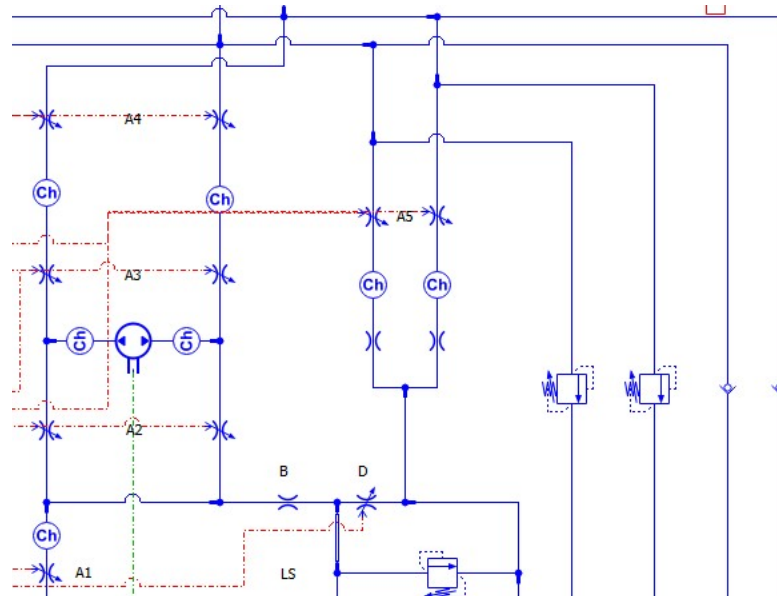


Figure 1.4: OSPD valve

1.3 Adjustments on actuator, pump and valve openings

The design of the actuator is made considering the typical size of steering cylinders of articulated wheel loader: in such a vehicle the steering is done with two asymmetric double acting cylinders, placed one per side of the articulation joint, Figure 1.5. For this type of vehicle, the bore can range from 70 mm, for compact loaders, to 90-110 mm for large loaders; the rod dimension can range from 45 mm to 60 mm; the stroke can vary from 380 mm to 525 mm.



Figure 1.5: Double acting linear actuator symmetric

For this case study it is considered a large wheel loader, 18-25 t class, and for sake of simplicity the steering actuator is modeled as a single symmetric double acting cylinder. To do that the biggest dimensions for both the bore, rod and stroke are taken, and through the Formulas 1.4 the equivalent symmetric actuator is designed. Using this solution, both the actuator velocity and forces are comparable to the ones of the original steering actuator system. The final sizes of the actuator are reported in Table 1.1.

$$A_p = \frac{\pi D^2}{4}, \quad A_a = \frac{\pi(D^2 - d^2)}{4} \quad (1.1)$$

$$A_s = A_p + A_a = \frac{\pi}{4}(2D^2 - d^2) \quad (1.2)$$

$$D_s = \sqrt{d^2 + \frac{4A_s}{\pi}} \quad (1.3)$$

$$(1.4)$$

Bore	Rod	Stroke
100 mm	50 mm	500 mm

Table 1.1: Steering cylinder dimensions

Considering that the steering system is designed to move the actuator from one end-stop to the opposite in 6 complete turns of the steering wheel, the chosen

maneuver has a maximum steering angle of 720° , thus requesting a piston shift from the middle position of approximately 65.5% in both directions. Increasing the dimension of the actuator the flow rate request for the maneuver changes, thus the pump characteristics should be recomputed. Taking into account the aforementioned dimensions of the piston, and imposing a maximum time of 5 seconds to complete the 6 steering wheel rotations, it is possible to compute the highest flow rate demanded by the actuator:

$$V = \frac{S}{t} = \frac{0.5}{5} = 0.1 \text{ m/s} \quad (1.5)$$

$$A = \frac{\pi D^2}{4} = 7.854 \cdot 10^{-3} \text{ m}^2 \quad (1.6)$$

$$Q_{in} = V \cdot A \cdot 60 \cdot 1000 = 47.1 \text{ L/min} \quad (1.7)$$

$$(1.8)$$

Based on the results obtained, the pump size can be set at 30 cc/rev, thus guaranteeing that while the maximum flow is requested by the actuator the pump is able to feed enough flow rate with relatively low motor speed, lower than 2000 rpm.

Knowing that the average flow rate to the actuator is increased, the opening areas of the OSPD valve should be revised. In order to compensate for the higher flow demanded, the areas of the valve are enlarged by a factor of 1.54, that is the ratio between the old and the new actuator stroke. In addition, the orbit motor should also be enlarged for the same reasons as the opening areas of the valve; it is set at 500 cc/rev, so the steering system can feed the actuator even in case of system failure.

As regards the safety system, the design developed by Shalabi[21] is kept unchanged, thus relief and check valves are connected in parallel both in the supply line and at the exit of the orbit valve. The maximum cracking pressure is raised at 150 bar due to the higher maximum pressure that this kind of steering unit can reach. Furthermore, the maximum controlled pressure of the differential pressure limiter is increased at 200 bar and the differential pressure control is set at 20 bar. This solution leads the power supply to reach higher maximum pressure, making possible the functioning of the electro-hydraulic steering system.

1.4 Electro-Hydraulic Proportional Valve

In parallel to the OSPD valve is added a EHPV so to exploit the electro-hydraulic actuation, as shown in Figure 5. The basic idea is taken from the study of Changjoo Lee et al.[1] where the orbit valve has the main function of safety and redundancy

and it is not possible to have the two systems in combination steering. In order to implement the autonomous driving, a controller is added in closed loop with the steering cylinder, making possible to evaluate the openings of the proportional valve based on the position of the actuator and the steering input.

The electrically actuated proportional valve designed is a three position four port valve with closed center. The input to the EHPV is an electric signal computed by a controller, and to fully open the passage area of the valve is needed a current of 40 mA. For sake of simplicity the pressure drop characteristics of the valve are considered almost ideal: at the maximum opening area of the ports the corresponding pressure drop is set at 0.1 bar for all the four paths. Moreover, the flow rate at maximum opening is not the same for all ports, as shown in Table 2 the characteristic flow rate into the steering cylinder is lower than the one going towards the tank. This way the back pressure is lowered, trying to avoid too small pressure that can make the system unstable in case of driving load and preventing cavitation phenomenon.

P to A	B to T	P to B	A to T
5 L/min	10 L/min	5 L/min	10 L/min

Table 1.2: Maximum flow rate in each port

The input to that electro valve is an electric signal computed through a Proportional controller (P). Such controller receives as input the difference between the steering signal and the position of the actuator. The signal of steering is an electric input that represent the desired steering angle, can be set through an external computer and sent to the steering system. Then, this input is compared with the position signal acquired at the steering cylinder level. A position sensor is implemented in the scheme to evaluate the exact piston displacement, then this signal is sent to a CPU that elaborates the data to make it comparable with the steering input. In this case study the CPU work is done by a function block where the position signal is converted in angle through the following Formula 1.4:

$$\theta = \frac{720 \cdot x}{0.164} \quad (1.9)$$

The error evaluated between the two angles is the input to the P controller, that returns the entering signal to the EHPV. The P controller has only the proportional gain to be evaluated, thus a trial-and-error technique is enough for the computation of the parameter. In order to choose the amplitude of the gain both the accuracy and the velocity of the response of the system to the steering input are considered. At the end of iteration process the outcome value of the controller parameter is

10, with this value the system is able to respond fast enough to the input and the output is close to the required maneuver. This type of control system, shown in Figure 1.6, belongs to the closed loop control system, and has the peculiarity of automatically regulate its output.

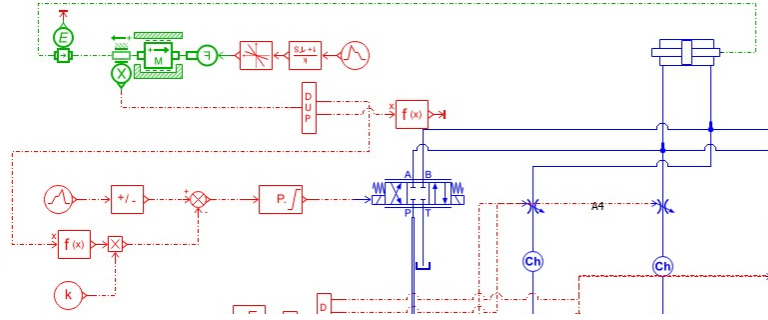


Figure 1.6: Controller

The result obtained is due to the use of the position signal of the steering actuator in feedback to compare the output with the desired set point. This way, the system can make adjustments to maintain proper performance without the intervention of any human being. This feature is at the base of the autonomous driving system, in fact the electro-hydraulic steering system designed does not receive the input from the mechanical rotation of the steering wheel, but an operator should be present in case of any failure. Actually, the presence of the OSPD valve ensures the reliability of the system thus an operator is needed when the control on the maneuver is switched to the manual steering mode. When this happens, the electrically actuated steering valve automatically returns in neutral position, closing the oil flow passage through the valve.

1.5 Simulation setup

The simulation of the steering system requests an input signal that defines the desired maneuver. For this case study the steering maneuver consists of a first turn to the right till the steering angle reaches 720° and this configuration is kept for 2 seconds. Then, as is possible to see in Figure 1.7, the desired angle pass from 720° to -720° in almost 4 seconds and the turn to the left is kept at the same angle for 2 more seconds. At the end, the vehicle returns with straight wheels and the maneuver is complete.

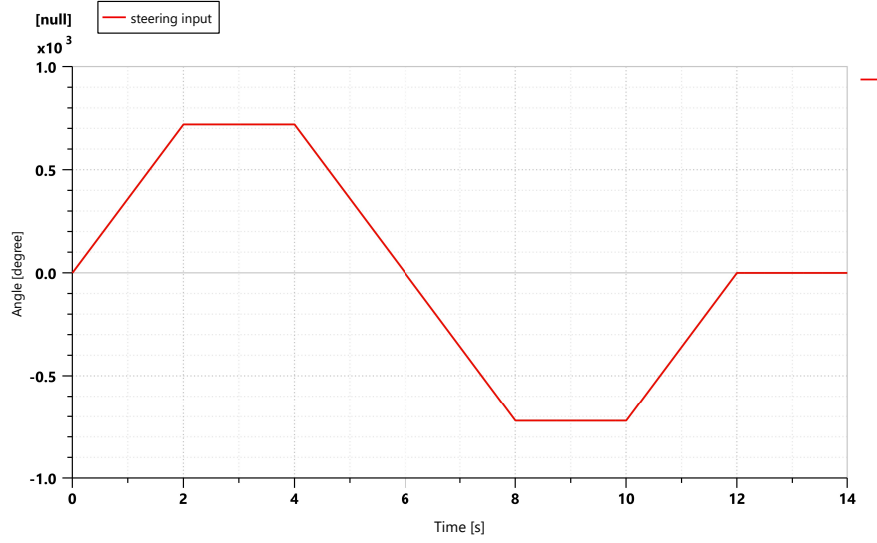


Figure 1.7: Steering input

Considering that the system is designed to make the piston move from one end-stop to the opposite one in the equivalent of 6 turns of the steering wheel, the chosen maneuver does not allow the piston to complete the stroke. Moreover, to complete the 720° steer in 2 seconds a velocity of circa 0.08 m/s is requested for the piston, that corresponds to the 80% of the maximum speed of the system.

Concerning the Amesim model some accuracies should be added, in fact knowing that the OSPD valve and the EHPV cannot operate simultaneously is necessary to impose zero as input for one of the two steering systems when the other is running. Moreover, in the control loop of the EHPV a constant parameter is present that should be equal to 1 when the steer is performed by the electrically actuated valve, instead when the maneuver is carried out by the OSPD valve that parameter should be changed to 0. This way the closed loop system that controls the EHPV can be respectively turn on (1) or shut down (0), it is possible to see this element in Figure 1.6.

Another characteristic that is important to control is the load sensing feature. In this case study, while the EHPV controls the flow rate to the steering cylinder, the load sensing feature of the power supply is designed to be shut off. This is due to the fact that the electrically actuated valve should be fed by a constant displacement source. As shown in Figure 1.3, two on-off valves should be added to

the load sensing circuit: one that connects the pump side to the load side and the other should be added to the pilot line that receives the load pressure. This way when the EHPV is running the pressure of the load is disconnected from the load sensing circuit and a pilot line connects the two sides of the load sensing valve to the same pressure. Consequently, there is no action of the control piston and the pump works at its maximum displacement.

Additionally, to verify the proper functioning of the steering system the interaction between the tire and the ground should be considered. This type of interaction is not easy to replicate due to the high variability of the conditions of off-road terrains. Since the vehicle speed is not really high, lower than 40-50 Km/h, in the evaluation of the parameters that describe how the forces at the contact point evolve, longitudinal slip can be avoided. For sake of simplicity only the lateral forces will be accounted, and for further simplify that evaluation a constant lateral slip is considered. Hypothesizing that the vehicle weight is 25 tons and a coefficient of lateral slip equal to 0.2, it is possible to evaluate the lateral force per axle with the following Formulas 1.12:

$$F_{y,tot} = \mu_y \cdot m \cdot g \approx 50 \text{ kN} \quad (1.10)$$

$$F_{y,front \text{ axle}} = \frac{F_{y,tot}}{2} \approx 25 \text{ kN} \quad (1.11)$$

$$(1.12)$$

Once evaluated the magnitude of the force it is necessary to design the shape of the function that this lateral force should follow. To have a generalized overview of the response of the steering system to different conditions, three different simplified situations will be analyzed:

- Pure resistive load, in this condition the lateral forces stand against the movement of the tires. As it is possible to see in Figure 1.8, the load is present only during the effective motion of the wheels and for sake of simplicity is kept constant in that interval.
- Pure driving load, in this condition the lateral forces pull the wheels, thus the steering system should limit this force to exploit the desired maneuver. The maximum module of the driving load is much lower than the one used in the resistive case, indeed as is possible to see in Figure 1.9 it is set at 5 kN. This depends on the fact that it is not necessary to design a steering system capable of managing too high driving load.
- The last situation is shown in Figure 1.10 and aims to capture a case of steer where the forces are a combination of resistive and driving load.

