Master’s Dissertation

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Energy Management Strategy for a Hydraulic Hybrid Wheel Loader

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Abstract

In recent years, aligned with the concern over climate change many companies have started using hybrid technology due to its increasing potential in driveline's efficiency and reduction in fuel consumption. This paper deals with the design of an energy management strategy, ECMS, for a mid-size wheel loader equipped with a hydraulic hybrid system since previous research indicates that hydraulic hybrid system is promising for use in heavy-duty and construction machinery. This study employed a qualitative research approach via secondary data collection. Data was gathered from various scientific articles and academic books. The sources were selected based on their relevance to the research topic, and their contribution to the existing body of knowledge. A systematic review of the literature was carried out, focusing on recent publications and foundational books in order to ensure a comprehensive understanding of the subject. The literature was then analyzed to identify the prevailing arguments and the gaps in the current research. These findings shaped the basis for the discussion and helped conduct the research conclusions. A complete hydraulic hybrid wheel loader simulation model and a control strategy based on Equivalent Consumption Minimization Strategy (ECMS) have been developed for the evaluation of hybrid technology. The result indicates that the technology is able to reduce fuel consumption by about 8% for the construction machine which operates in a repetitive drive cycle demanding high transient power. These findings prove that once more the hybrid system is an effective manner to reduce fuel consumption and consequently engine emissions.

Contents

Nomenclature

iPm	Pump/motor gear ratio	[-]
Dpm	Pump/motor displacement	[cc/rev]
Fwheel	Traction force	[KN]
mFuel	Fuel rate	[l/min]
nIce	Engine speed	[rad/s]
nImp	Impeller speed	[rad/s]
nProp	Propeller shaft speed	[RPS]
nProp_rad	Propeller shaft speed	[rad/s]
nTurb	Turbine speed	[rad/s]
P	system pressure	[bar]
PLoss	Accumulator pressure loss	[bar]
P0	Pre-charge pressure	[bar]
Pmax	Maximum pressure	[bar]
Qloss	Pump/motor flow loss	[l/min]
Qpm	Pump/motor flow	[l/min]
t	time	[s]
Tloss	Pump/motor torque loss	[Nm]
Timp	Impeller torque	[Nm]
Tpm	Pump/motor torque	[Nm]
Tprop	Propeller shaft torque	[Nm]
Ttra	Transmission output torque	[Nm]
Tturb	Turbine torque	[Nm]
TWH	Work hydraulics torque	[Nm]
u	Control signal	[-]
Vwheel	Vehicle speed	[Km/h]
X	System state	[-]
epsilonPM	Pump/motor relative displacement	[-]

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1. Introduction

Negative environmental impacts of fossil fuels steer automotive companies towards highly efficient vehicles with low fuel consumption. Therefore, new technologies such as alternative fuels, hybridization and electric vehicles have been investigated by automotive industries [1]. However, in heavy-duty vehicles such as wheel loaders different types of hybrid technology which is hydraulic hybrid systems are more favorable since the driving cycles are normally repetitive and high transient power is needed [2]. One previous article indicates the advantages in terms of fuel efficiency of a hydraulic hybrid wheel loader [3]. This project has investigated the fuel consumption reduction of a hydraulic hybrid wheel loader designing an ECMS controller through controlling a variable displacement hydraulic pump. Fuel consumption reduction can be achieved because the control algorithm utilizes an alternative path during the driving cycle when the secondary power source, a hydraulic accumulator, is charged, while the internal combustion engine operates only to supply the excess power required or remains idle.

Moreover, vehicles with internal combustion engines require a transmission system to transfer torque and speed from the engine to the wheels. The transmission systems can be automatic or manual transmission. In wheel loaders because of high traction forces and repetitive driving cycles the automatic transmissions are commonly used to avoid too much clutch wear in the case of manual transmission. Automatic transmissions take advantage of a torque converter which is the big contributor of power losses in drivelines during slip operations [4]. When high traction forces are needed, for example during bucket filling phases, high slip occurs in torque converters. The expectation is that by introduction of the hybrid system and control algorithm the power losses reduced since another power path is available [5].

1.1 Objectives

The purpose of this dissertation is to enhance the fuel efficiency of a wheel loader while maintaining the operability acceptable. The focus of this work includes the development of an energy management strategy which controls the displacement of a variable displacement hydraulic pump/motor connected to the secondary power source which is a hydraulic accumulator. The controller is evaluated on a simulation model of the wheel loader developed in a simulation environment using a backward-facing simulation approach.

1.2 Delimitations

This project is confined to evaluating the performance of the hydraulic hybrid wheel loader on two typical wheel loader driving cycles including the short loading cycle and the load-carry cycle using the bucket to transport the material. However, due to the lack of experimental data the output power of the work hydraulic pump which controls the bucket tool has not been considered in this work; this data can be fed to the model, if available, without requiring any modifications to the EMS algorithm.

1.3 Thesis outline

The following chapter introduces essential theory to equip readers with the necessary background needed to understand the topics explored in this thesis. The theory contains some information about wheel loaders, one of the typical simulations driving cycle for this type of vehicles, wheel loader's powertrain components and hydraulic elements plus hybrid systems. Chapter three focuses on hydraulic hybrid technology while the following chapter is dedicated to modelling and simulation. The fifth chapter introduces the energy management strategy concept followed by the sixth chapter that explains the ECMS control strategy specifically. The simulation outcomes are explained in chapter seven with the conclusions following, and future works in chapter eight and nine respectively.

2. Theoretical backgrounds

2.1 Wheel loaders

Wheel loaders are heavy machinery commonly used on construction and building sites. There are different types of wheel loaders named front-end loaders, front loaders, wheeled loaders, bucket loaders, scoop loaders, and skip loaders [6]. Wheel loaders use an arm to lift and lower its bucket [6]. Large wheel loaders are often four-wheel drives and exploit the counterweight principle to be able to carry the materials. These machines can dig, carry, and transport. Apart from construction sites they are also used for farming operations and land development projects [6]. One of the main advantages of wheel loaders is their adaptability with various tools replaced with the bucket such as forks, forklift attachment, clamshell bucket and snowplow [5]. The figure below shows some of these equipment.

Figure 2.1: Example of different tools that can be mounted on wheel loaders for the different purposes. Picture source: [7]

This paper considers a mid-size wheel loader equipped with a bucket to fill, carry and unload gravel from a gravel pile to a load receiver site. The loading cycle is presented in the next section 2.2. The wheel loader size and the counterweights mass are the crucial factors in the amount of load that the machine can carry [8]. Engine, chassis, work functions are defined based on the load capacity that the wheel loader is specified for. In addition, one previous study proves that the characteristic of the pile has a direct impact on machine productivity. The smaller and uniform particle size is more favorable from machine productivity point of view [9].

2.2 Drive cycle

The drive cycle used for the simulation in this paper is based on the two typical wheel loader operating cycles including short loading cycle and load-carry cycle. In the short loading cycle, that sometimes-called V-cycle or Y-cycle, the wheel loader approaches the gravel pile, fills the bucket, then reverses, reaches to the load receiver point and unload the bucket and then reverses back into the start position [10]. The phases are shown in figure 2.2.

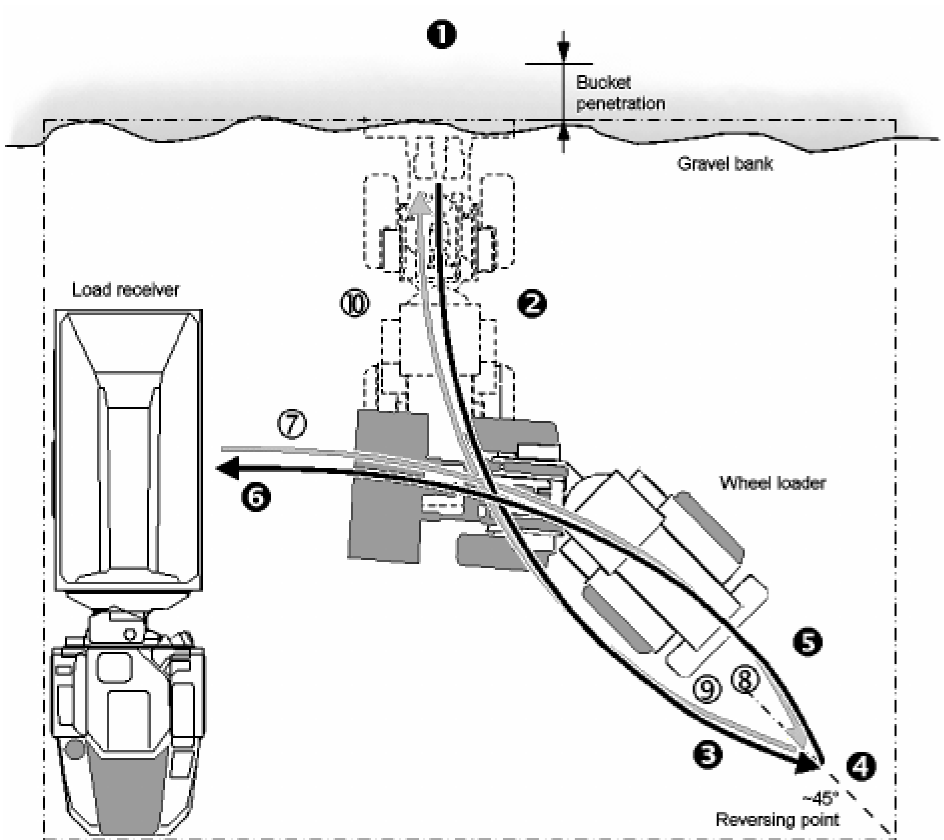

Figure 2.2: Short loading cycle

In the load-carry cycle, that sometimes-called long loading cycle, the load receiver is positioned at a farther distance thus longer transportation phase up to 400m in forward gear is required [10]. Figure 2.3.