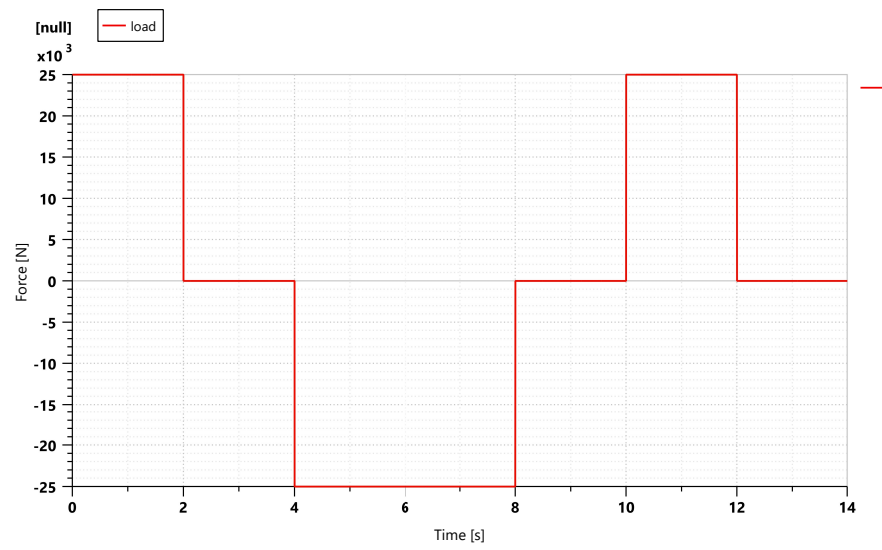


Figure 1.8: Pure resistive load

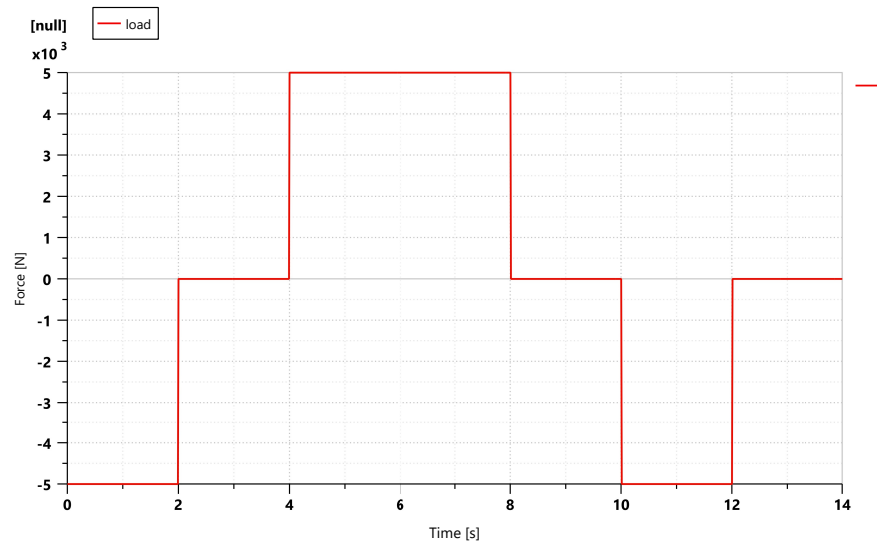


Figure 1.9: Pure driving load

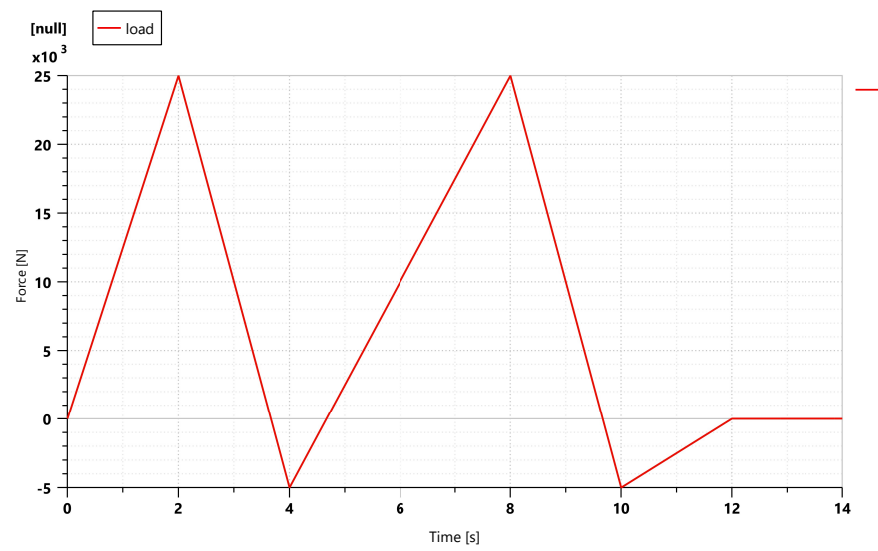


Figure 1.10: Combined load

1.6 Performances analysis

Firstly, it is necessary to verify the accuracy of the steer with the modified OSPD valve. The position sensor attached to the actuator measures the piston displacement, the history of location of the piston is reported in Figure 1.11. This output is then compared to the input of steering by turning it from an angle to the desired position of the piston with the Formula 1.4.

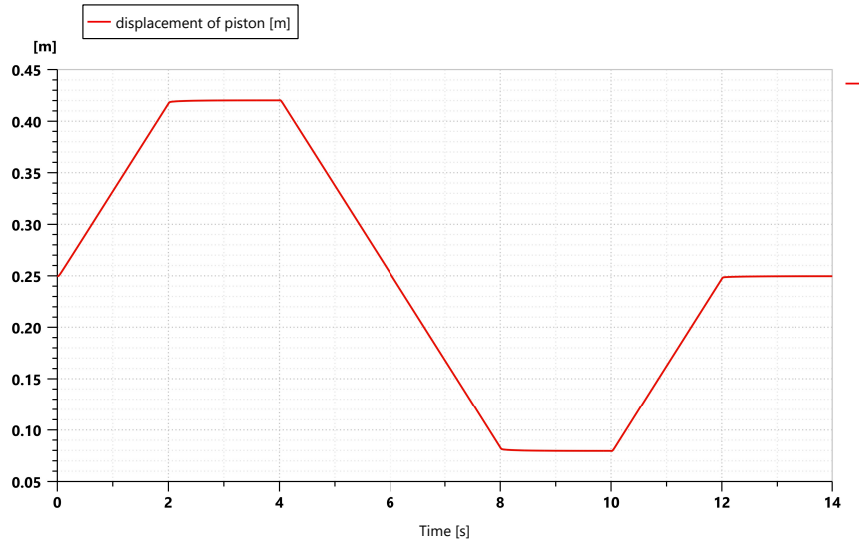


Figure 1.11: Piston displacement with OSPD valve acting

Comparing the actual position of the piston with the desired one emerges that the steering system can steer with an acceptable accuracy, identifying a relative error of circa 3%. Due to this uncertainty, in all the cases analyzed the vehicle behaves as an oversteering vehicle. Moreover, adding two sensors for the measurement of the pressure one at the pump level and the other at the actuator level, it is possible to compute the energy of the system. In particular it is possible to evaluate the energy spent to run the pump and the useful energy, with these two values it is possible to estimate the efficiency of the steering system. At the end of simulation the expended energy is 38.9 KJ while the useful energy is 16.5 KJ, thus from the Formula 1.6 results an efficiency of the steering system of circa 42.3%. This result is consistent with the type of power steering unit analyzed.

$$\eta = \frac{E_u}{E_{exp}} \quad (1.13)$$

Concerning the steer with the EHPV a similar analysis can be done. In Figure 1.12 is reported the curve representing the actual piston position, this output is then compared with the input of steering. In order to make comparable the two values is necessary to transform the input angle of steer into a desired position of the piston using the Formula 1.4.

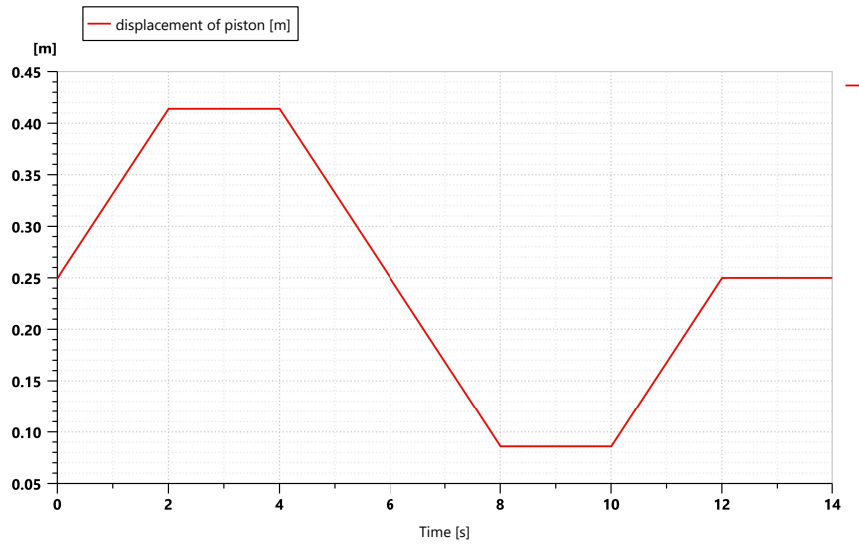


Figure 1.12: Piston displacement with EHPV acting

The simulated automated steering system is capable to achieve high accuracy level, in fact the relative error evaluated under such conditions is lower than the 1%. This result is mainly due to the usage of the closed loop control system of the EHPV. With this type of controller installed it is possible to operate and adjust instantaneously the openings of the valve and thus the final piston position.

For what concern the energy analysis of the steering system with the EHPV operating, the methodology is similar to the one adopted with the OSPD valve case. Taking the outputs from the two sensors it is possible to evaluate the efficiency of the steering system using the Formula 1.7. In this case, the energy spent to move the pump is much larger than the one evaluated in the case of the steer

operated by the OSPD valve, resulting in 110.1 KJ. This is due to the fact that the EHPV system studied is designed to have constant pressure power supply, thus removing the load sensing feature of the system operating with the OSPD. This way the pump works always at the maximum pressure of the supply circuit, set at 100 bar by the relief valve, with the addition of a small offset due to some pressure drop in the few elements present in that section. Instead, the useful energy evaluated at piston level is 15.8 KJ, that is a value close to the one found in the previous case. As a result, the efficiency evaluated for this type of steering system is circa 14%. This value is much lower than the 42.3% of the OSPD case and is due to the increase of the expended energy by the power supply. To conclude, this type of steering system does not have advantage both in the energy efficiency neither in the accuracy, in fact the design of the OSPD valve functioning can be further improved bringing to a higher level of accuracy. At the same time the steering system implemented with the EHPV can operate autonomous steer with a quite simple architecture and with high accuracy. It is important to remember that all the results discussed in this section derive from simulations where lot of simplification are made, especially on boundaries conditions.

1.7 Further developments

Looking at the results, the EHPV steering system modeled works with a very low energy efficiency, while the OSPD valve used for safety reasons has a much higher efficiency. Although the EHPV system can implement the autonomous steering feature the energy loss is too high, thus a modification should be applied to make the system more realistic and implementable in real-word applications. A possible solution could be to keep the Load Sensing (LS) feature, already present in the OSPD valve original system, also when the EHPV is acting. To do this a LS line should be added to the load side of the EHPV, this line should always sense the maximum pressure between the two chambers of the actuator. As it is possible to see in Figure 1.13, a shuttle control valve is added at the ends of the electrically actuated valve to select the higher pressure in the two lines and sends a pilot flow to the LS valve. Also in this case an on/off valve is added to the pilot line to disconnect the LS valve from the load when this feature is not requested.

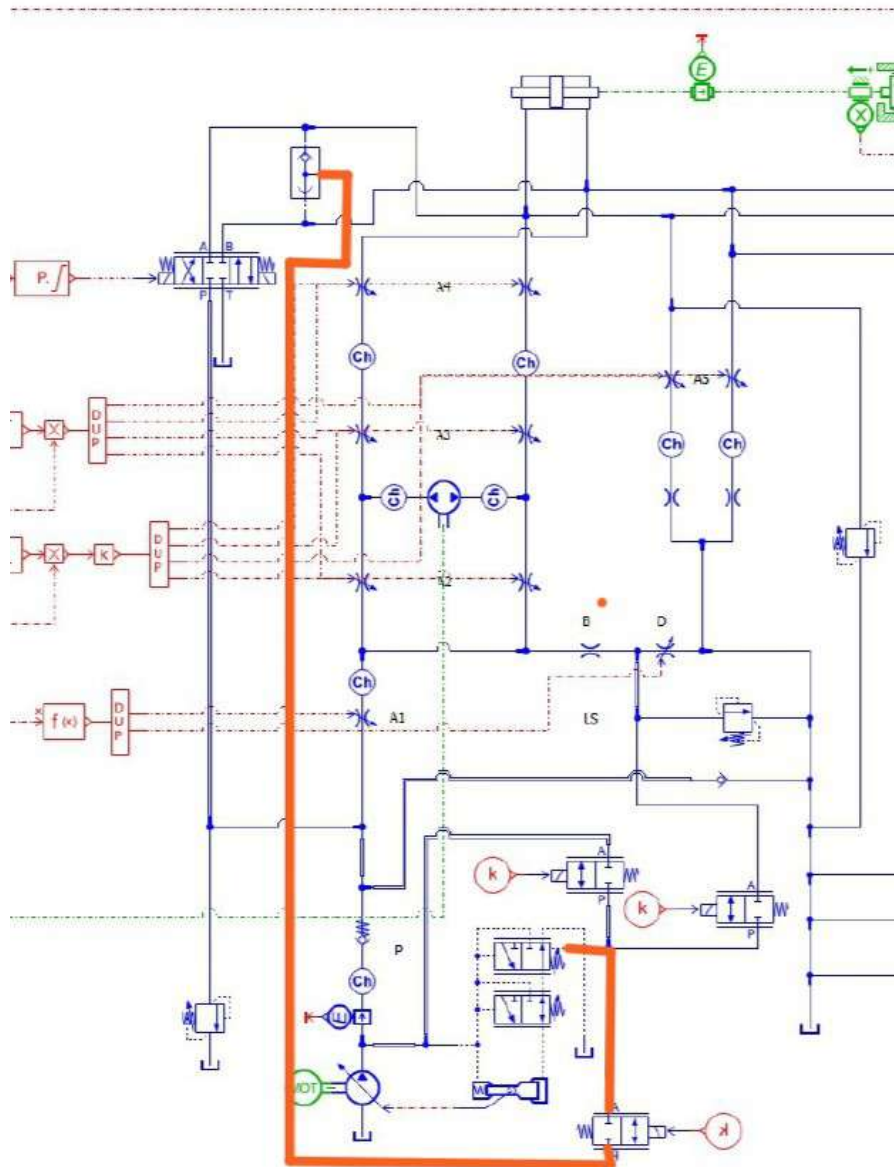


Figure 1.13: EHPV with LS feature

With this implementation the displacement of the pump can be regulated based on the pressure of the load, thus reducing the throttle losses linked to the fixed displacement feature of the flow generation group that characterized the previous

case study. The EHPV system with LS keeps the high accuracy and the reliability of the basic one studied in the previous sections, but the energy spent to drive the pump is decreased reaching the value of 26.6 kJ. The useful energy is maintained the same thus the energy efficiency pass from 14% to 59.4%, making the main steering method more efficient than the one used for safety reasons.

Capitolo 2

Autonomous steer using variable speed prime mover

2.1 Introduction

The second system analyzed is able to operate autonomous steering by exploiting a different feature with respect to the first one, which uses an electrically actuated valve with a proportional controller. In this new case the flow rate is determined acting on the speed of the prime mover, maintaining fixed the pump displacement. The basic idea of functioning for this autonomous steering system is taken from the steer-by-wire solution developed by Singh et al. [5], an hydraulic scheme of this circuit is represented in Figure 2.1. The type of vehicle considered for the analysis is an off-road vehicle of medium-large dimensions and with low vehicle speed reachable. An example of such a vehicle can be a wheel loader of 18-25 t class.