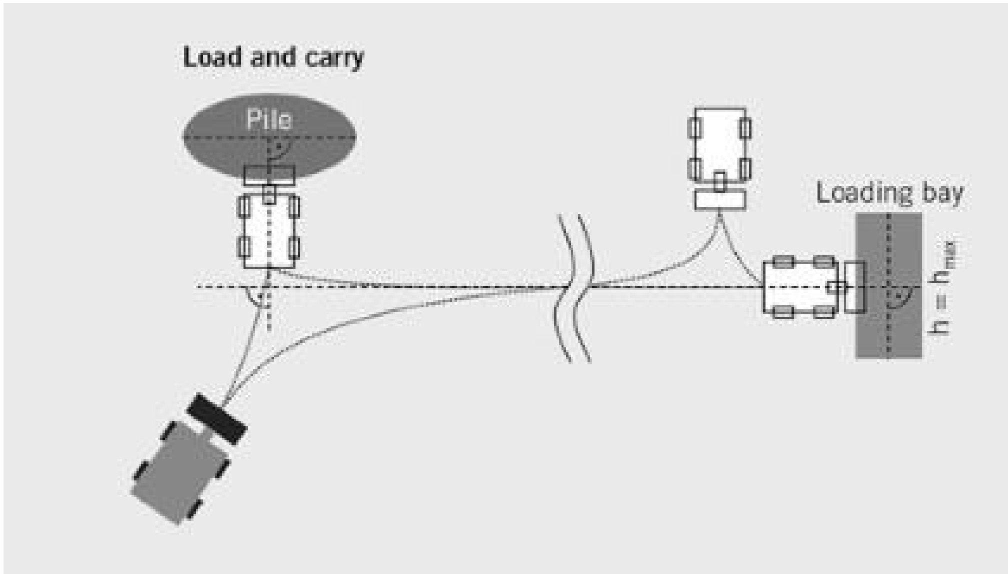

Figure 2.3: load-carry cycle

2.3 Powertrain elements

A schematic picture of wheel loader's powertrain is shown in the next figure.
Primary subsystems of a conventional wheel loaders include:

- An Internal combustion engine that transforms chemical energy in fossil fuel into mechanical energy. The volume and the number of cylinders of the engine can be different based on the output power specified.
- Torque converter (TC): TCs are a substitutional of clutch systems commonly used in cars with manual transmission, but TCs used in wheel loaders mainly due to its torque multiplication ability which is essential for giving the wheel loader additional force at low speeds.

  Moreover, TCs have more advantages in terms of:
  - Smooth power transmission: TCs enable smooth transmission of power between engine and the drivetrain.
  - Load flexibility: TCs allow the engine to operate at optimal speed while automatically adjusting the power delivery to the wheels.

- Power damping: TCs are able to damp the fluctuations in a rugged environment.
- High efficiency in low speeds operations: Wheel loaders frequently operate at low speeds during tasks like digging and material handling. TCS enables efficient operation at these low speeds.

- Transmission system with different gear ratios
- Final drive that magnifies torques

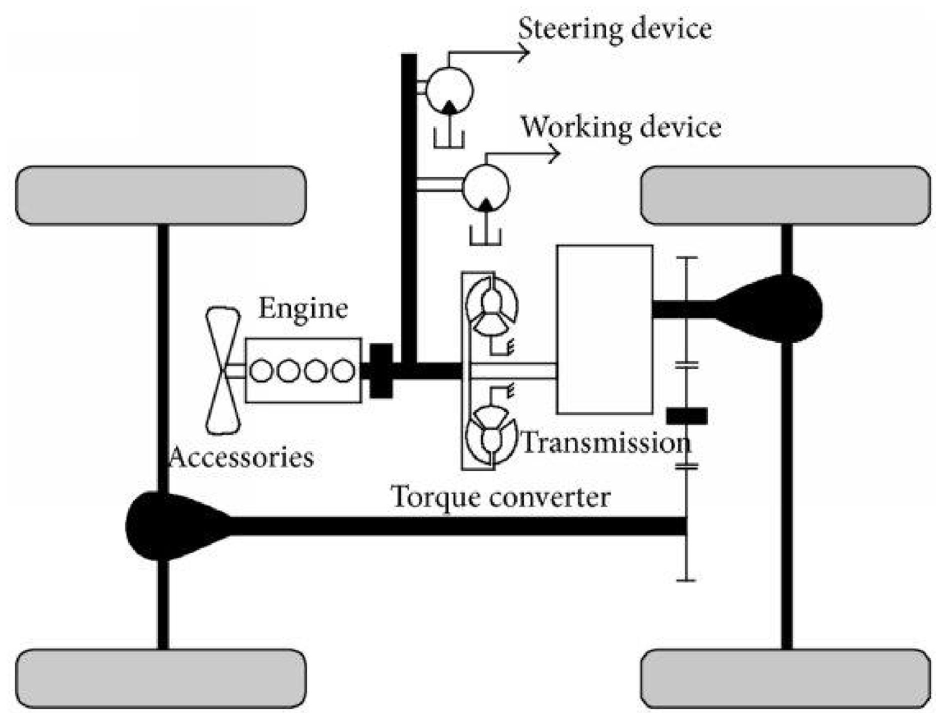

Figure 2.4: A typical wheel loader drivetrain illustration

2.3.1 Torque Converter

An internal combustion engine is connected to a manual transmission system by a friction clutch. The friction clutch enables the vehicle to stop without switching off the engine. However, in automatic transmission vehicles another coupling system which is torque converter (TC) is utilized. The torque converter is a type of fluid coupling device used to transfer rotational power from the engine to the transmission system. The two main properties of a torque converter are torque amplification and decoupling the internal combustion engine from the rest of the driveline. The torque converter is made of 3 key components that are immersed in the transmission oil which is the medium to transfer the power. The three components are [23]:

- Impeller (pump): the impeller is attached to the engine's output shaft, and it spins with engine's crankshaft speed. The transmission oil is forced outward toward the turbine by centrifugal force, moving away from the center.
- Turbine: the turbine is coupled with the transmission input shaft and receives oil from the pump. The turbine has curved blades designed to catch the fast-moving fluid and as the fluid hits the turbine blades, it exerts force on them and causes the turbine to rotate. However, the turbine normally spins slower than the impeller due to the losses.
- Stator: the stator is located between the impeller and turbine. The stator redirect fluid returning from the turbine back to the impeller. The main purposes of the stator are changing the direction of the fluid to the same rotational direction of the impeller and reducing the fluid velocity. Therefore, efficiency will increase since the losses will be reduced. A graphical illustration of a TC is presented in figure 2.5.

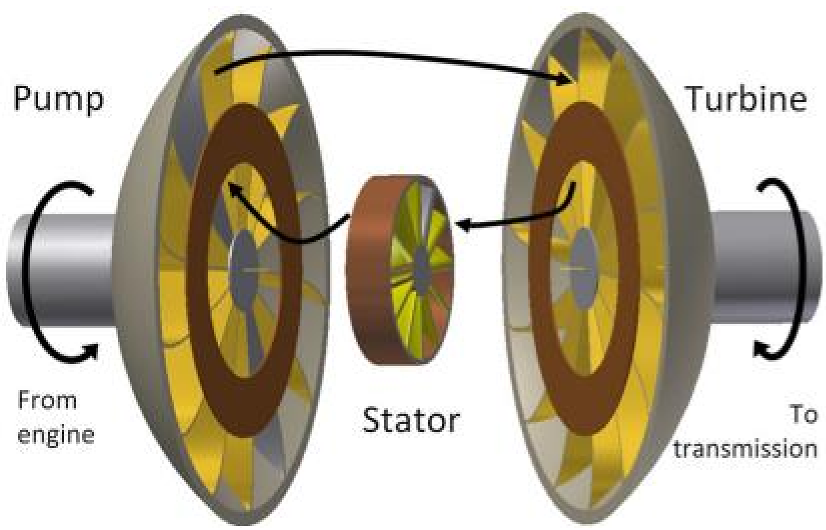

Figure 2.5: A schematic picture of a typical torque converter. Figure source: [22]

The torque converter operates in three distinct stages: Stall, acceleration and coupling. At stall the vehicle is stationary with the engine running, such as when idling in gear with brakes applied, thus in the stall stage, the engine drives the impeller, but the turbine remains stationary since the vehicle is not moving. Maximum torque multiplication occurs in this stage because the stator redirects the fluid, providing extra force to help the turbine move when the vehicle starts. This is particularly useful when accelerating from a complete stop, providing greater torque to the wheels. At acceleration (torque multiplication stage) the vehicle begins accelerating from a low speed. In this stage the turbine starts to rotate but at a lower speed than the impeller. This difference in speed between the impeller and turbine allows torque multiplication, which provides additional power for acceleration. However, the torque multiplication gradually decreases as the turbine's speed approaches the impeller's speed. Coupling stage happens when the vehicle reaches a steady driving speed, and the engine and transmission speeds are nearly synchronized. i.e. in coupling stage, the turbine and impeller operate with nearly identical rotational speeds. In addition, no torque multiplication occurs in this stage. In fact, many torque converters engage a lock-up clutch at this stage to create a direct connection between the engine and transmission, minimizing slippage and improving fuel efficiency by effectively locking the turbine to the impeller [23].

The two important equations which are slip ratio and torque ratio can be obtained by equations 2.1 and 2.2:

$$\nu = \frac{\omega_{out}}{\omega_{in}} \tag{2.1}$$

$$\mu = \frac{T_{out}}{T_{in}} \tag{2.2}$$

Where ω_{out} is the turbine speed, ω_{in} is the impeller speed, T_{out} is the turbine torque and T_{in} is the impeller torque. ν is a positive value between zero and one.

The efficiency of torque converters can be calculated using equation 2.3:

$$\eta = \frac{T_{out}.\,\omega_{in}}{T_{in}.\,\omega_{in}} \tag{2.3}$$

Torque converters rely on fluid coupling rather than a mechanical connection, leading to some degree of slippage between impeller and turbine. This slippage reduces efficiency, since not all the engine's power is effectively transferred to the transmission, especially at lower speeds [24].

2.4 Hydraulic systems

A Hydraulic hybrid system consists of different hydraulic equipment such as hydraulic pump/motor, accumulator, and valves which are presented in this chapter.

2.4.1 Hydraulic control Valves

There are a lot of types of hydraulic valves that are widely used in hydraulic systems. The most common valves are explained briefly in this chapter. In figure 2.6 some of the most common valves are indicated.

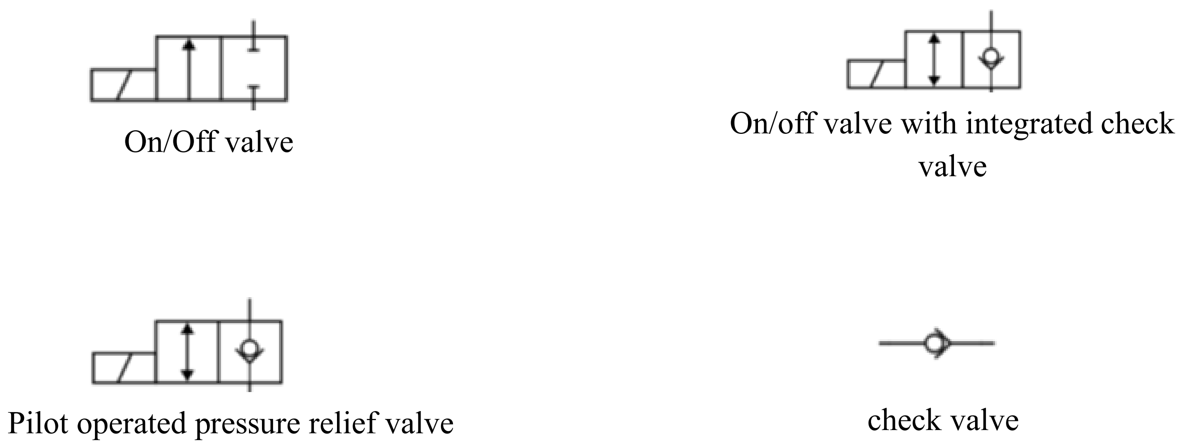

Figure 2.6: Four frequently used valve symbols in hydraulic systems.

on/off valves can be normally open (NO) or normally closed (NC), and they close or open when energized, but the proportional valves can be controlled continuously [5]. In figure 2.6 an on/off valve and an on/off valve with integrated relief valve are shown.

The *pressure relief valve (PRV)* is used as a safety valve in hydraulic circuits since it opens if the system pressure reaches a defined value adjusted by a pre-tension spring therefore it can cause a reduction in pressure of the system. The pilot-controlled pressure relief valves are one of the most common valves of this type. The pilot-controlled pressure relief valves take advantage of a small hole in the cone that houses a spring, designed to maintain the size of the spring small [13].

Check valves are one-directional valves in which a piston is pushed against its seat via a spring to seal the flow in the opposite direction. The pressure drops over the valves must be the lowest possible otherwise the losses in the system increase [13].

2.4.2 Accumulator

Accumulators in hydraulic circuits work as an energy storage system. They are equivalent to supercapacitors in an electric circuit since they accumulate and release energy rapidly that means high specific power. Nevertheless, they are characterized by low specific energy i.e. they require large volume for the storage of high amount of energy [13]. There are three main types of accumulators available [13]:

- weight-loaded accumulators
- spring-loaded accumulators
- gas-loaded accumulators

The gas-loaded accumulator is the most used in modern applications. In this type of accumulator, compressive energy is obtained by an inert gas such as nitrogen or argon. These gas-loaded accumulators are divided into three main architectures that are: piston, bladder, and diaphragm accumulators.

In the *piston type accumulator,* a piston is used to separate the oil from the inert gas. The piston moves as a function of oil's pressure and the compressing gas. The moving piston is equipped with an appropriate seal to guarantee the separation of the fluid from the gas [13]. The *diaphragm accumulator* uses a diaphragm composed of rubber or neoprene to create a barrier between the compressible gas and the oil. Nonetheless, due to the presence of elastomeric material in this type of accumulators, they have a limited life span; therefore, this is the reason that this type of accumulator is not suitable for heavy duty applications [11]. *The bladder type accumulator* is similar to the diaphragm type that uses a rubber bladder for the separation purpose. In figures 2.7 and 2.8 three examples of these accumulators are exhibited. In addition, as shown in figure 2.8 at the bottom part of the accumulator there is a properly designed valve which opens the port to the fluid side and prevents the bladder from extruding out of the accumulator [14].

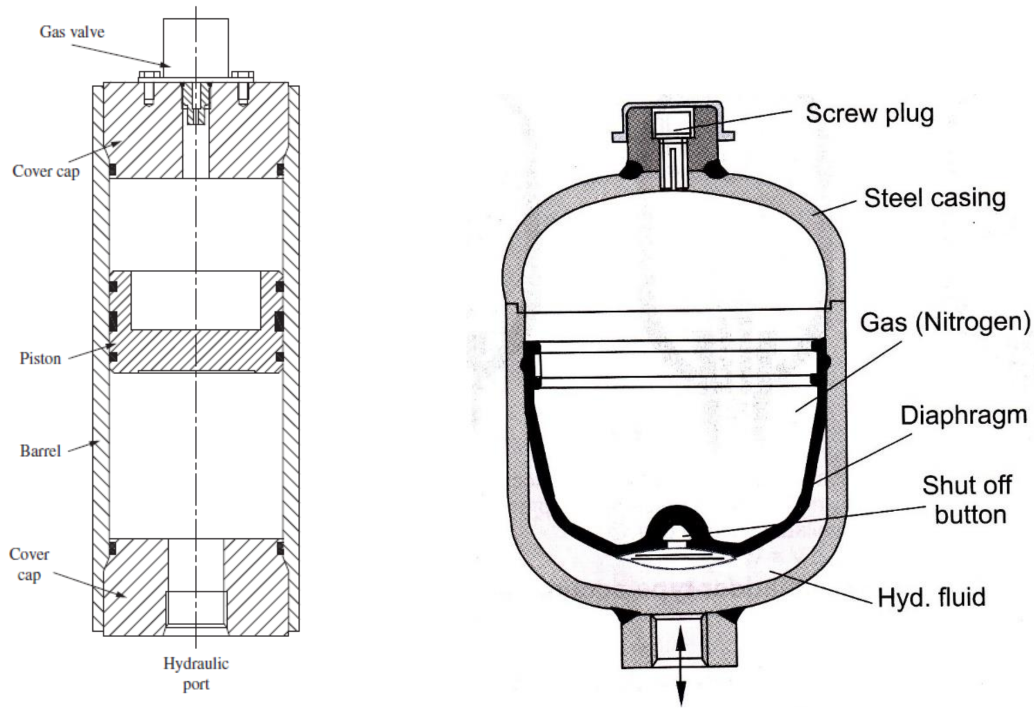

Figure 2.7: Two types of common hydraulic accumulators. The left picture is the piston type, and the right picture is the diaphragm type accumulator

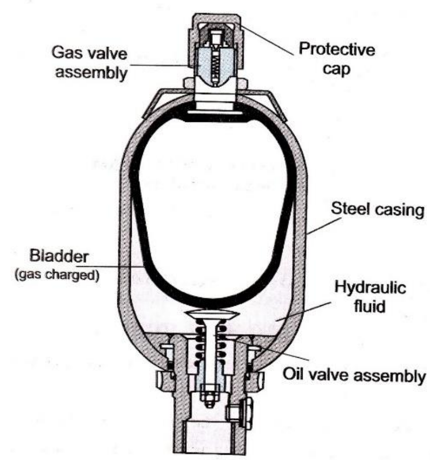

Figure 2.8:bladder type accumulator

The usage of accumulators in a hydrostatic circuit can brings different advantages for example, in case of flow fluctuation they help stabilize the system. Moreover, they provide shock cushioning effect since hydraulic systems are often subjected to pressure spikes which can have negative impacts on other components [14]. Additionally, thermodynamic processes influence the accumulator's pressure level due to the temperature's effect on the inert gas.

2.4.3 Hydraulic pump/motor

A Hydraulic system is normally driven by a hydraulic pump/motor coupled with a driving mechanism that can be an internal combustion engine or an electric motor. The hydraulic pump/motor can convert mechanical energy to hydraulic energy as a pump or can convert hydraulic to mechanical energy as a motor [11]. The hydraulic power can be approximated by two physical quantities that are pressure and volumetric flow rate. The flow provided by the pump is described by equation 2.4 and the torque demanded by the pump to maintain a certain pressure is defined by equation 2.5 [12]. Additionally, a hydraulic pump is designed to deliver flow rather than pressure; Pressure arises as a result of resistance to flow created by various consumers of that flow.

$$Q = \varepsilon D n \eta_{vol} \tag{2.4}$$

$$T = \varepsilon D \frac{\Delta p}{2\pi} \eta_{hm} \tag{2.5}$$

In the up mentioned equations, the quantities are: ε is the displacement setting of the hydraulic pump/motor, D is the displacement of the hydraulic pump/motor, n is rotational speed, η_{vol} is the volumetric efficiency, Δp is differential pressure, η_{hm} is the hydromechanical efficiency. Hydromechanical losses are because of the difference in pressure over the pump/motor and frictions in the machine that can come from different sources such as viscous friction in the lubricating interface while volumetric losses come from leakages in the machine and depend on the compressibility of the fluid [11]. Hydraulic pump/motor are divided into two main categories based on their work elements: Rotational, and piston machines. In rotational machines the work element is rotating whereas in piston machines the work element moves linearly. The other difference is in their sealing gaps; this explains why piston machines are better for high speed and high-pressure applications since these types of hydraulic machines have smaller and fewer sealing gaps. On the other hand, rotational machines are characterized by larger and more sealing gaps [11]. The most widespread hydraulic machines include [11]:

- Axial piston machines
    - Swashplate type axial piston machines
    - Bent axis type axial piston machines
- Radial piston machines
- Gear machines
    - External gear machines
    - Internal gear machines
    - Gerotors
- Vane-type machines

Gear and Vane-type machines are rotational machines while axial and radial machines are piston machines. Two types of hydraulic pump/motor are explained below.