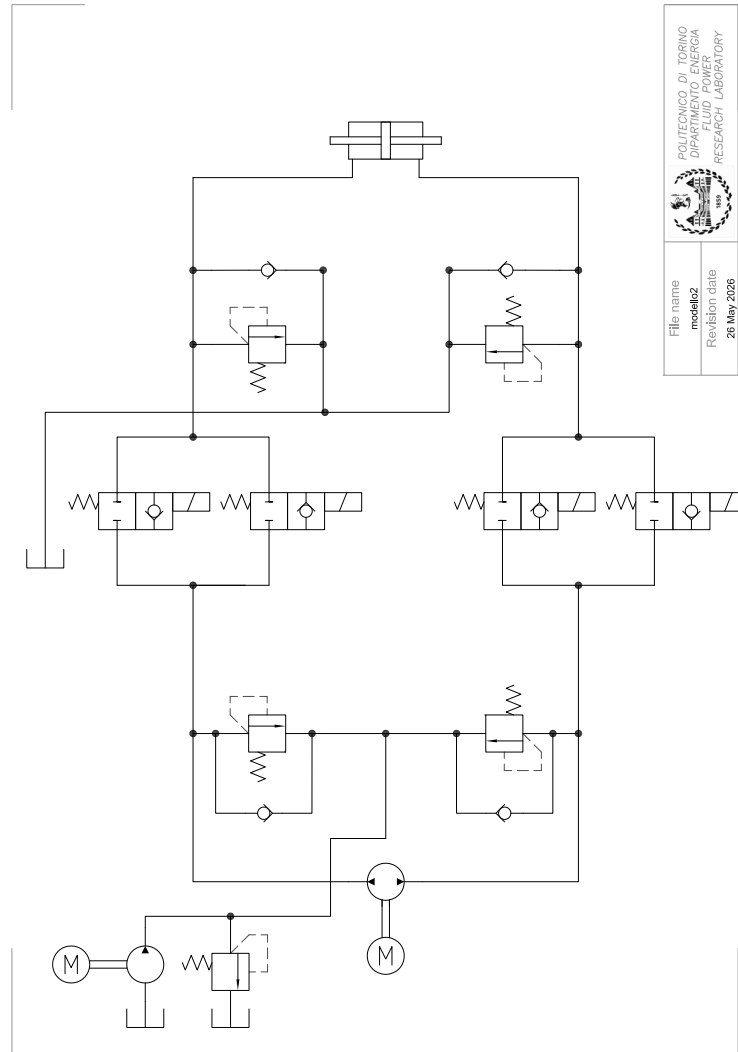


Figure 2.1: Hydraulic scheme of the second model

2.2 Electro-hydraulic steering system

In such system the control of the flow rate is made by adjusting the velocity of the prime mover, thus avoiding most of the throttle losses due to the rate control with a proportional valve. In Figure 2.2 a model developed in Amesim is shown, that is the software in which the whole study will be carried out, as is done for the first model.

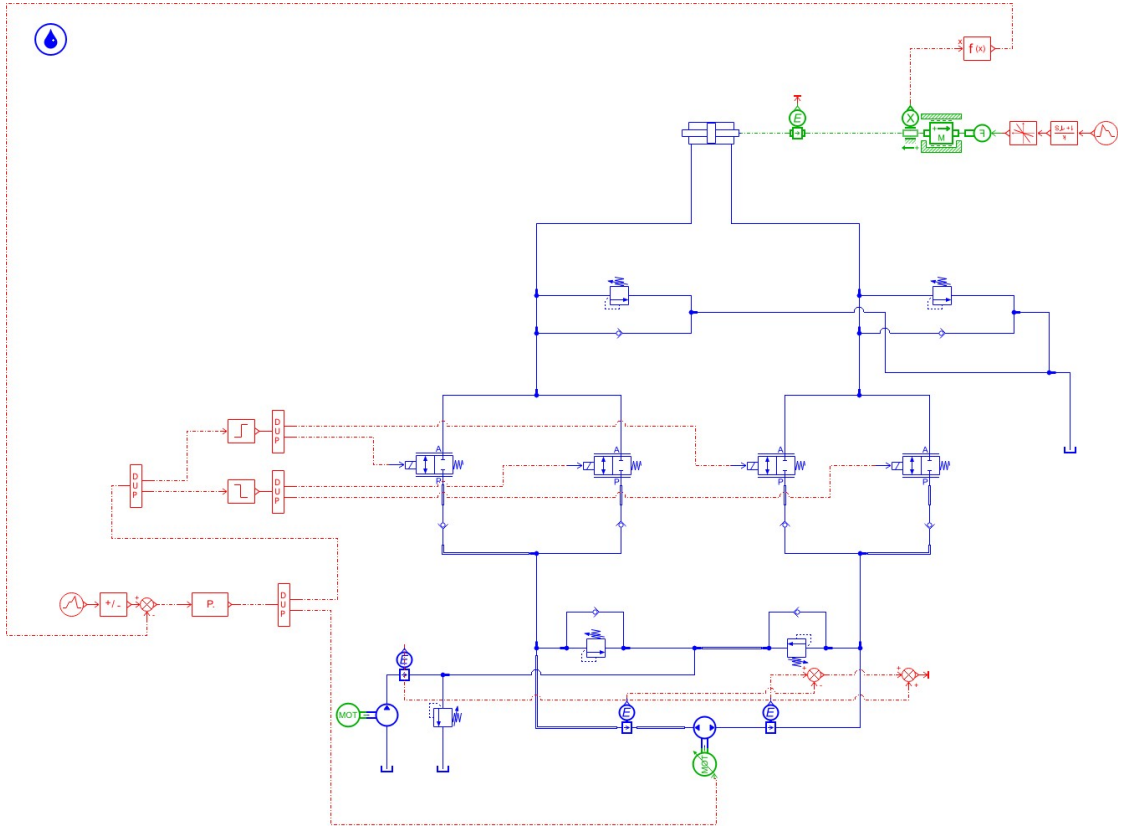


Figure 2.2: Autonomous steering system with control on the motor speed

The peculiarity of this system is the fact that it is a closed loop system where the oil is made recirculate by a double direction of flow pump with fixed displacement. The power supply unit is then completed with a secondary and smaller fixed displacement pump connected to a tank. The main reason to add this secondary flow generation group is to make achievable the flow rate requested by the actuator in case any auxiliaries connected to the circuit request flow. Moreover, the second pump can be useful also to compensate for leakages in the steering system. As it is possible to see in Figure 2.3, the two pumps are driven by two different motors: the main pump, the one on the right, is moved by a variable speed electric motor, instead the secondary pump has a fixed velocity prime mover. The fact that the prime mover velocity of the main pump can be modulated allows to decrease the displacement of the pump, in fact by increasing the motor speed also the flow rate will rise. This way the efficiency of the power supply unit can be increased, indeed a compromise between speed and pump displacement should be done to avoid too high velocity of the motor when the maximum flow rate is requested.

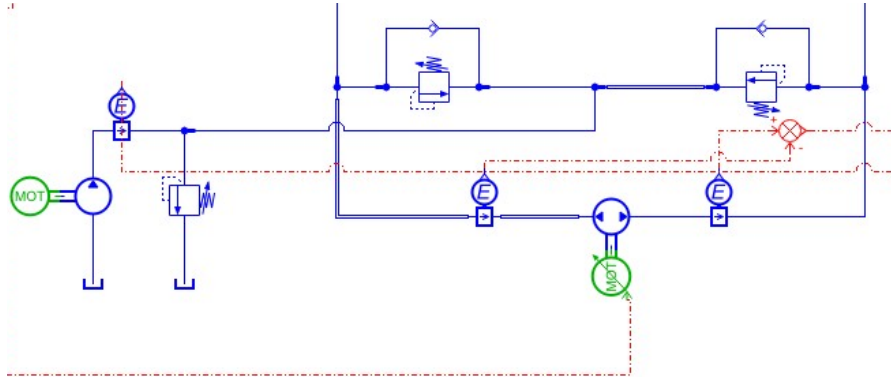


Figure 2.3: Flow generation unit

For the design of the pump displacement it will be considered a steering actuator with the same dimensions reported in the Table 1.1, present in the previous chapter. The aim of this solution is to achieve comparability of all the steering system analyzed in this thesis work. Thus, the evaluation of the maximum flow rate requested by the actuator is the same done for the case of the electro-hydraulic steering system with EHPV, circa 45 L/min will be considered. The smaller pump is designed to generate 15 L/min, this could be a reasonable value considering that is one third of the maximum flow rate requested by the system. Considering that the motor speed is set at 1500 rpm, the pump displacement should be at least 10 cc/rev. Concerning the main pump design the objective is to achieve a flow rate around 30 L/min with not too high motor speed; hypothesizing a rotary velocity of 1500/2000 rpm for the electric motor the displacement of the pump should be at least 20 cc/rev. This is a reasonable value because a relatively small increase of the motor speed is requested to reach the maximum requested flow rate, additionally in any case also the secondary pump can be used to achieve the desired flow rate.

To regulate the speed of the electric motor is used a closed loop control system, as for the EHPV case. The controller takes as input the actual position of the piston, that is measured with a linear sensor attached to the steering cylinder. This information is then processed by a function block so to transform the linear position into an angle position, this process is made through the Formula 1.4. This angle is compared with the desired steering angle thus producing an error and, as is possible to see in Figure 2.4, this outcome is given as input to a Proportional controller (P).

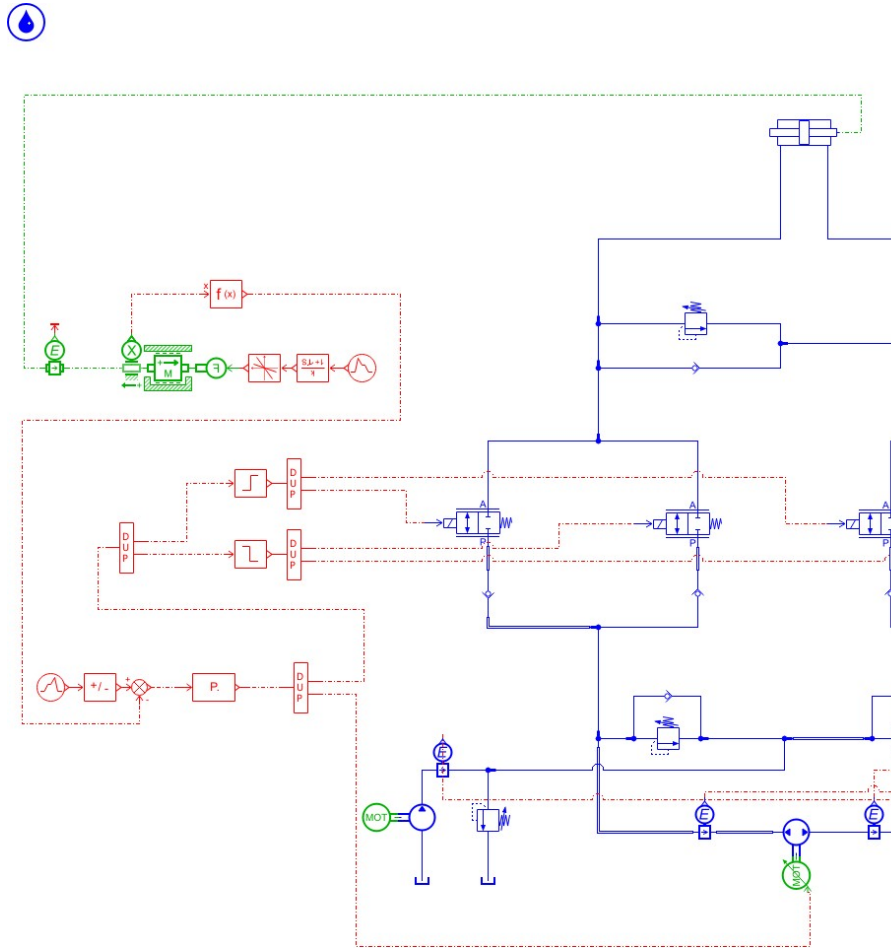


Figure 2.4: Closed loop control system

The design of the parameter of the controller is made through a trial-and-error technique, trying to achieve a solution where the actual piston position is as close as possible to the desired one. From this process it turns out that an optimal value of the proportional gain is -30. The output signal is used both to control the vehicle speed and to manage the openings of the valves. The possibility to decide the velocity of the motor allows the system to self-regulate the flow rate without the need of a proportional valve, thus limiting the throttle losses. In fact, as is possible to see in Figure 2.5, in this steering system four on-off valves are present with the aim of directing the oil flow. These valves are positioned in pairs each side of the pump and they cannot have the same position simultaneously, thus only one valve per side can be in on position.

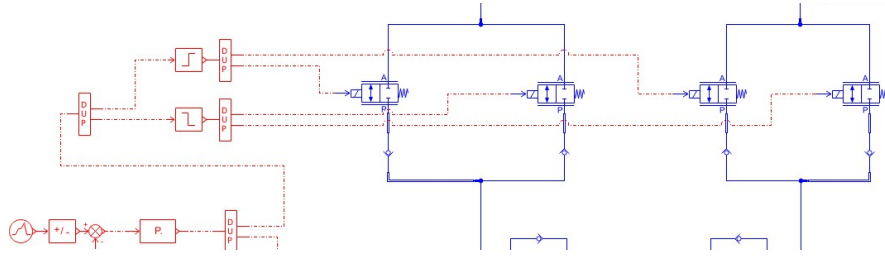


Figure 2.5: On/Off Valves configuration

Going in further detail, each on-off valve is in combination with a pressure controlled valve so to allow only one direction of flow. With this valves configuration for both sides of the pump allow both in and out going flow, for this reason a controller is needed to manage the openings. The idea beyond this type of control is quite simple, the output from the P controller could be positive or negative to allow the motor spin in both directions. The flow rate generated by the pump will follow the direction imposed by the direction of rotation of the prime mover, so the same signal used to regulate the motor speed can be used to manage the valves openings. For example, if the signal is positive only one on-off valve per side will be set in open position and these two valves will necessarily allow opposite direction of flow. This way the flow can go from the pump to one chamber of the steering actuator, meanwhile the other chamber will be connected to the other side of the pump allowing the flow to go back in it. In order to do that the output signal from the controller is duplicated and then the positive and negative parts are split through a high-pass and low-pass filter. The pressure relief valves and pressure control valves represented in Figure 2.6 ensure the safety of the steering system, imposing a maximum pressure of 150 bar and preventing the phenomenon of cavitation.

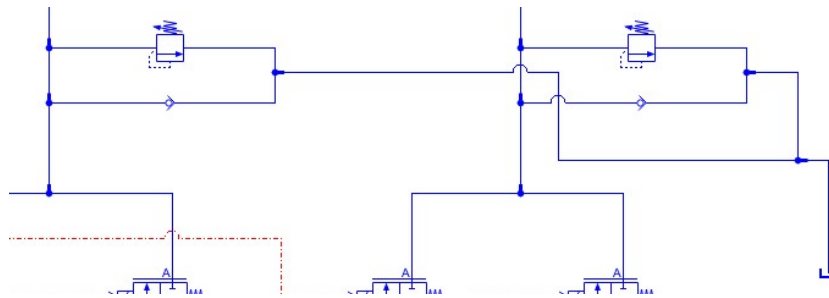


Figure 2.6: Safety system

2.3 Simulation setup

The steering input is kept the same as the previous case study, Figure 1.7, so to impose the same boundaries to the steering system. The cornering ranges from a desired angle of steer of 720° to -720° , this way the piston in the steering cylinder will move for the 65% of the total stroke as analyzed in the first case. Going into further details, the time in which the system is requested to operate makes the desired flow rate high enough to be considerate challenging for the system. In fact, looking at the Figure 2.7 it is possible to see that the oil rate reaches values on the order of 30 L/min, almost $\frac{3}{4}$ of the maximum rate for which the system was designed.

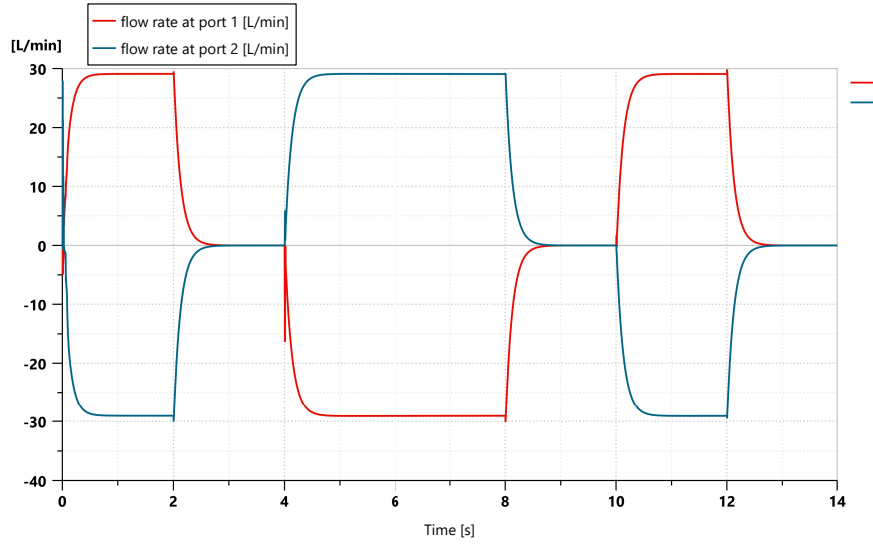


Figure 2.7: Flow rate requested by the actuator

For what concern the interaction between tire and ground the same boundaries are considered, the velocity of the vehicle is hypothesized lower than 40/50 km/h thus only the lateral slip coefficient will be take into account. Based on the same class of weight of the vehicle, of circa 25 tons, and considering a lateral slip coefficient of 0.2, the magnitude and the shape of the lateral forces are kept similar to the ones used in the previous analysis. The three types of lateral forces reported in Figures 11, 12, 13 are considered, this way the steering system is subjected to a large spectrum of external conditions. To make smoother the force function, also in this case a first order lag is added to the system; imposing the time constant at 0.1

s and the “k” gain at 1, the step-like behavior of the forces is eliminated. This way is easier for the system to response to the external forces, avoiding instantaneous change of the pressure on the actuator side that will be difficult to follow by the flow generation unit.

2.4 Performances analysis

Taking into account all the simplifications made for the tire-ground interaction, not considering the vehicle dynamics and the almost ideal hydraulic components of the model, a preliminary study of the main parameters is carried out to understand the behavior of the electro-hydraulic steering system.

The first analysis to be made is on the accuracy of this steering system and its controller. In Figure 2.8 is shown the piston position during the simulation, this outcome is then compared to the desired steering input to determine the error. The position of the piston in the steering cylinder is measured through a sensor attached to the actuator and the output is given in meters. Knowing that the steering input measurement unit is degrees, to compare these values is necessary to convert it to a position in meters through the Formula 1.4.

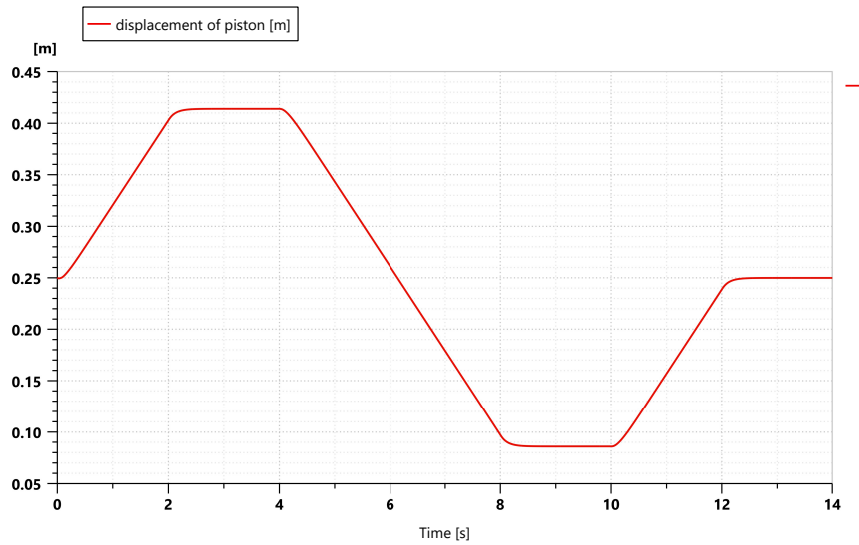


Figure 2.8: Piston displacement with variable speed motor solution

The error evaluated this way is lower than the 1%, the system is able to ensure a good accuracy and almost no errors in replicating the given input. This high

reliability is mainly due to the control system and the accuracy of its design. In fact, the use of a closed loop control system allows to make adjustments almost in real time thus resulting in high accuracy level of this autonomous steering system. The error does not increase switching between the three scenarios of resistive, driving and mixed load; this result is important to verify that the behavior of the steering system does not change modifying the external load case. Therefore, this autonomous system can guarantee high level of accuracy even with various driving scenarios.

Another important behavior of the steering system to be checked is the evolution of the pressure during the simulation. It is necessary to verify that the maximum pressure does not reach too high value that might deny the safety of the system. As it is possible to see in Figure 2.9, the pressure in the two chambers of the steering actuator does not reach values higher 75 bar; having the maximum peak at 70.5 bar only in the first instant of the second cornering action. Moreover, the difference in pressure between the high and low pressure side of the actuator is maintained roughly constant at 40 bar, thus guaranteeing a back pressure always lower than 30 bar.

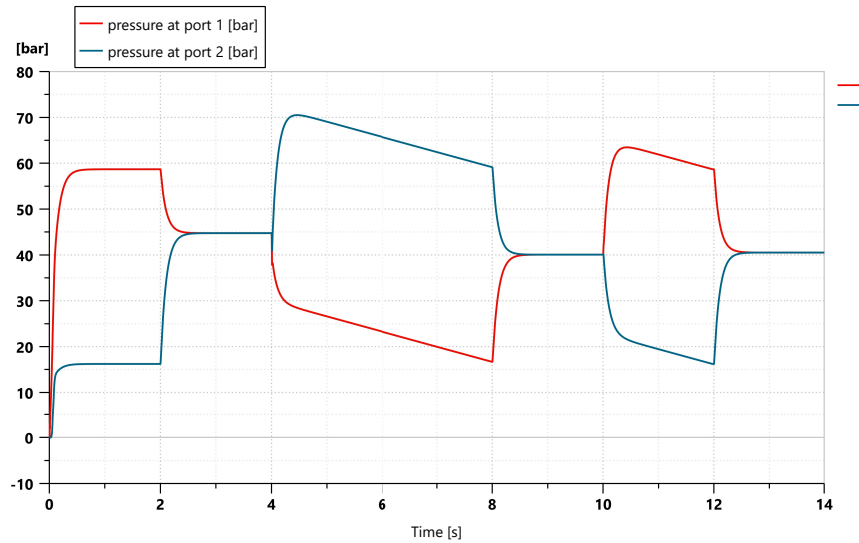


Figure 2.9: Pressure at cylinder's ports with resistive load

Additionally, in all three segments the back pressure tends to decrease, reaching a stable value of 15 bar that is a reasonable value for this type of systems. The value of the minimum pressure is imposed by the secondary flow generation unit,

in particular the calibration of the relief valve defines the system back pressure. That valve is designed with a spring pre-tension of 10 bar, as it is possible to see in Figure 2.10 the relief valve is regulating for the whole duration of the simulation, thus imposing the minimum pressure to the circuit. This first analysis is made with the resistive load, but things do not change when driving load is introduced. As shown in Figure 2.11, the main difference is that the maximum pressure reached is 23 bar, it is much lower than in case of resistive load due to the lower force simulated. Moreover, in the three segments of the steer the pressure trend is rising, instead of decreasing, but this characteristic only depends on the nature of the load. The minimum pressure is kept between 13 bar and 14 bar by the low pressure source as in the resistive load case, thus decreasing the delta pressure between high and low pressure sides.

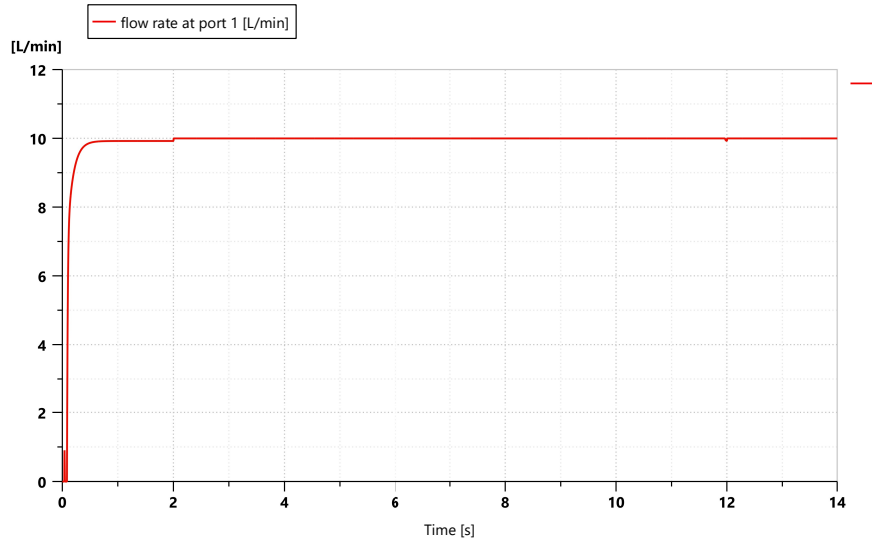


Figure 2.10: Flow rate discharged by the relief valve

The main features of the pressure characteristics inside the steering cylinder are the same also when the third load case is introduced, simulating a mixed load case. In fact, looking at Figure 2.12 which represents the pressure characteristics of the steering actuator, the maximum pressure reached during the simulation is roughly 65 bar. Moreover, when one chamber reaches its local maximum pressure in the other one the pressure is at its local minimum, providing 40 bar of pressure difference between the two sides and a low back pressure.

Concerning the energy analysis of the steering system the resistive load case will

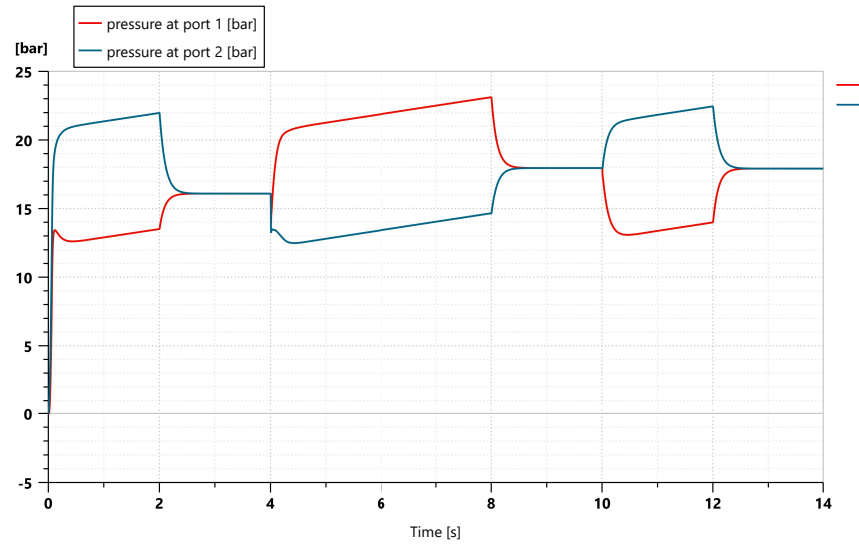


Figure 2.11: Pressure at cylinder's ports with driving load

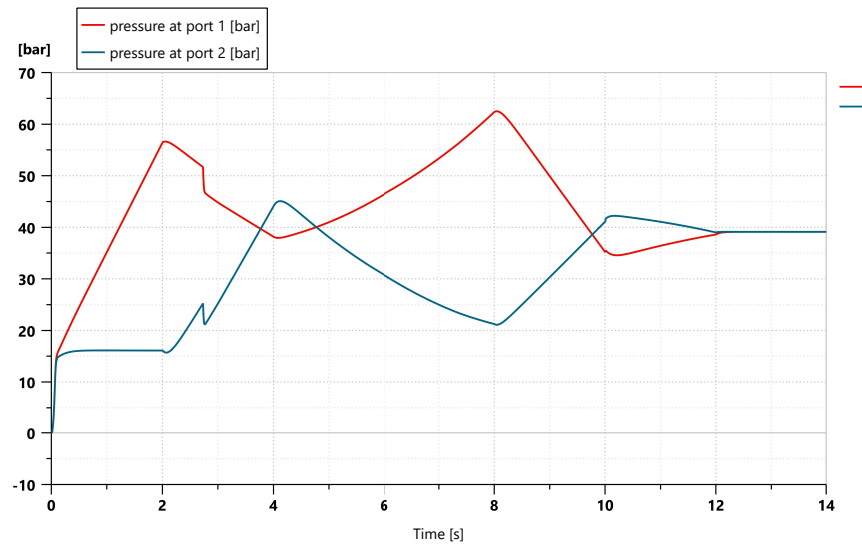


Figure 2.12: Pressure at cylinder's ports with mixed load

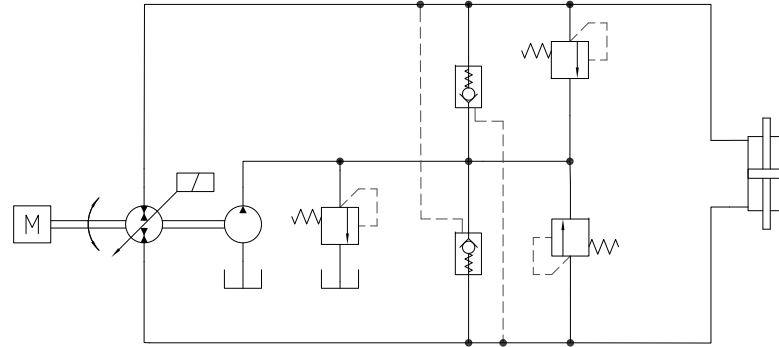
be considered, this way the power generated by the flow generation unit will be used to win the resistive load. Therefore, a comparison can be made between the energy used by the power supply and the energy spent to move the steering actuator, subjected to external forces. To measure the energy spent by the main power supply two sensors are used, one each side of the pump, thus it is possible to evaluate the amount of energy used to provide power to both the chambers of the actuator. Indeed, the output from the two sensors are summed together and the total energy spent by the pump is evaluated, resulting in 17.2 kJ; to this value should be added 3.5 kJ of energy spent to drive the secondary pump for a total of 20.7 kJ. Another sensor is attached to the steering cylinder to evaluate the useful energy, the output value is 15.7 kJ that is consistent with the ones evaluated in the analysis of the EHPV and the OSPD valve solutions. This is due to the fact that the boundary conditions of the case study are the same, resulting in similar useful energy consumption. Once computed both the useful and the expended energy, it is possible to evaluate the efficiency of the steering system by using the Formula 1.6. From that calculus the total efficiency of the system is 75.8%, which is an high value but consistent with the hypothesis made for the simulation. In fact, the system can be considered almost ideal having minimum pressure drops across the valves and no leakages are considered, resulting in 8% of the total energy lost by these valves. The other element that increases the energy loss is the pressure relief valve attached to the smaller pump, in fact this valve is always in regulation discharging flow rate and decreasing the overall efficiency of the system. Moreover, this type of steering system allows to reduce at zero the throttling losses by generating the exact flow rate that the steering actuator request. This feature is mainly due to the implementation of the closed loop controller that can adjust and regulate almost in real time the speed of the electric motor, thus allowing the flow generation unit to produce the exact amount of flow rate requested. To conclude, this type of steering system can achieve good result both in terms of energy efficiency and accuracy of the steer while maintaining high level of safety. Further developments of the model can be done to test the advantage of installing an electrically actuated proportional valve to increase the redundancy of the system. This valve can be fed with another hydraulic circuit with its own flow generation unit, using a system already present in the vehicle that can replace the main one in case of any failure.