In a *bent axis configuration,* the drive shaft drives the cylinder block of the pump/motor that houses several pistons. The reciprocating motion of the pistons causes displacement volume to change. As the cylinder block turns, each piston chamber alternately connects to the inlet or outlet port by a valve plate. Figure 2.9. The rotation of the shaft leads to reciprocating movements of the pistons which are connected to the driving flange via ball joints which allow Pistons in the bent axis type machine to tilt from the bore axis. Various displacement can be achieved when the Swivel angle is changed by the motion of the cylinder block and the valve plate.

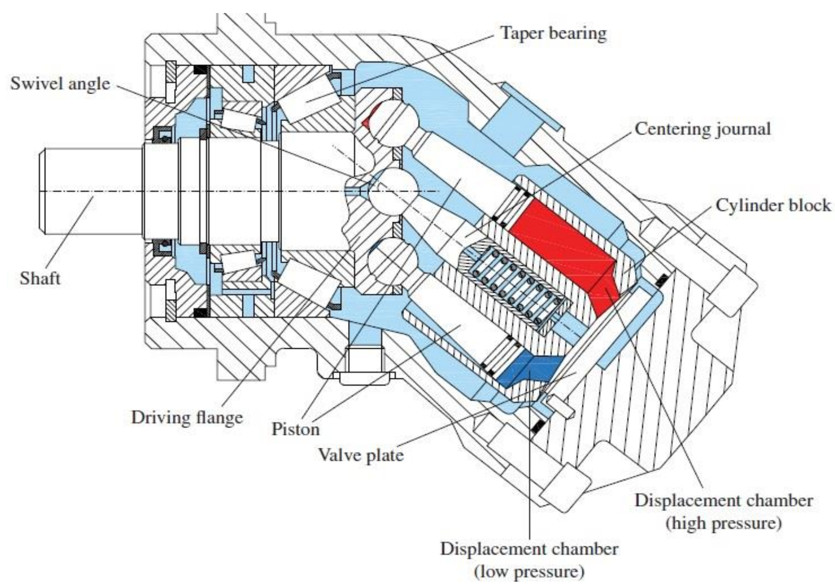

Figure 2.9: Bent axis-type axial piston machine. Figure source: [11].

In *vane-type* machines as is illustrated in figure 2.10 the vanes are positioned in the rotor and have radial movements. The vanes are pushed against the cam ring by springs. A displacement chamber is formed in the space between the cam ring, the rotor and the two neighboring vanes. In addition, the shape of the cam ring is responsible for the various displacement volume thus with a multiple displacement cam ring a variable displacement hydraulic machine is achieved. Each displacement chamber can displace the oil from inlet to the outlet twice per one revolution of the drive shaft. The vane-type machines are characterized by low flow pulsation, low noise and high mechanical efficiency since no side load is transmitted to the shaft but on the other side their maximum working pressure is not high (280 bar) and they are very susceptible to cavitation [11]. However, in a single stator vane machine the displacement chamber can be modified by the change in the eccentricity of the rotor with respect to the stator [11]. A single stator vane machine is indicated in figure 2.11.

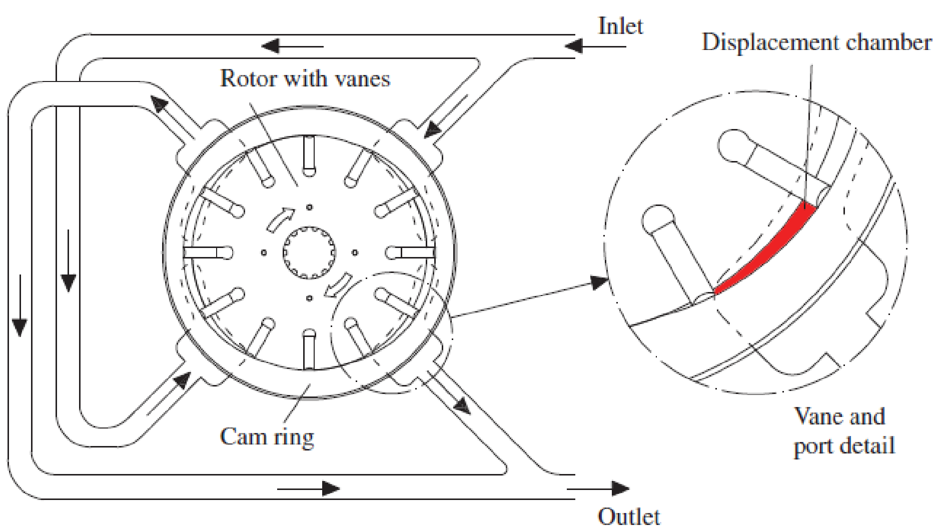

Figure 2.10: Vane- type hydraulic machine

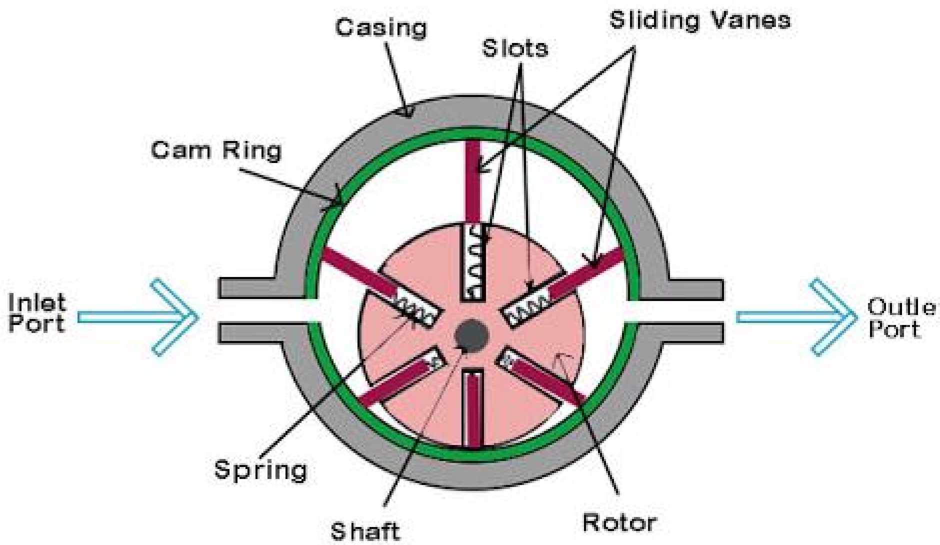

Figure 2.11: single stator vane machine

2.5 Hybrid powertrains

The hybrid vehicle means combining two or more sources of power that can directly or indirectly provide propulsion. The primary source of energy typically comes from chemical energy stored in the fossil fuels used by internal combustion engines, and depending on the secondary source, hybrid can mean hybrid electric (HEV) in case of the presence of electric motor, fuel cell hybrid and hydraulic hybrid. The secondary energy source is normally of the reversible type, allowing for regenerative braking and facilitating energy recovery [15]. In the case of hybrid electric vehicles different electrical motors are used such as standard DC, induction AC, brushless DC, etc. and the energy storage system is usually an electrochemical battery. In some configurations a second electric motor can function as a generator to charge the energy storage [15]. Normally internal combustion engines are not efficiently operating at low load and low speed. In addition, fossil fuels have negative impacts on the environment. Therefore, hybridization is a pathway to address the low efficient operation of internal combustion engines and address the negative effects of fossil fuels. Other advantages of hybrid systems are enabling regeneration of energy, and optimal powertrain sizing for a specific powertrain or downsizing of the internal combustion engines [16]. hybrid vehicles are classified based on their configurations in parallel, series, and combined hybrid [15].

2.5.1 Series hybrid

In series hybrid the primary and secondary movers are not connected mechanically but the connection is achieved electrically. Moreover, the connection between the prime mover and wheels is achieved with different mechanisms with respect to the conventional configuration [5]. The two configurations including a series electric hybrids layout and a series hydraulic hybrid layout are shown in figures 2.12 and 2.13 respectively.

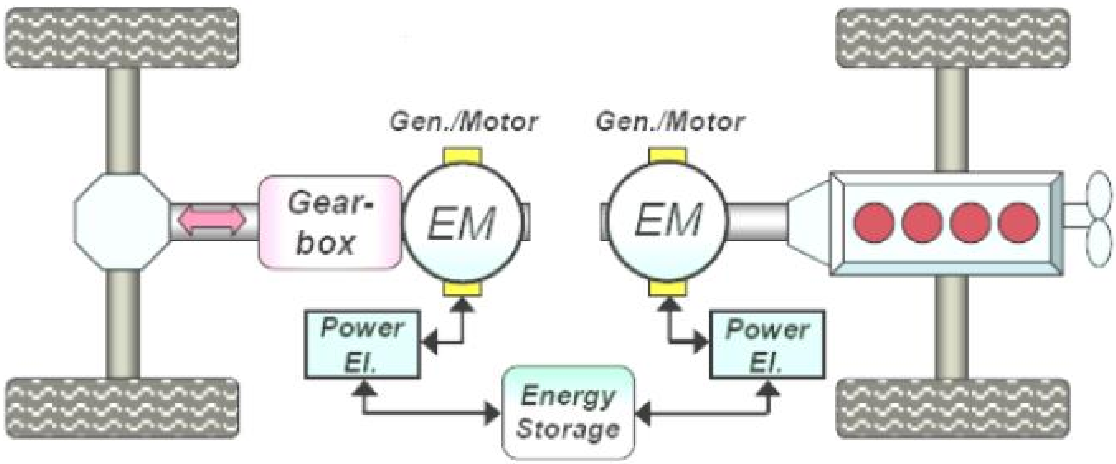

Figure 2.12: A series electric hybrid layout. Figure source: [2]