Capitolo 3

Autonomous steer using variable displacement pump

3.1 Introduction

The third steering system analyzed operates autonomous steering with a different solution with respect to the first two systems analyzed. In fact, this case study will analyze a power steering system where the flow generation unit has a variable displacement pump, allowing to regulate the flow rate adjusting it to the requested one. To better understand the operation of this system, a simple hydraulic scheme is reported in Figure 3.1. The vehicle considered for this analysis is the same used in the previous chapters: a wheel loader with medium-large dimensions and limited range of speed.



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Figure 3.1: Hydraulic scheme of the third model

3.2 Steering system

The autonomous steering system analyzed is based on the steer-by-wire solution developed by Daher et al. [6]. This system takes advantage of a variable displacement pump for the control of the flow rate to the actuator, thus reducing the throttle losses due to the usage of a proportional valve. The model of this steering system developed in Amesim is reported in Figure 3.2.

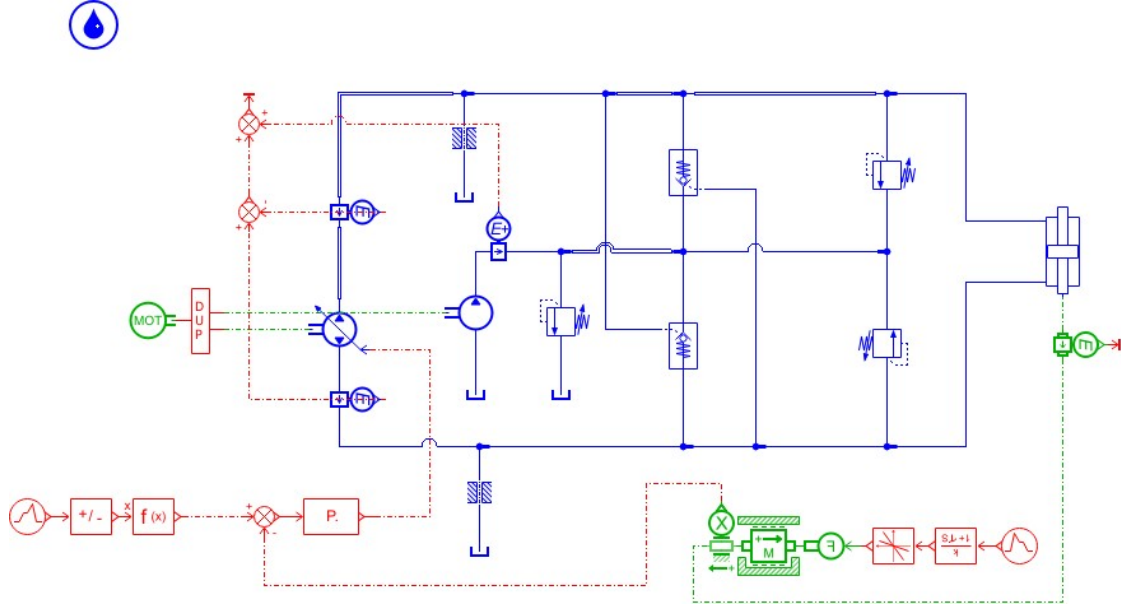


Figure 3.2: Autonomous steering system with control on the main pump displacement

Similarly to the steering system with the variable speed prime mover, the system analyzed has a closed loop configuration with a double direction of flow pump that makes the oil recirculate. The variable displacement main pump along with a fixed displacement secondary pump are driven by a fixed speed prime mover. The secondary pump is characterized by a smaller displacement with respect to the main one; its main function is to make available flow rate to the steering system in case auxiliaries, connected to the circuit, request for flow. Moreover, the secondary pump can be useful to compensate for small oil leakages, as simulated in this case study. In Figure 3.3 is reported a zoom of the power supply unit where it is possible to see that the two pumps are connected to the same prime mover through 2 distinct shafts. Since the speed of the prime mover is fixed the primary pump should be designed in such a manner that at its maximum displacement is able to fulfill the maximum flow rate requested by the actuator. This way when the required flow rate is lower than the maximum, the primary pump displacement is

lowered maintaining high efficiency of the flow generation group.

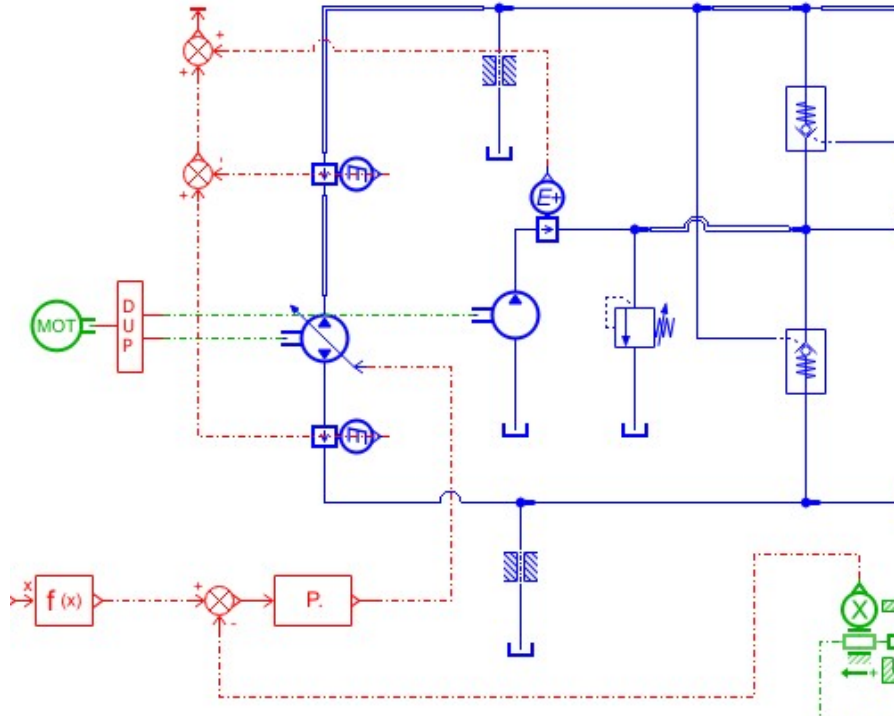


Figure 3.3: Power supply unit

Going in further detail, the dimensions of the steering actuator reported in Table 1.1 will be kept unchanged to make the systems comparable; also the hypothesis on the steering conditions, like the speed of the piston and the maximum time to complete the end-to-end stroke, will be kept the same thus resulting in similar flow rate demand. Considering a maximum rate of 45 L/min and hypothesizing the motor speed at 1500 rpm, the maximum displacement of the primary pump can be set at 30 cc/rev. This way the flow rate requested can be satisfied without occurring in too high partialization of the pump displacement, thus avoiding excessive reduction of the pump efficiency. Moreover, in a variable displacement pump if the maximum displacement is designed too high, while the demanded flow is lower than the maximum, the pump will work at too low fractional displacement resulting in a decrease of the efficiency.

Concerning the design of the secondary pump the speed of the prime mover is 1500 rpm; since the maximum demanded flow rate by the steering actuator can be fulfilled by the main pump, the rate brought by this secondary flow generation unit can be set relatively small. In this case study, the displacement of the smaller pump is 5 cc/rev thus adding 7.5 L/min that can be a consistent rate for the usage

of small auxiliaries and is enough to recover the leakages simulated in this case study.

The modulation of the pump displacement is made through a closed loop control system, as for the previous cases. As is possible to see in Figure 3.4, a linear sensor is attached to the actuator to measure the position of the piston instant by instant, then this information is sent to a function block that processes it. In fact, in this block the linear position of the piston is transformed into a steering angle through the formula 1.4, this passage is important to make the actual steer and the desired one comparable. To evaluate the error between the two steering values the steering input sign is inverted and then a summing function is used. The resulting error is sent to the Proportional controller (P) so to produce a correction factor for the modulation of the pump displacement.

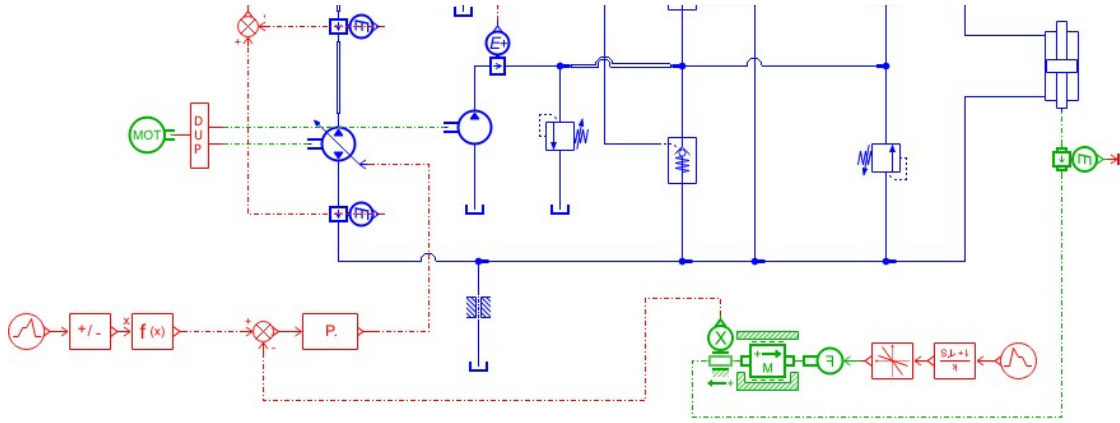


Figure 3.4: Controller with modulation on pump displacement

The Proportional controller (P) uses a proportional gain to evaluate the fractional displacement of the main pump so that the generated flow rate matches the one required for the desired maneuver. To evaluate the optimal value for the proportional gain it is used a trial-and-error technique, this method applied in cases where few variables should be handled is very effective and quick. The optimal gain found this way is -60, this value allows to minimize the error between the actual steer and the desired one reaching the stability in small time. Thanks to the closed loop configuration of the control system it is possible to adjust the pump displacement in real time, this feature is an important purpose for this type of steering system. Indeed, regulating the displacement of the pump while the speed of the prime mover is kept fixed results in controlling the flow rate to the actuator. This way throttle losses due to the usage of a proportional valve are

avoided, increasing the efficiency of the system.

The output from the P controller is exactly the fractional displacement of the pump that generates the requested flow rate by the actuator, in real life implementation probably another component should be added to the system. Indeed, to control the displacement of the pump an electrical commanded actuator should be used, in this case the output of the P controller will not be directly the displacement of the pump but should be an electric signal to move the actuator connected to the variable displacement pump. For sake of simplicity in this case study this intermediate passage is not present and the controller can directly adjust the pump displacement.

Another function of the secondary flow generation could be exploited in the case of an asymmetric actuator, as studied in the work of Daher et al. [6]. Under these conditions, the 2 sides of the steering actuator will be subjected to different flow rate so the smaller pump is used to compensate for this difference. This secondary flow generation unit is always connected to the low pressure side of the actuator by means of the two pilot-operated check valves represented in Figure 3.5, this way it is possible to both provide or absorb flow preventing evacuation. For safety reason two relief valves are connected to the two sides of the steering cylinder, thus limiting the maximum pressure reachable by the hydraulic steering system.

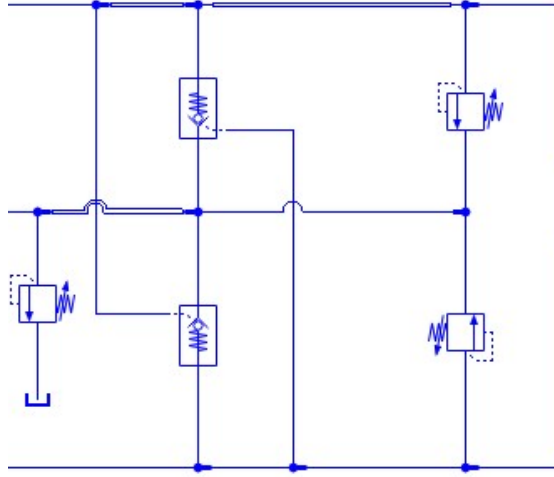


Figure 3.5: Check and relief valves configuration

3.3 Simulation setup

Aiming to maintain the system comparable with the others analyzed in this thesis work, the steering input is kept the same as in previous cases. The input maneuver represented in Figure 1.7 consists of two consecutive steers, the first to the left and the second to the right, and finally the steering wheels are realigned in the last seconds. During the steer, the desired angle for both directions reaches a maximum absolute value of 720° thus moving the piston inside the steering actuator for almost 65% of its maximum stroke, never reaching the end-stops of the actuator. Looking at the requested flow rate shown in Figure 3.6, the steering input given imposes to the system quite high flow rate with respect to the maximum rate for which the flow generation unit is designed.

The interaction between the tire and the ground is modeled under the same hypothesis made also in the other case study. Indeed, low vehicle speed are considered and for sake of simplicity only the lateral force behavior is taken into account. To perform the analysis of the steering system three cases of lateral force will be considered so to investigate the behavior of the system in different scenarios.