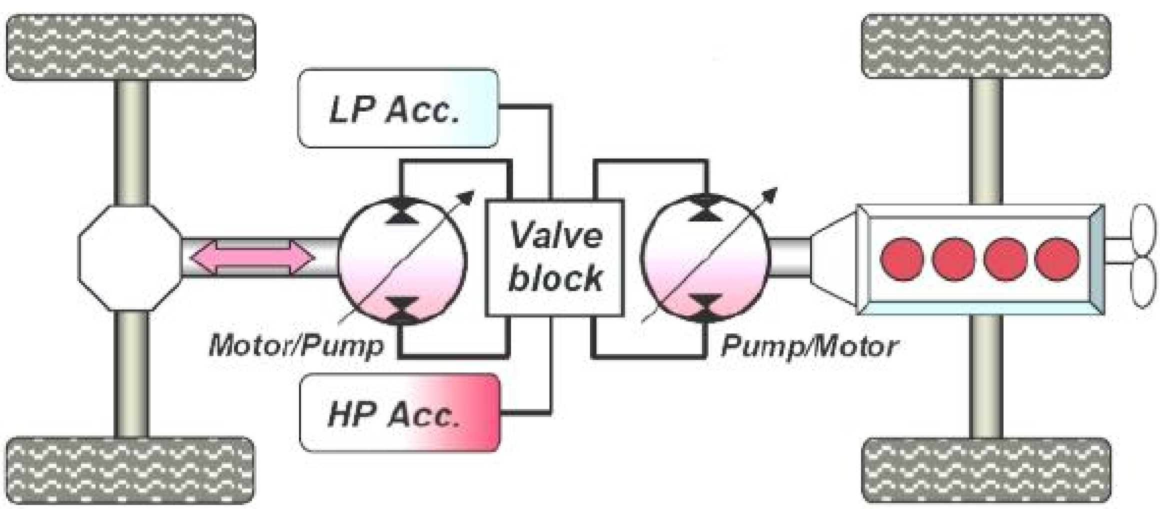

Figure 2.13: A series hydraulic hybrid layout. Figure source: [2]

In the series configuration the mechanical power provided by an internal combustion engine is transformed to electrical power as indicated in the figure 2.12, and hydraulic power as indicated in the figure 2.13 through an electric motor/generator or a hydraulic motor/pump in each configuration and stored in an energy storage system. Then the stored energy converted back to mechanical power to drive the wheels through another electric motor/generator or hydraulic motor/pump. In series hybrid due to the absence of mechanical connection between primary mover and wheels more components are required compared to the parallel hybrid, but this absence of mechanical connection also brings an advantage in the sense that the internal combustion engine can operate at optimal points in terms of torque and speed, thus the optimal fuel consumption and high internal combustion engine's efficiency can be achieved. However, lower efficiency for this configuration is probable since the power provided by internal combustion engine should be transferred to the wheels through more components therefore, power losses increase [15]. Likewise, the recuperation of kinetic energy similar to the parallel hybrid can be done during regenerative braking from the wheels to the energy storage system.

2.5.2 Parallel hybrid

The primary and secondary movers in parallel configurations are mechanically linked as is shown in figure 2.14. The prime mover that is usually an internal combustion engine can drive the wheels as in conventional vehicle, and the secondary mover operates to assist the prime mover or to recover the kinetic energy [15,16].

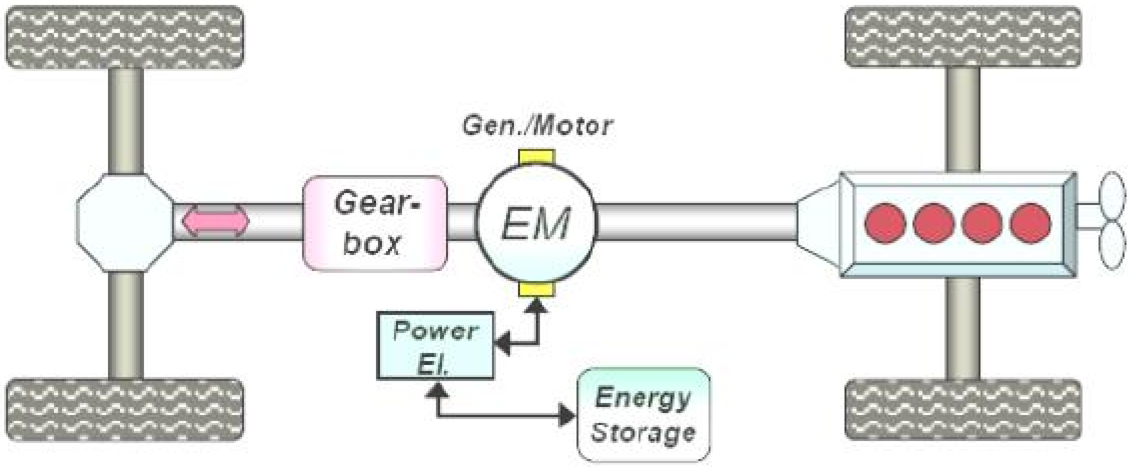

Figure 2.14: A parallel hybrid configuration equipped with an electrical motor as the second mover. Figure source: [2]

In figure 2.15 a parallel hydraulic hybrid configuration is also shown.

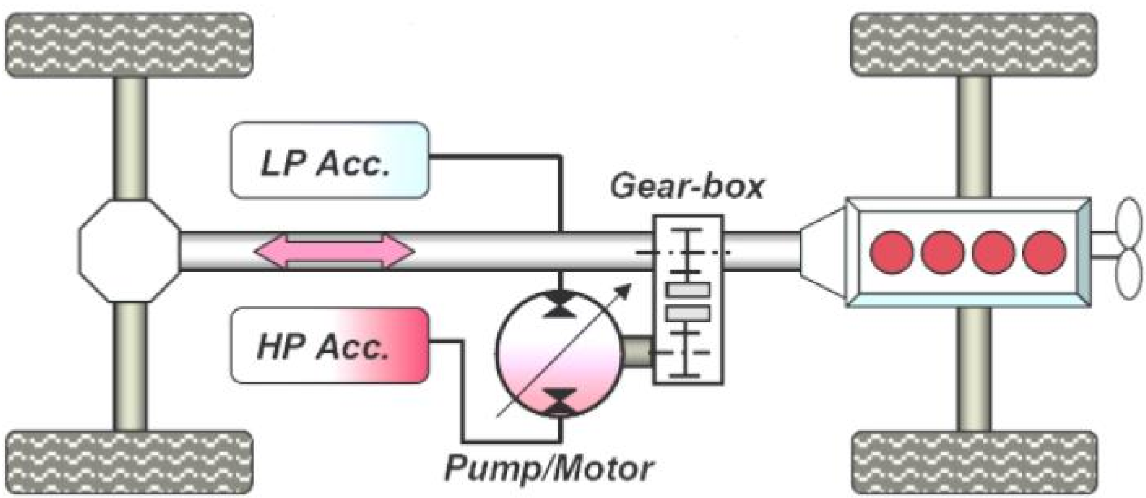

Figure 2.15: A parallel hybrid configuration equipped with hydraulic hybrid system. Figure source: [2]

The parallel configuration utilizes fewer components compared to the series hybrid but the mechanical connection between the movers has negative effect on mechanical efficiency of the movers and usually the internal combustion engine operates on suboptimal operating points [15].

2.5.3 Combined hybrid

Combined or complex hybrid configuration is achieved through a combination of parallel and series hybrid. This layout as it is indicated in figure 2.16 can work as a parallel configuration or series configuration employing a clutch on the mechanical shaft. In addition, this layout has the possibility to work in the power split mode utilizing both parallel and series configurations at the same time by using a planetary gear set. In this configuration the energy management strategy determines how to distribute the power between the internal combustion engine and hydraulic motor/electric motor to optimize performance and efficiency.

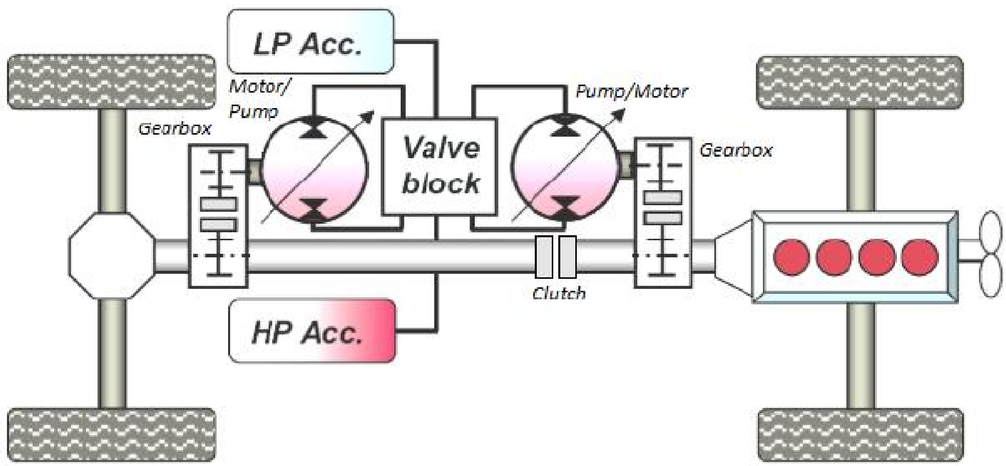

Figure 2.16: A combined hydraulic hybrid configuration. Figure source: [2]

3. Hydraulic Hybrid system

The hybrid concept considered in this paper is implemented by the addition of an add-on hybrid system (enclosed by dashed line) to a conventional wheel loader driveline as indicated in figure 3.1 below. A variable displacement hydraulic pump/motor is mechanically connected to the driveline to be able to achieve a parallel hybrid configuration. In this layout the internal combustion engine is directly connected to a torque converter and a Power Take-out (PTO) that is driving the P1 hydraulic pump. The P1 pump is moving the bucket. The torque converter's turbine shaft (output) is connected to a transmission system to transfer torque and speed to the wheels through the propeller shaft. By controlling the displacement of the hydraulic pump/motor mechanically connected to the transmission output shaft, a braking torque can be generated, and the hydraulic fluid is pumped into the accumulator; then the stored energy in the accumulator can be utilized in addition to the traction force to the wheels [4]. Accordingly, the required power that the internal combustion engine should provide is reduced and consequently fuel consumption and emission production will be reduced too. In this system two hydraulic on/off valves are used. The V1 valve positioned between hydraulic pump/motor and the accumulator acts to isolate the energy stored in the accumulator in conditions that the hybrid system does not operate. The V2 valve is used to bypass the pump to minimize drag losses when it is not used [4].