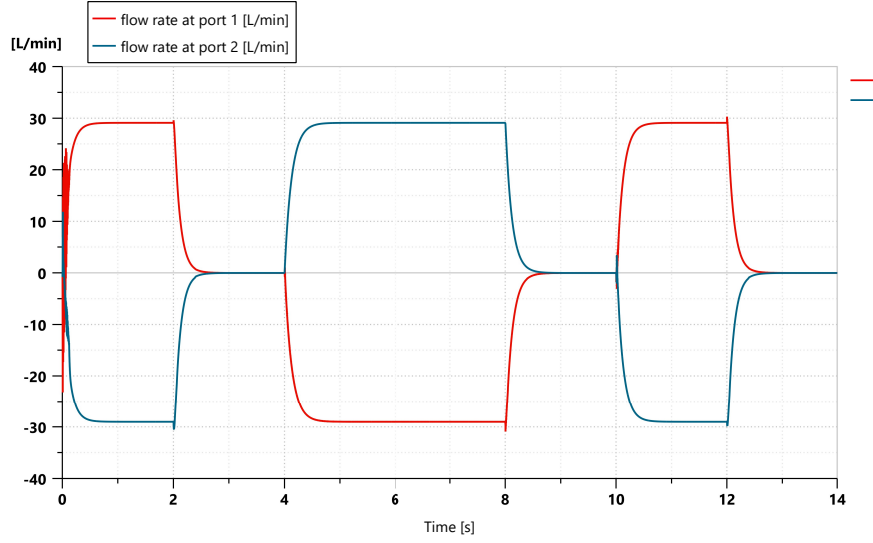


Figure 3.6: Flow rate at steering cylinder

Moreover, the lateral slip coefficient is hypothesized with a value of 0.2, the weight of the vehicle is kept in the same range thus resulting in the same force magnitude and functions: resistive load shown in Figure 1.8, driving load Figure 1.9 and mixed lateral force Figure 1.10. Going into further details, the first two loads have a shape like a square wave that is difficult to follow by the system, indeed some fluctuations of the pressure and the flow rate are generated when rapid changes of the load are present. To overcome this issue a first order lag block is added; the time constant “tau” is set at 0.1 s and the “k” gain at 1, with these values the curve of the load is made smoother and the generated fluctuations of pressure and flow are avoided.

3.4 Performances analysis

Firstly, an analysis of the actuator and the piston displacement should be done to verify the accuracy of the steering system. To do that a comparison between the actual piston position and the desired one is made, since the measured displacement is in meters the desired steering angle used as input should be transformed in meters through the formula 1.4. The actual piston position represented in Figure 3.7 is measured with the same sensor used to give displacement information to the control loop.

The error evaluated with this strategy is lower than the 1%, as for the previous cases, thus ensuring good system accuracy. The desired input is replicated without

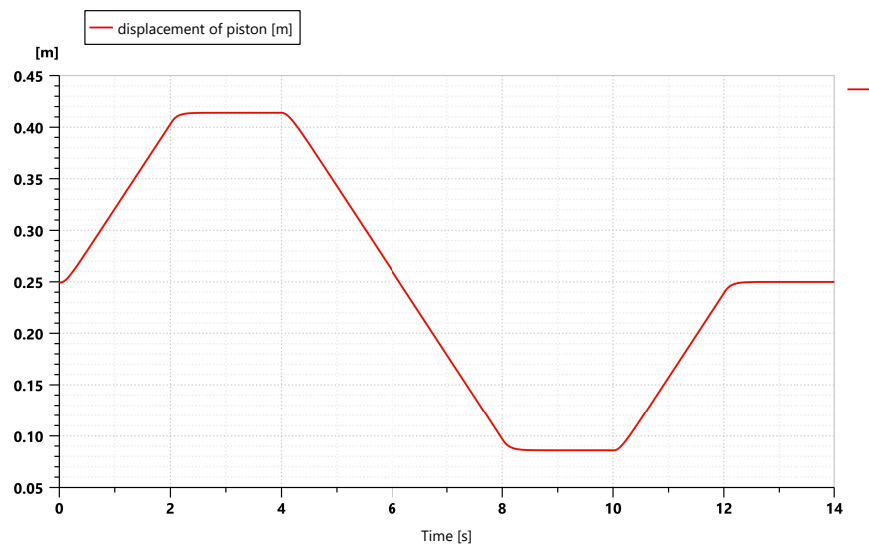


Figure 3.7: Piston displacement with variable displacement pump control

the increase of the error even changing the scenario from resistive load to driving load or mixed load, thus guaranteeing high reliability in a large range of off-road applications. Both the fact that the error of the actual piston position is low and the fact that it will not change with different external load scenarios are due to the control loop configuration. Indeed, using a feedback closed loop control system is possible to adjust almost in real time the position of the piston making the autonomous steering precise and safe.

To verify the correct behavior of the autonomous steering system, it is important to also inspect the evolution of the pressure. It is important to verify that the pressure within the hydraulic system does not reach a peak that is too high, which could affect the safety of the system. Moreover, the pressure difference between the high and low pressure side of the steering actuator should remain in the range of 40 bar like in the case of the EHPV and the variable speed prime mover case. Looking at Figure 3.8, where the time evolution of the pressure in the two actuator ports is reported for the case of external resistive load, it is possible to see that the maximum pressure is 56 bar. Instead, the low pressure side is always kept at 13 bar by the low pressure source; the pressure relief valve present in the secondary flow generation unit is designed with a spring pre-tension of 10 bar thus imposing the pressure to the low-pressure side of the circuit. The small oscillations present are due to the inertia of the flow rate while the steering actions are started or ended.

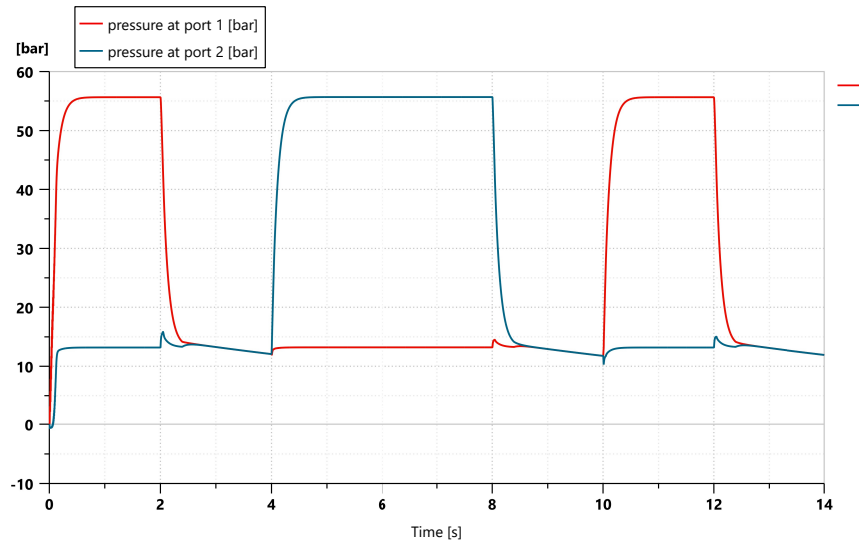


Figure 3.8: Pressure at actuator's ports with resistive load

The situation is similar when the load is switched from resistive to driving, as is possible to see in Figure 3.9. In fact, the minimum pressure of the system is kept at 13 bar thanks to the low pressure source, maintaining a high level of control on the steer setting the back-pressure of the actuator. In this case the maximum pressure is roughly 22 bar, much lower with respect to the value reached in the previous case study, depending on the amount of external force imposed. The overall behavior is the same also when the mixed load case is studied, the maximum pressure reached is 54 bar, which is a value in line with the ones measured in the resistive load cases. Moreover, as shown in Figure 3.10, also in the third case study of this autonomous steering system the low pressure source kept the circuit low pressure side always at a minimum of 13 bar. Thus, ensuring a back pressure high enough to guaranty the control also in the case of the driving load.

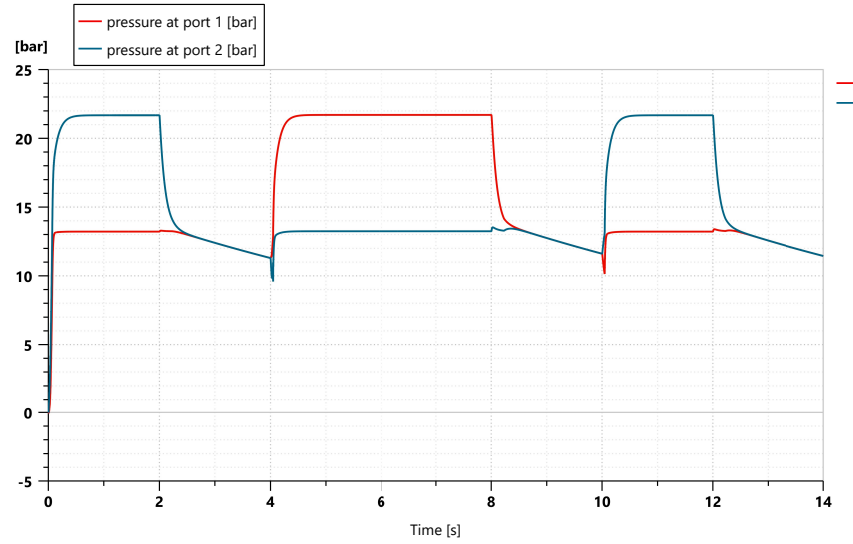


Figure 3.9: Pressure at actuator's ports with driving load

Another characteristic that should be studied to make a comparison with the other autonomous steering system analyzed is the energy efficiency; this parameter is evaluated by dividing the useful energy by the one expended, as reported in Formula 1.6. The energy expended is mainly the one used to move the primary pump but in a system like this, where a secondary flow generation group is included, the contribution of energy use to drive the secondary pump should be considered. The case study is kept the same as in the previous chapters, thus the eternal resistive load will be considered to compare the results of different solutions with the same boundaries. Concerning the energy expended, the main pump is the one that gives

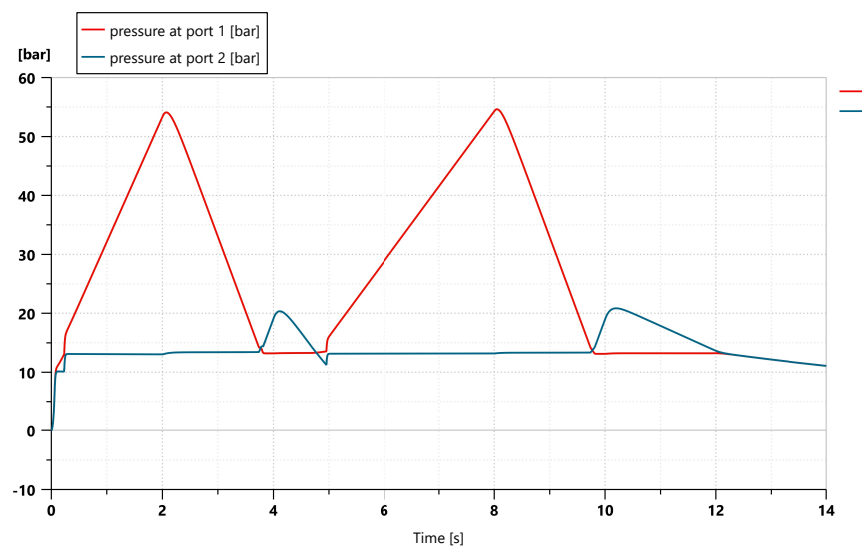


Figure 3.10: Pressure at actuator's ports with combined load

the biggest effort contributing with 15.8 kJ to the total 18.3 kJ of energy spent. The useful energy is 15.7 kJ, considering this, the 2.5 kJ spent to drive the smaller pump are the principal loss in efficiency of the system; in fact, looking at Figure 3.11, it is possible to see that the entire flow rate generated by this secondary unit is discharged with the relief valve most of the time.

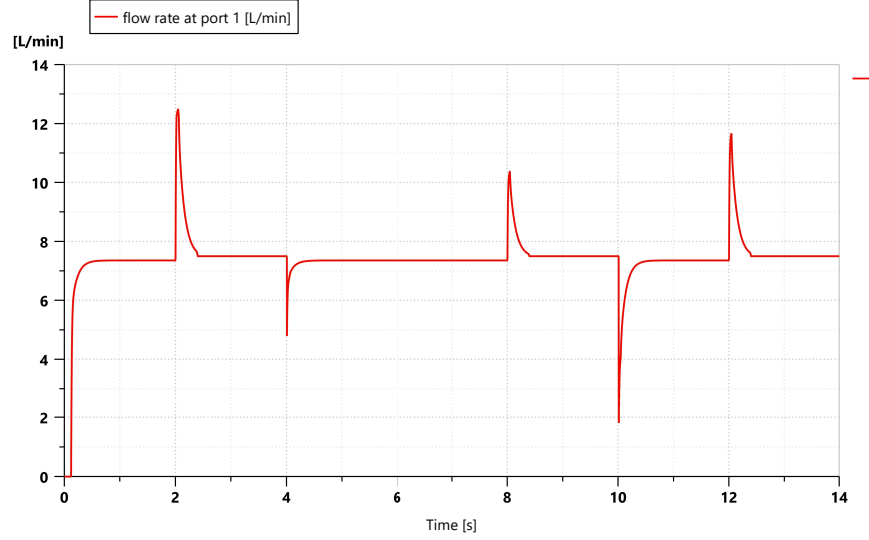


Figure 3.11: Flow rate discharged by the relief valve of the secondary flow generation group

The total energy efficiency evaluated for this autonomous steering system is 86%. Since in this system is not present any directional or proportional valve that could generate losses, the high value of the efficiency is consistent. Indeed, the flow rate is regulated by the controller acting directly on the main pump displacement thus feeding the hydraulic circuit with the exact amount of flow that the actuator requests. In addition to the losses linked to the secondary flow generation unit, another contribution comes from the flow losses generated with the two laminar orifices, but these types of losses do not affect in a relevant manner the final total energy efficiency. In conclusion, this steering system reaches high value in terms of energy efficiency, the safety is always ensured and the accuracy of the maneuver is kept at a high level.

It is important to keep in mind that relevant simplifications are made during this study, like the absence of vehicle dynamics, the simplified tire-ground interaction and the almost ideal hydraulic components of the model. For these reasons, the

results obtained must be taken as a preliminary study of the behavior of the electro-hydraulic steering system.

Capitolo 4

Electro-hydraulic steering system by Danfoss

4.1 Introduction

The last solution analyzed is the one found by Danfoss [3], this type of pure steer-by-wire steering system exploits a similar idea of the first one, indeed in this steering system the flow control is modulated through an Electro-Hydraulic Proportional Valve (EHPV). Moreover, as is possible to see in Figure 4.1, there are two more Electro-Hydraulic (EH) valves that, along with the EHPV, are controlled by an ECU. This solution is already implemented in some off-road vehicles thus this case study can give an industrial view, adding a critical view to the thesis work.