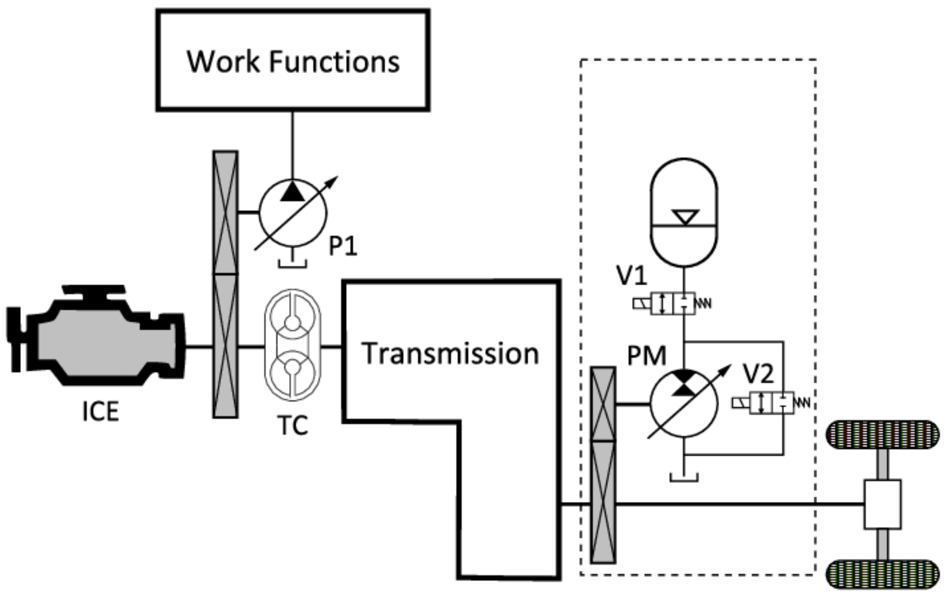

Figure 3.1: Hydraulic hybrid wheel loader drivetrain. Figure source: [4]

This add-on system has a minimal impact on the overall composition of the wheel loader since no modifications are made on the conventional wheel loader driveline components. It offers benefits in terms of fuel consumption, and efficiency for low cost.

3.1 Hydraulic hybrid system work in the drive cycle

In this section the potential functionalities of the add-on hydraulic system in each part of the drive cycle are discussed. Each short loading cycle and load-carry cycle consists of 6 phases. Each phase is described along with the potential functionality of the hydraulic part.

1. **Bucket filling**: Since the highest traction force is required in this phase, the stored energy in the accumulator should be employed. The advantage is that a lower amount of power is transferred through the torque converter thus higher efficiency of the torque converter is possible to achieve.
2. **Reverse from gravel pile**: At the end of this phase, recovery of the kinetic energy during braking is desirable to recharge the accumulator and prepare it for the following phase.
3. **Approaching the load receiver site**: In this stage using the energy stored in the accumulator is favorable during the acceleration of the vehicle since it can lead to lower fuel consumption, lower emission and higher torque converter efficiency.
4. **Unloading the bucket**: Dring this phase the wheel loader normally is stand still but to speed up the hydraulic systems the internal combustion engine's speed should be raised. The additional power from internal combustion engine can be exploited to recharge the accumulator instead of being dissipated in brakes.
5. **Reverse from the load receiver site**: Usage of the stored energy in the accumulator is desired during acceleration and similar to phase 2 recharging the accumulator during braking is achievable.
6. **Approaching the gravel pile**: In this phase the usage of the stored energy during acceleration is not recommended due to the fact that a high state of charge is required for the next bucket filling phase.

4. System modeling and simulation

A backward-facing simulation is exploited in this thesis to simulate the wheel loader motion, Figure 4.1. In addition, a recorded vehicle speed and traction force from a drive cycle shown in figure 4.2 are fed to the model. The parameter values are provided in table 3 In the appendix.

4.1 Backward-facing simulation model

The model has one input signal u which regulates the displacement of the variable displacement hydraulic pump/motor in the add-on part, and a state variable x, that indicates the pressure level of the accumulator i.e. the state of charge of the system. x is defines based on the equation 4.1:

$$x = \frac{p - p_0}{p_{max} - p_0} \tag{4.1}$$

Where p is the pressure level of the accumulator, p_0 is the pre charge pressure of the accumulator, p_{max} is the maximum pressure of the accumulator.

The model is updated at each timestep according to the equation 4.2:

$$x_{k+1} = f(x_k, u_k, t_k) \tag{4.2}$$

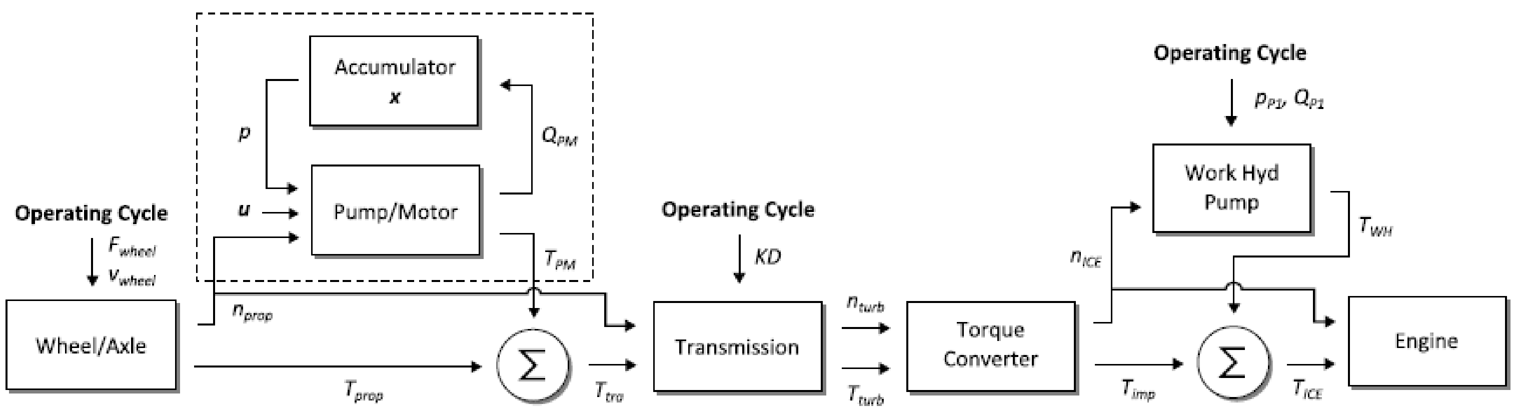

Figure 4.1: Backward-facing model of the wheel loader. Figure source [4]

The backward-facing model indicated in figure 4.1 is explained in the following sections from left to right.

4.1.1 Propeller shaft

The power of the engine is transferred to the wheels via the propeller shaft placed between the transmission system and the final drive. The force and the speed of the wheels, F_{wheel} and v_{wheel} respectively, are specified from the driving cycle. With the assumption of stiff tires, which means there is no slippage, the speed n_{prop} and the torque T_{prop} of the propeller shaft are calculated using equations 4.3 and 4.4:

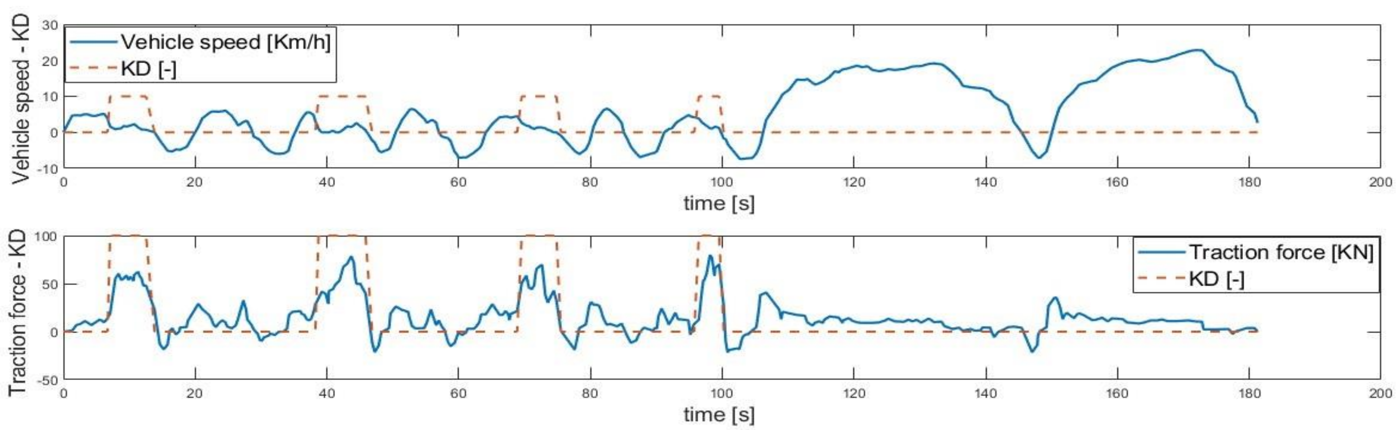

Figure 4.2: Vehicle speed and traction force recorded in a driving cycle made of three short loading cycles and one load-carry cycle. KD is the kick-down signal; it shows the bucket filling phase.

$$n_{prop} = \frac{v_{wheel}}{r_{tire}.\, i_0.\, 2\pi} \tag{4.3}$$

$$T_{prop} = \frac{F_{wheel}.\, r_{tire}.\, i_0}{\eta_{axle}^{j}} \tag{4.4}$$

Where r_{tire} is the radius of the wheels, i_0 is axle ratio, η_{axle} is axle efficiency, and j is equal to 1 in the case of positive output power and -1 in the case of negative output power.

4.1.2 Hydraulic hybrid equations

In this simulation model u is the unique control variable used to control the displacement of the hydraulic pump/motor:

$$\varepsilon_{PM} = u \tag{4.5}$$

The hydraulic pump/motor torque and flow rate are calculated by the equations 4.6 and 4.7 respectively:

$$T_{PM} = \frac{\varepsilon_{PM}.D_{PM}.P}{20.\pi} \mp T_{loss} \tag{4.6}$$

$$Q_{PM} = i_{PM}.\varepsilon_{PM}.D_{PM}.\frac{n_{prop}}{1000} - Q_{loss} \tag{4.7}$$

Where D_{PM} is the maximum displacement of the hydraulic pump/motor, P is the pressure of accumulator, T_{loss} is the torque loss of the pump/motor, i_{PM} is the gear ratio of the pump/motor, Q_{loss} is the flow loss of the pump/motor. In this thesis the efficiency of the hydraulic pump/motor is considered 95%.