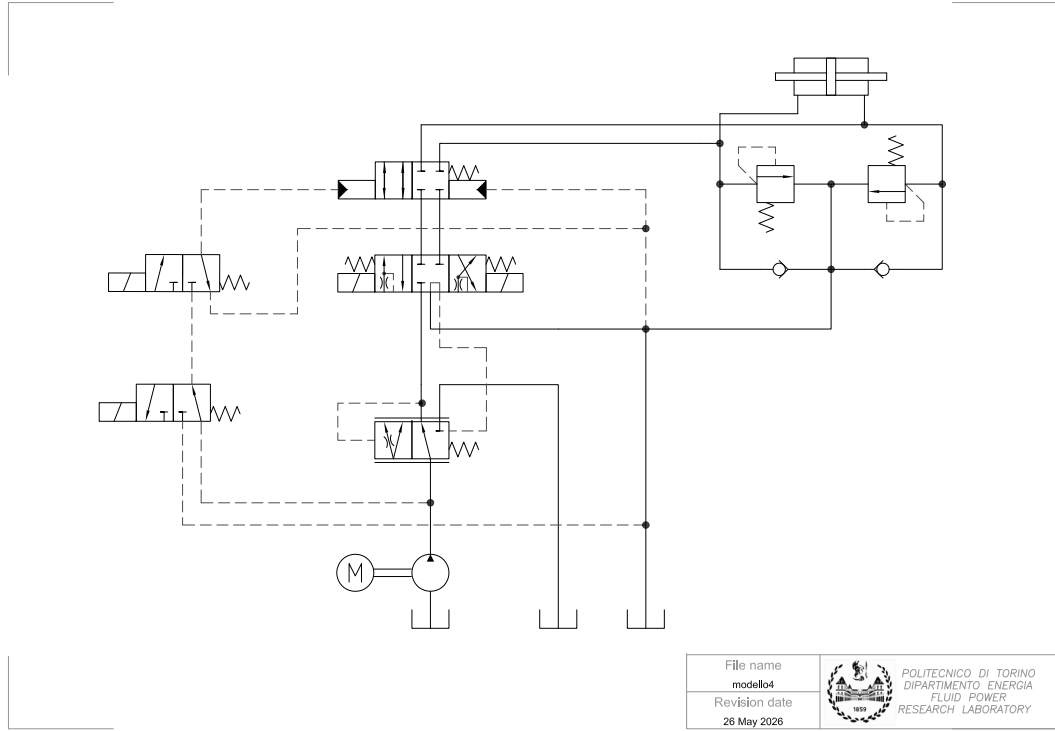


Figure 4.1: Hydraulic scheme of the fourth model

4.2 Electro-hydraulic steering system simplified

The autonomous steering system represented in Figure 4.2 takes inspiration from the pure steer-by-wire system developed by Danfoss [3], but for sake of simplicity some components are removed and some operations are simplified. An example could be that the PVED CLS control unit that is not present in the developed model, the control instead relied on a feedback control loop with a controller that modulate the output. Moreover, one of the two EH valves in the original scheme was designed as a pilot reduction valve, but aiming to simplify the model this valve is modified in an electrically actuated valve with the same features of the original.

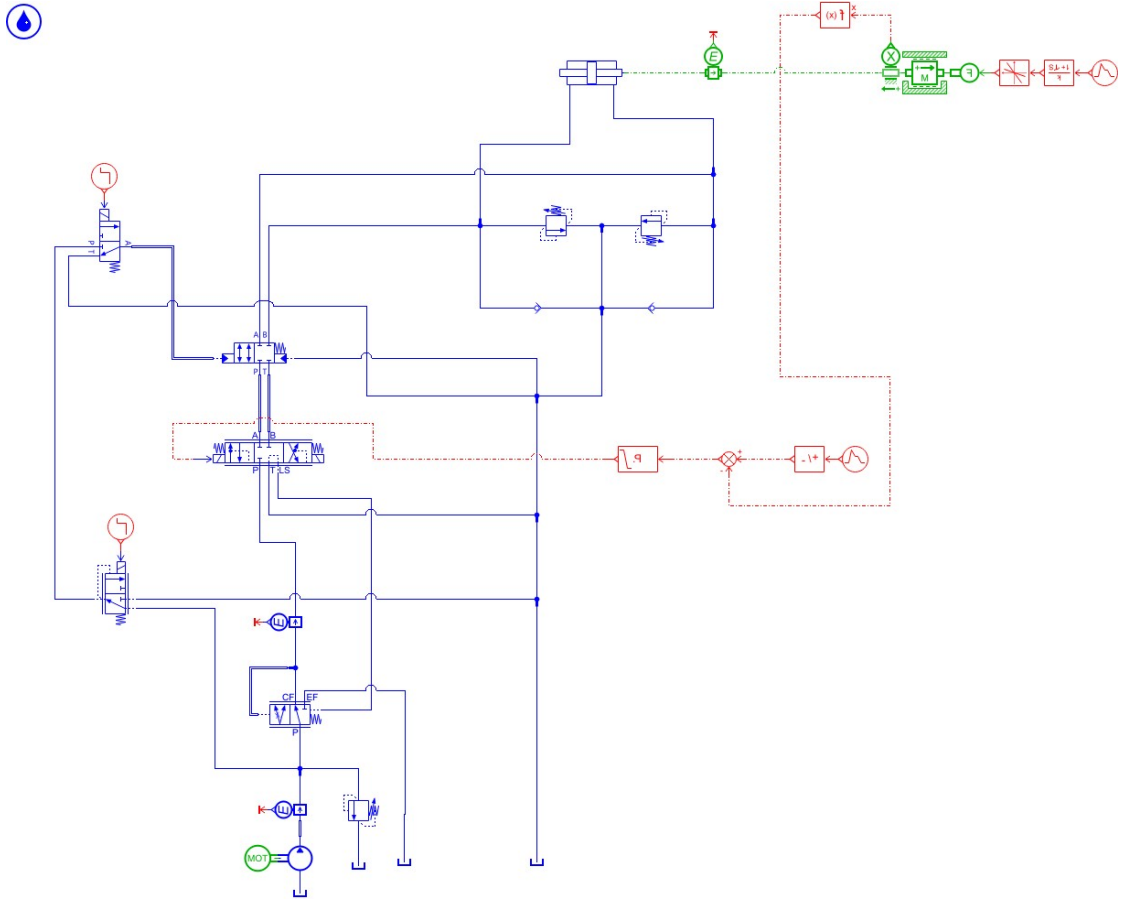


Figure 4.2: Steering system with Danfoss valve

Starting from the power supply unit, the system presents a fixed displacement pump driven by a fixed speed prime mover to ensure a constant flow rate. In addition to that, a priority valve is implemented in the line that connects the flow generation unit to the steering unit, thus the same pump can also feed auxiliaries when the flow requested by the steer is not the maximum one. Knowing this, the design of the fixed displacement pump should be done ensuring at least the maximum flow rate that the steering actuator can request, to evaluate this rate the dimensions of the cylinder reported in Table 1 are considered. Also, the time to move the piston from one end-stop to the opposite is kept 5 seconds as for the previous cases, thus resulting in a maximum flow request of 45 L/min. Considering that the prime mover speed is set at 1500 rpm, so to have a value neither too low nor too high thus avoiding low efficiency areas, the pump displacement should be at least of 30 cc/rev to provide always the maximum rate to the circuit. These values

are consistent with the ones used in the previous cases, thus a fair comparison between the different autonomous steering solutions can be done.

The priority valve can regulate the flow rate through the steering system, maintaining a constant delta pressure between the pressure in the CF line and the Load Sensing (LS) pilot line connected to the proportional valve; the delta pressure is set at 10 bar by a pre-loaded spring. In this study the extra flow is deviated to the tank, but this rate can be used to feed auxiliaries thus increasing the efficiency of the system. In addition to the priority valve, there are some other valves that have the aim of ensuring the safety of the system, as is possible to see in Figure 4.3 there is a cut-off valve (COV) that can disconnect the EHPV from the actuator. This COV receives two pilot pressure signals to regulate, the right side is connected to the tank thus to 0 bar (or atmospheric pressure) in case of correct operations of the system. The pressure on the left side of the COV is modulated through the two EH valves, so during normal operations the valve senses the pressure in the power supply line thus exceeding the force applied by the spring. Instead, when the electro-hydraulic steering should be switched off, the left side of the cut-off valve is connected to the tank pressure thus the spring pushes the valve to close it.

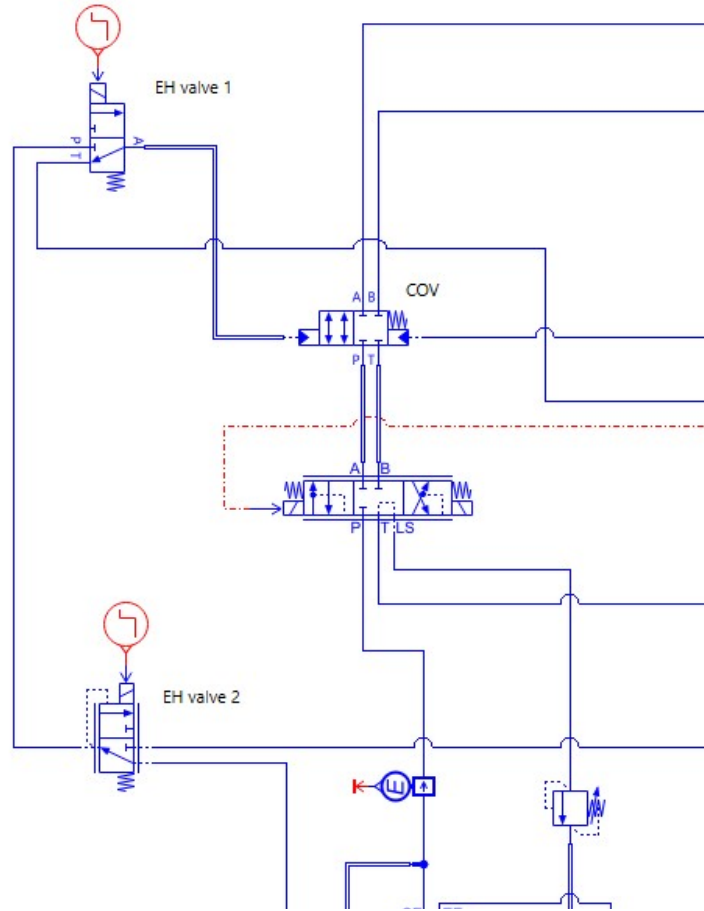


Figure 4.3: Simplified EH valves architecture

The proportional valve is a three position five port valve with closed center, the main difference with the EHPV seen in the first chapter is that a load sensing port is added. A pilot line is used to connect the high pressure side to the priority valve so that it can sense the load of the steering system thus modulating the flow rate towards it. The electric input to this EHPV is given by the controller and the maximum opening corresponds to a current of 40 mA. Moreover, the design of the opening areas and the pressure drop are simplified hypothesizing the valve almost ideal, thus the pressure drop between the ports is set at 0.1 bar for all the connections while the flow rate at maximum opening is conceived equal for all the ports and is 5 L/min.

Concerning the modulation of the proportional valve, a Proportional controller (P) is used to provide the electric signal to the solenoids. As it is possible to see in Figure 4.4, the controller takes the output of a linear sensor attached to the

actuator and compares the measured piston position with the one imposed by the steering input. The P controller is characterized only by one gain factor thus the design of this parameter can be done easily with a trial-and-error technique; through this method the proportional gain value result equal to 10. With this value the error of the system is minimized and the speed of the response of the steering system to the input is enough to be implemented in real life applications.

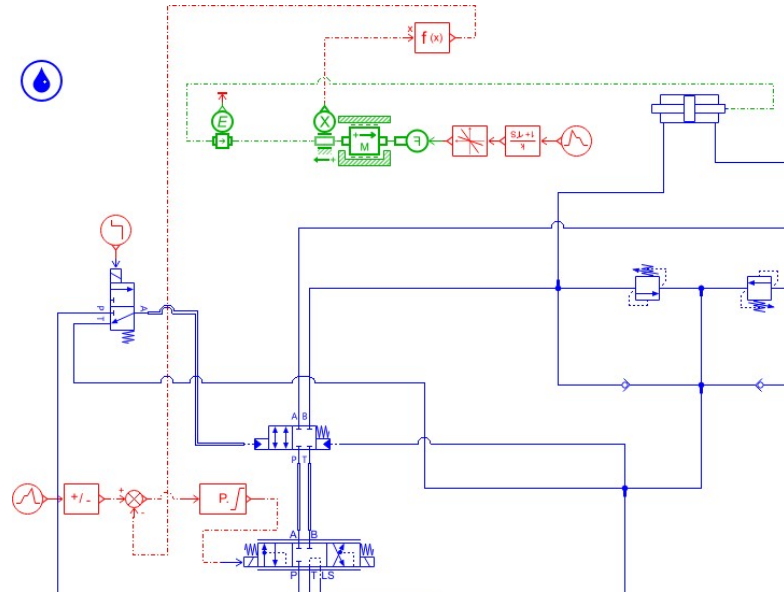


Figure 4.4: Controller for Danfoss solution

Safety is ensured with the combination of the two EH valves; indeed, if the ECU senses any unintentional movement of the EHPV the steering actuator is disconnected from the proportional valve. Furthermore, two pressure relief valves, combined with two pressure control valves, are implemented on the lines that enter the steering cylinder ports; this set of valves is reported in Figure 4.5. In this way the maximum pressure of the system is imposed at 150 bar, if the pressure exceeds this value the pressure relief valve discharge flow to the tank. Instead, the pressure control valve is used to prevent the phenomenon of cavitation, if the pressure in the circuit goes to low this valve opens the passage with the tank and a flow rate goes back from the tank to the system.

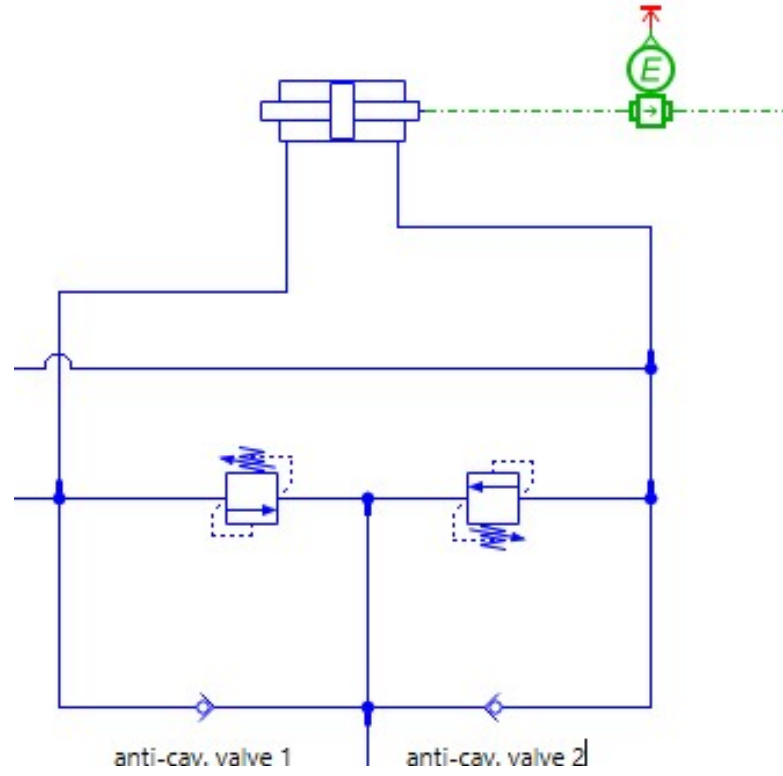


Figure 4.5: Safety valves architecture

4.3 Simulation setup

To test how this autonomous steering system behaves, a steering input should be given to it. Aiming to make a comparison with the previous systems the boundaries must be the same, thus the input seen for the previous chapter will be used also here. In Figure 1.7 is shown the time evolution of the steer, it is characterized by a first turn to the left and a second one to the right, exploiting in both cases a first segment where the angle increases followed by 2 seconds where the steering angle is kept constant. Considering that the steering actuator has the same dimensions for all four cases, this input can test the system with quite good results requesting a flow rate that reaches almost 75% of the maximum for which the system is designed. In Figure 4.6 is reported the time evolution of the flow rate to the actuator, from that it is possible to see that for all 3 segments in which the steering input changes the flow rate is 30 L/min, indeed in all these sections the inclination of the steering angle curve is equal, resulting in same request of steering speed.

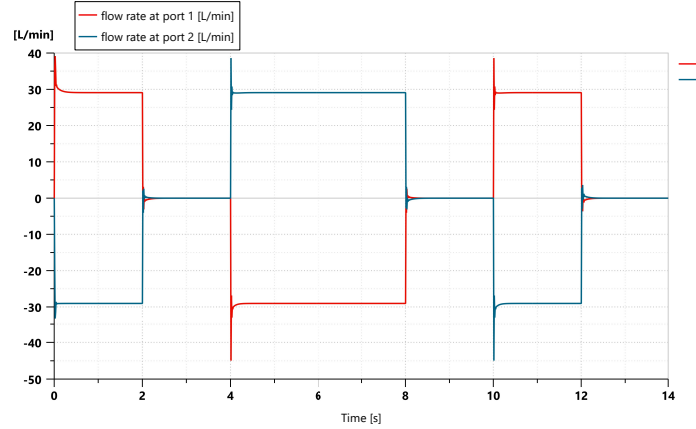


Figure 4.6: Flow rate requested by the steering cylinder

In addition to that, also the external conditions should be the same as in previous analysis in order to make a final comparison with all the solutions found. More specifically, the interaction between the tire and the ground is modeled in the same way as in the other cases, hypothesizing a lateral slip coefficient of 0.2, the vehicle weight of 25 tons and low speed. The same simplification will be taken into account: considering the very low vehicle speed only the lateral forces will be considered thus avoiding the longitudinal dynamics, the slip coefficient is kept constant to simplify the evaluation of these lateral forces. To give a more general view of how the system reacts and behaves during the operations three case studies will be considered: in the first one a resistive load is imposed, Figure 1.8, the second is characterized by a driving load, Figure 1.9, and the last will simulate a mixed load, Figure 1.10. Also in this case a first order lag is implemented to smoothen the lateral force curve thus enhancing the capacity of the hydraulic system to respond to the fast changes of external conditions; to achieve this result the time constant is set at 0.1 s while the “k” gain is 1.

4.4 Performance analysis

Going through the performance analysis is important to refer all the results obtained to the significant simplification made for this simulation; starting from the simplified

interaction between tire and ground, the vehicle dynamics that are not considered and the almost ideal components of the Amesim model.

The analysis of the results should start checking the accuracy of the autonomous steering system. Actually, the solution analyzed in this chapter, developed by Danfoss [3], is already implemented in purely steer-by-wire vehicles, but the autonomous steer feature should be tested. To do that, the output of a linear position sensor attached to the steering actuator is compared with the steering input. That sensor measures the position of the piston inside the cylinder, this value can be linked to the steering angle through the Formula 1.4. From the comparison between the desired and actual position of the piston, it is possible to calculate the error and so the accuracy of the steering system is evaluated. In Figure 4.7 is reported the actual piston displacement in meters, the simulation starts with the piston positioned in the middle of the steering cylinder thus having straight wheels. Then two identical steers are operated in both directions to test the system and forcing the piston to cover most of the actuator displacement, circa 65% of the total stroke.

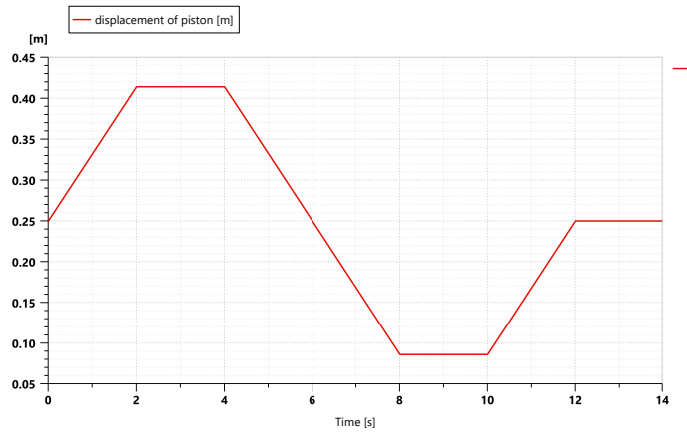


Figure 4.7: Piston displacement with Danfoss solution

The percentage error evaluated this way is on the order of 1%, this means that the autonomous steering system analyzed can ensure high accuracy for the imposed steering input. This result does not change by switching between the different load scenarios, enhancing the reliability of the system. This is a consequence of the type of controller used for the management of the steering system, indeed the closed-loop configuration allows to act on the piston position by correcting it almost in real

time. The implementation and design of such a control system is fundamental to implement the autonomous steer feature in real word applications; although the proportional controller designed for this case study is a simple solution that could be further upgraded. To inspect the overall behavior of the steering system is important to check also the pressure evolution during the simulation. Indeed, the maximum pressure should not reach a value too high that can affect the safety of the system; moreover, the pressure difference between the two cambers of the steering actuator should remain in the order of 40 bar. This is due to the fact that the boundaries are kept the same for the different case studies, in this way a fair comparison between the different solutions can be done. Starting with the resistive load case scenario, reported in Figure 4.8, the maximum pressure reached in the actuator is 53 bar that is a reasonable value considering also the previous cases. The back-pressure is 11 bar, thus leading to a delta pressure of 42 bar consistent with the one hypothesized and compliant with the cases analyzed in the previous chapters.

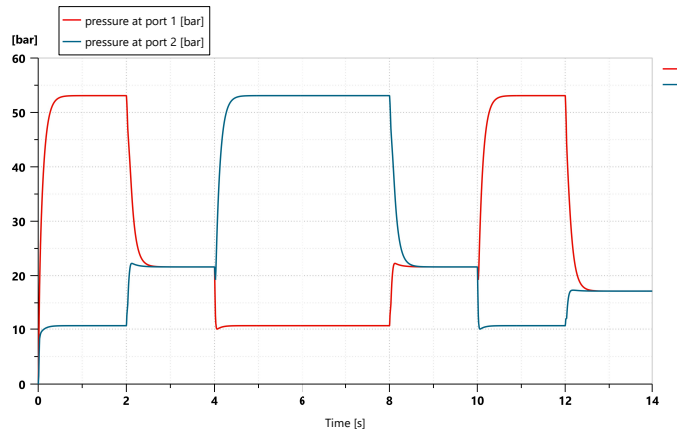


Figure 4.8: Pressure inside the actuator with resistive load

Passing from the resistive load case to the driving load scenario the pressure is much lower for the whole simulation but the overall behavior does not change. Looking at Figure 4.9 it is possible to see that the back-pressure reaches values close to 2 bar, thus ensuring a minimum pressure to guaranty the controllability of the steering system.

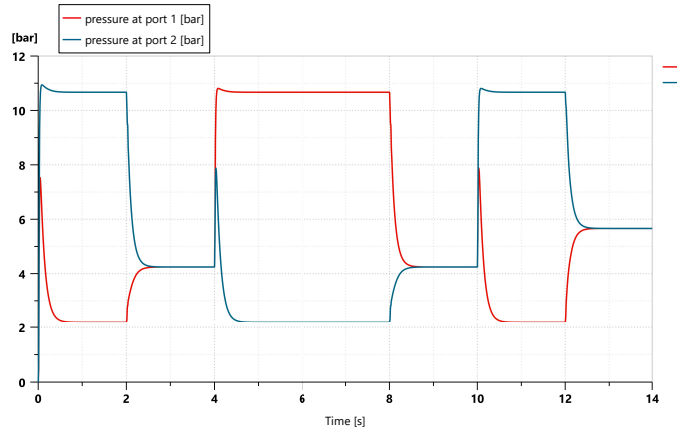


Figure 4.9: Pressure inside the actuator with driving load

Instead, changing from the driving load to combined load case things change. As shown in Figure 4.10, the maximum pressure is 51 bar and is reached in the first 2 seconds of the simulation implying a pure resistive load in that section. In this phase also the desired delta pressure is respected, indeed the back-pressure is 10 bar similarly to the one found in the pure resistive load scenario. Going further, in between 5 and 8 seconds of simulation the back-pressure drops to 0 bar thus decreasing the level of control of the system during this period. This behavior depends on the the external force applied to the steering system, during these 3 seconds the load is driving the actuator. To face this challenging scenario, while the pressure in the chamber is at 0 bar, the anti-cavitation valves operate by opening the passage between the tank and the actuator port. This way, as is possible to see in Figure 4.11, a certain amount of flow is allowed to enter the chamber, thus avoiding the pressure from being under 0 bar and preventing cavitation phenomenon.

To conclude this last analysis the energy efficiency of the steering system is computed to have complete overview of the behavior of the different solutions implemented. As described in the previous chapters, both the useful and the expended energy must be evaluated in order to evaluate the total energy consumption through the Formula 1.6. Even in this case the scenario chosen for the energy analysis is the resistive load one, this way the analysis is simplified and the behavior of the system is more easily understandable. Considering that, the energy spent to drive the pump is 38.4 kJ that is one of the highest value found during this study. Indeed,

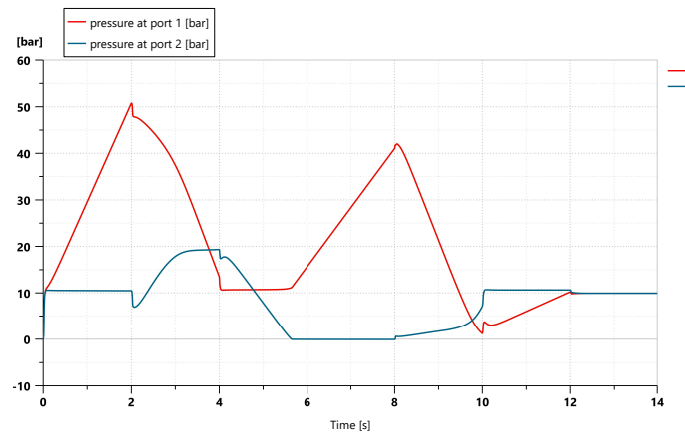


Figure 4.10: Pressure inside the actuator with combined load

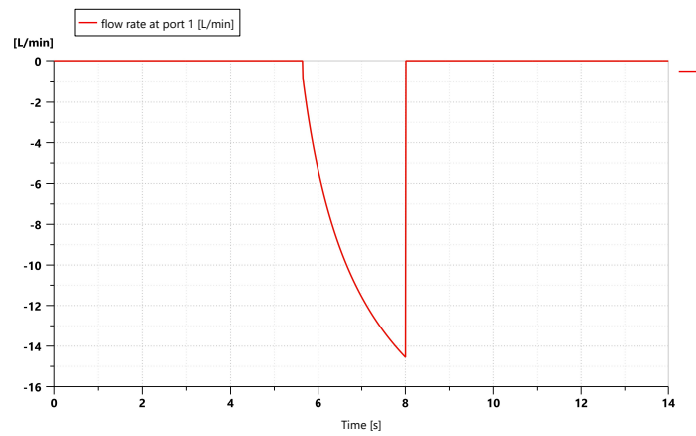


Figure 4.11: Flow rate through anti-cavitation valves

the energy used by the autonomous steering system is 24.5 kJ, this value is lower with respect to the energy spent by the power supply unit due to the priority valve. In fact, during the whole simulation the flow rate requested by the steering system is lower than the flow rate generated by the fixed displacement pump, the extra-rate is managed by the priority valve. As is possible to see in Figure 4.12, the priority valve splits the flow rate between the two ports CF and EF, the first one is connected to the steering system and has the priority; instead, the second one can be connected to auxiliaries thus increasing the total efficiency. To simplify the model in this case the EF port is connected to the tank, thus the flow rate through this port is discharged generating significant loss of energy.

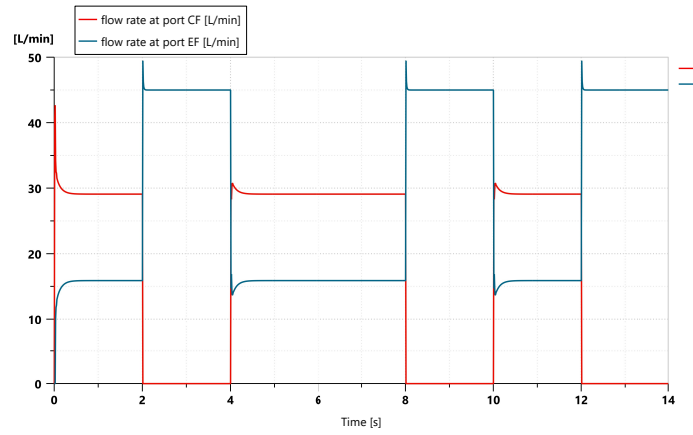


Figure 4.12: Flow rate through the priority valve

Concerning the useful energy, it is possible to assert that is aligned with the values found during the previous analysis, in fact it accounts for 15.8 kJ. With these values the total energy efficiency of the system is 63.8%, this value is consistent with the type of system modeled. However, the total efficiency of the system can be further improved by adding to the model one or more auxiliary hydraulic systems that could use the extra-flow rate deviated by the priority valve. To conclude, this autonomous electro-hydraulic power steering system is able to ensure high level of accuracy and reliability, facing with low total energy efficiency that could be easily increased with some attentions on real-word in vehicle implementation.

Capitolo 5

Conclusion

The aim of this thesis project is to make a review and a preliminary model of the main electro-hydraulic steering architectures for off-road vehicles, inspecting the possibility to implement autonomous driving features in such systems. From the literature analysis is highlighted that electro-hydraulic solutions are increasingly investigated to improve the steering performances, enhance system flexibility and reliability and enable the implementation of advanced control functions.

From the reviewed solutions, the most interesting are chosen and a preliminary dynamic model is developed in Amesim. The four architectures modeled are steer-by-wire systems characterized by different approach concerning the flow rate management. The first and the last models have a proportional valve to control the flow rate to the actuator, but the last model differ from the first one by the purely steer-by-wire nature of the system; instead the second solution modeled is characterized by a variable speed motor to drive the pump together with four on-off valves. The third architecture modeled covers another type of steering system where the pump displacement is variable and no valves are needed to manage the flow rate to the steering cylinder; both the second and the third models rely on steer-by-wire architectures.

Through the simulation of the models and the analysis of parameters such as the pressure and the flow rate, it is possible to understand the behavior of the different electro-hydraulic steering systems. Due to the simplification made, especially on boundaries conditions like the interaction between tire and ground and vehicle dynamics, only a preliminary check of the safety and accuracy of these systems can be made. However, by comparing the results obtained, the solution with the variable speed prime mover and the one characterized by the variable displacement pump are giving significant benefits in terms of energy efficiency, making the analyzed solutions interesting. Moreover, by adding Proportional Integral Derivative (PID)

controller to the electro-hydraulic steering systems, a first investigation of the integration of autonomous driving features in off-road vehicles is made.

Although both the models and the simulations setup are limited by simplified assumptions related to the preliminary nature of this work, the results obtained could be useful basis for future developments. Advanced studies may include the integration of more detailed models for the vehicle dynamics and the implementation of more sophisticated control algorithms, thus allowing a more accurate comprehension of the behavior of these electro-hydraulic steering systems and enhancing the reliability of autonomous driving features in off-road vehicles applications.

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