The accumulator pressure can be calculated using equation 4.8 assuming ideal adiabatic gas compression and expansion (no heat transfer):

$$P = \frac{P_0 V_0^{\gamma}}{V^{\gamma}} - P_{loss} \tag{4.8}$$

Where $V_0$is the total volume of the accumulator, γ is the polytropic index, and P_{loss} is the pressure loss assuming 97% accumulator efficiency. The gas volume of the accumulator is calculated by:

$$V = V_0 - \int Q_{PM}.\eta_{acc}^{j} dt \tag{4.9}$$

Where η_{acc} is the efficiency of accumulator, j=1 for charge, and j=-1 during discharging the accumulator.

4.1.3 Transmission system

Remaining torques , $T_{tra,}$ that the transmission needs to provide to fulfill the required propeller shaft torque is:

$$T_{tra} = T_{prop} - i_{PM}.T_{pm} \tag{4.10}$$

In previously studied wheel loaders, a multi-gear mechanical transmission was considered in which the proper gear is selected based on the driving condition. This necessitates the implementation of a gear shifting strategy [17]. In this model a four-speed mechanical transmission shifting gears at fixed vehicle speed is used in order to avoid additional input signal to the model [4]. In addition, the assumption is that the gears are changed instantaneously. The gear shifting strategy with the gear ratios i_{tra} are represented below:

$$i_{tra} = \begin{cases} \mathrm{i_{F1}} \text{ for KD=1} \\ \mathrm{i_{F2}} \text{ for } 0 < \mathrm{v_{wheel}} < \mathrm{v_{23}} \text{ and KD=0} \\ \mathrm{i_{F3}} \text{ for } \mathrm{v_{23}} < \mathrm{v_{wheel}} < \mathrm{v_{34}} \text{ and KD=0} \\ \mathrm{i_{F4}} \text{ for } \mathrm{v_{34}} < \mathrm{v_{wheel}} \text{ and KD=0} \\ \mathrm{i_{R2}} \text{ for } \text{-}\mathrm{v_{23}} < \mathrm{v_{wheel}} < 0 \text{ and KD=0} \\ \mathrm{i_{R3}} \text{ for } \text{-}\mathrm{v_{34}} < \mathrm{v_{wheel}} < \text{-}\mathrm{v_{23}} \text{ and KD=0} \\ \mathrm{i_{R4}} \text{ for } \mathrm{v_{wheeel}} < \text{-}\mathrm{v_{34}} \text{ and KD=0} \end{cases} \tag{4.11}$$

The kick-down (KD) signal is activated when the extra high traction force is required (during bucket filling phase). This is the only time when the first forward gear $\mathrm{i_{F1}}$ is engaged while the first reverse gear is never used at all. A normal acceleration from zero speed is performed using the second forward gear $\mathrm{i_{F2}}$or the second reverse gear $\mathrm{i_{R2}}$ in case of acceleration in opposite direction. To calculate the torque and speed of the turbine shaft (output shaft of the torque converter) the equations 4.12 and 4.13 are used:

$$T_{turb} = \frac{i_{tra}.T_{tra}}{\eta_{tra}^{j}} \tag{4.12}$$

$$n_{turb} = \frac{n_{prop}}{i_{tra}} \tag{4.13}$$

Where η_{tra} is the efficiency of the transmission, $j = \pm 1$ for positive and negative power.

4.1.4 Torque converter

Having the turbine speeds and torques, in order to obtain the impeller speeds and torques, a torque converter model can be used. Indeed, the torque and the speed of impeller is a function of required speed and torque of the turbine:

$$T_{imp} = f(n_{turb}, T_{turb}) \tag{4.14}$$

$$n_{imp} = f(n_{turb}, T_{turb}) \tag{4.15}$$

These functions should be provided by component suppliers and are tailored to each converter design. While in this thesis the torque and speed of the impeller are calculated based on two graphs obtained from a 12-inches diameter torque converter which is a common size for a mid-size wheel loader [18]. Therefore, by extracting numerical data from these graphs figures 4.3, 4.4 and corresponding the calculated turbine speed obtained from the backward simulation model with the turbine speeds shown in the graph 4.3, the torque ratios for different turbine speeds are obtained. Then by knowing the torque ratios, the corresponding speed ratios for different torque ratios can be extracted from graph 4.4. In this thesis a torque converter with 90-degree impeller exit angle is considered. By knowing these ratios, turbine speed and turbine torque the impeller speed and impeller torque is achievable.

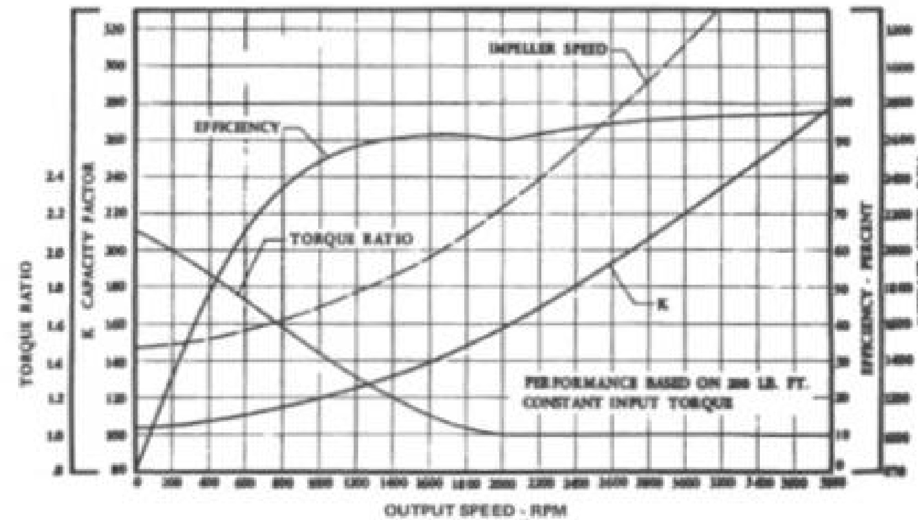

Figure 4.3: impeller speeds, efficiencies, and torque ratios VS turbine speeds. Figure source:[18]

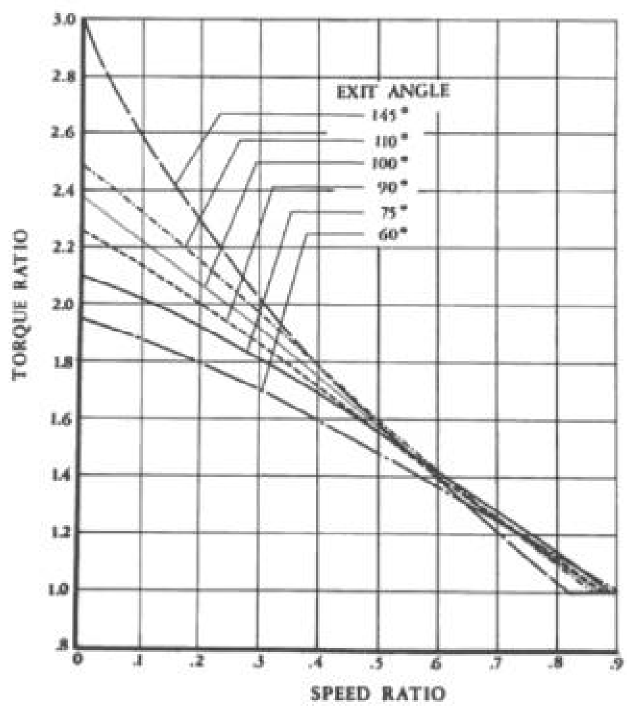

Figure 4.4: Torque converter speed ratio VS torque ratio for different impeller exit angles. Figure source:[18]

4.1.5 work hydraulic machine

The internal combustion engines' crankshaft is coupled with the impeller of the torque converter. Therefore, the rotational speed of the engine is:

$$n_{ice} = n_{imp} \tag{4.16}$$

The work hydraulic machine that is controlling the bucket movement is linked mechanically to the engine's output shaft by a gear with a constant ratio i_{WH}. The hydraulic machine's speed can be given by:

$$n_{WH} = n_{ice}.i_{WH} \tag{4.17}$$

The work function of the P1 pump/motor is not modelled in this thesis. The output power of the P1 pump/motor can be fed to the model. Likewise, the required torque and relative displacement of this pump/motor are achieved by equations similar to equations 4.6 and 4.7.

4.1.6 Internal combustion engine

The engine should be able to provide enough torques to fulfill the torque requirements of the working hydraulic, torque converter's impeller and other auxiliary systems. Therefore, internal combustion engine torque is:

$$T_{ice} = T_{imp} + T_{WH} + \frac{P_{aux}}{n_{ice}.2.\pi} \tag{4.18}$$

Where P_{aux} is the power needed for auxiliary systems such as fans, transmission charge pump, etc. and is assumed constant.

Then knowing the speed n_{ice} and the torque T_{ice} of the internal combustion engine the fuel consumption is determined using a lookup table for a 130KW engines' fuel consumption.

5. Energy management strategies

Hybrid vehicles rely on a hierarchical supervisory controller which determines required powers from each energy carrier on board. The purpose of the controller normally is to reduce the energy consumption in total while satisfying the powertrain needs and driver's demands. Control strategies can be classified based on the knowledge of drive cycles. If the drive cycle is known in advance like public transportation a non-causal controller can be used, but in the case of unknown or unpredictable drive cycle a causal controller might be used [1]. Another classification of controllers is:

- Rule-based controller
- Optimal controllers
- sub-optimal controllers

5.1 Rule-based controller

Rule-based controllers or in other words Heuristic controllers are controllers that are causal and operate based on Boolean logic. They are experience-based controllers, or they employ a simple logic to manage the power between the energy carriers. However, they can be very effective in terms of fuel consumption reduction in a hybrid vehicle [19]. The controller is designed based on specific vehicle parameters and operating conditions. The heuristic controllers can be set based on a set of if-then rules or conditions for a hybrid vehicle including a wheel loader, which are:

- If the state of charge (SOC) is lower than a specified limit, the hybrid system cannot be used for propulsion purposes.
- If the state of charge is higher than a specified limit, the charging process should be stopped.
- If the state of charge is high, in a specific gear the hybrid system can engage to support the primary mover.
- If the brake pedal is pushed and the state of charge is not in its upper limit, the hybrid system can be engaged to recover the kinetic energy and recharge the system.

These control architectures are easy to implement and computationally efficient, but they are not always the most optimal controllers in every situation [1].

5.2 Optimal and suboptimal controllers

An optimal controller is designed to achieve the best results in terms of performance criterion. This criterion can be fuel consumption, emissions, power efficiency or a combination of them. Hence, the driving cycle should be known in advance for obtaining the optimal solution. One of the powerful computational approaches used for solving optimization problems is Dynamic programming (DP). It is classified as an optimal control method, capable of identifying the best possible solution for a given state variable within complex multistage problems. DP requires a discretization of the state variables, control variables and time. Then in a backward phase the criterion for example fuel consumption is calculated at each time step for each single state variable, then in the forward phase the optimal cost-to-go (optimal trajectory) is calculated [20]. By integration of the trajectory over the driving cycle, the overall optimal fuel economy for that cycle would be determined. Dynamic programming serves as a solid foundation for developing suboptimal control strategies, as the optimal solutions derived can act as valuable benchmarks. However, DP tends to be computationally intensive; as the number of state variables increases, the computational costs grow exponentially [1]. Another problem that may happen in the case of complex hybrid architectures, is the so-called curse of dimensionality; since discretization of time, state variables, and control variables are required for this optimization method therefore for a complex hybrid architecture in which a high number of control variables must be defined this problem is probable.

Most often the optimal controllers are not practical in hybrid vehicles since the drive cycle should be known in advance. The sub-optimal controllers can be used instead, due to their causal properties. An analytical optimization method can be utilized to establish minimization functions and as they are computationally fast, real-time decisions are possible to minimize the fuel consumption [1]. The Equivalent-Consumption Minimization strategy (ECMS) is one of the most used sub-optimal controllers in hybrid vehicles. The controller is designed using a function that is determined analytically and is able to minimize fuel consumption at every timestep; in other words, it finds the best possible solution at each timestep. The controller will be explained in the next chapter in detail.

6. Control Strategy

6.1 Equivalent Consumption Minimization Strategy (ECMS)

As mentioned above rather than tackling a global minimization problem like dynamic programming, the ECMS solves a minimization problem for each time instant.

6.1.1 Mathematical formulation

The mathematical formulation of the minimization problem is:

$$\hat{J} = \int_{t_0}^{t_f} \min_{u \in U} \left[\dot{m}_{eqv}(u,t) \right] dt \tag{6.1}$$

S.t

$$SOC_{min} \leq SOC \leq SOC_{max}$$
$$u_{min} \leq u \leq u_{max}$$

$$\hat{u}(t) = \underset{u \in U}{\operatorname{argmin}}(\dot{m}_{eqv}(u,t) \tag{6.2}$$

$\hat{J}$ is the cost or in other word fuel consumption. A local optimization minimizes this quantity at each time step. $\hat{u}$ represents the optimal control strategy that minimizes the cost at each timestep.

6.1.2 Equivalence factor

In this control strategy an equivalence factor λ is introduced to represent the cost associated with utilizing energy from the reversible energy storage. However, this equivalence factor is significantly dependence on the driving cycle and an optimal equivalence factor for a specific driving cycle can result in increasing fuel consumption in other driving cycles. Since the drive cycle is unknown, many other works have been suggested such as the use of a lookup table generated from offline simulations that can be utilized in real-time applications. In this case, the equivalence factor is chosen based on the past and predicted driving conditions [21]. Hybrid vehicles normally contain two or more energy storages. Usually, one of these storages are reversible such as electrochemical batteries or accumulators in the context of hybrid electric vehicles or hydraulic hybrid vehicles which allow the energy flow into or from the storage. Using reversible energy is analogous to either using or conserving a specific amount of fuel from irreversible storage. Figure 6.1 indicates the ECMS concept during charging and discharging phase a reversible energy storage that is accumulator.

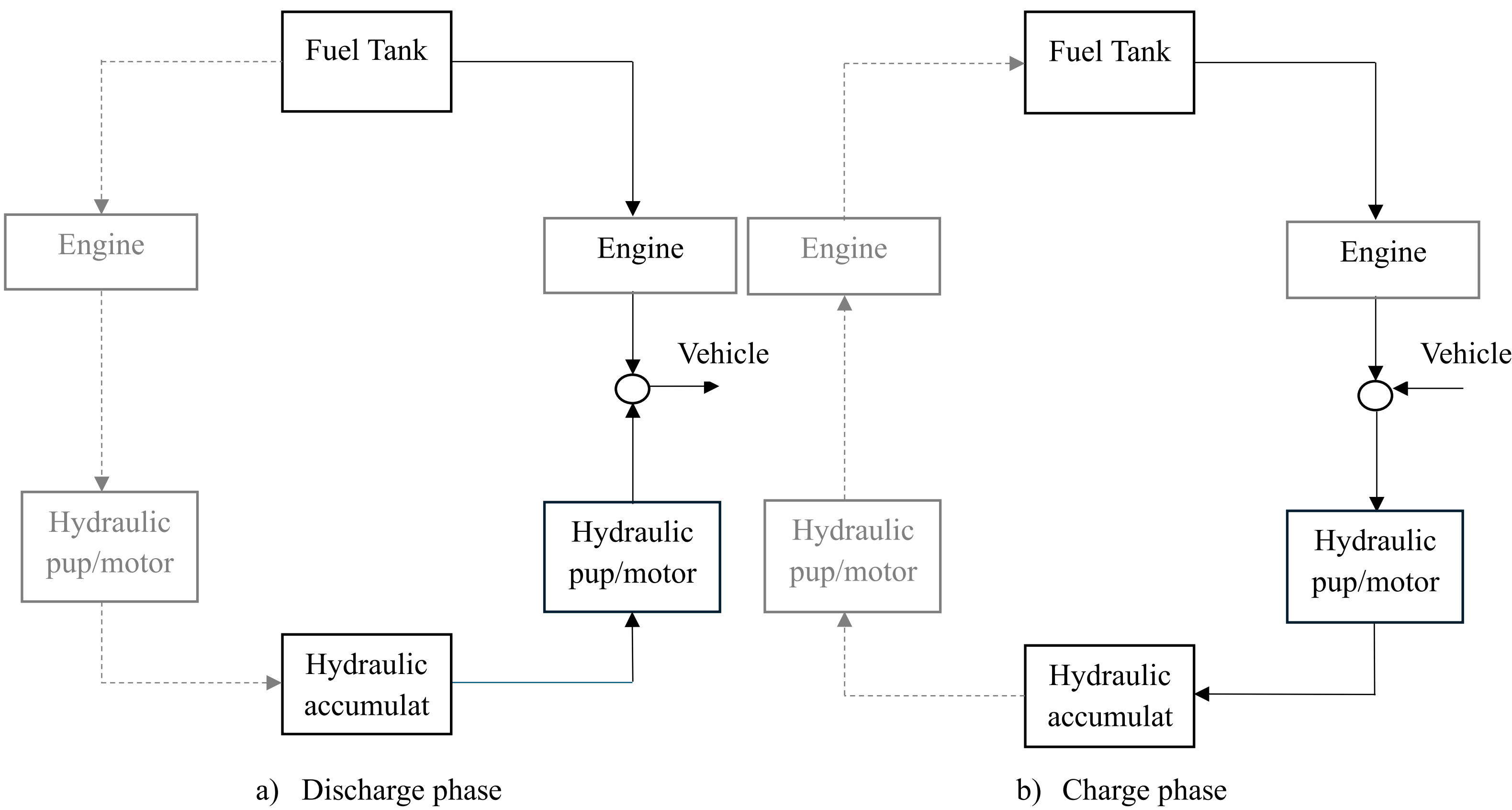

Figure 6.1: The paths of energy in ECMS for charging and discharging phases [25]

The equivalent fuel consumption is given by the equation 6.3:

$$\dot{m}_{f,eqv}(t) = \dot{m}_{fc}(t) + \lambda.\dot{m}_{hyb}(t) \qquad (6.3)$$

Where $\dot{m}_{fc}$ is the fuel consumption of the engine at each timestep, λ is the equivalence factor representing the efficiencies of different components involved in converting fuel into hydraulic power in this context, and vice versa, and $\dot{m}_{hyb}$

represents the projected fuel consumption when the reversible energy storage is employed in the future. $\dot{m}_{fc}$ and $\dot{m}_{hyb}$ are given by the equations 6.4 and 6.5:

$$\dot{m}_{fc}(t) = \frac{P_{eng}(t)}{\eta_{eng}.Q_{lhv}} \tag{6.4}$$

$$\dot{m}_{hyb}(t) = \frac{P_{hyb}}{Q_{lhv}} \tag{6.5}$$

Where Q_{lhv} is the lower heating value of the fuel which indicates energy content per unit of mass. The sign of P_{hyb} determines whether $\dot{m}_{hyb}$ is positive or negative, which in turn affects the equivalent consumption, making it either greater or lesser than the actual fuel consumption.

The equivalent fuel consumption is modified by using an appropriately constructed penalty function, p, in order to guarantee the state of charge is controlled properly and maintained within its limits. The equivalent fuel consumption is given by:

$$\dot{m}_{f,eqv} = \dot{m}_{fc} + \lambda.\dot{m}_{hyb}.p(SOC) \tag{6.6}$$

$p(SOC)$ is a correction function that account for the difference between the current state of charge SOC(t) and the target state of charge, SOC_{target}. The $p(SOC)$ is defined by the equation 6.7:

$$p(SOC) = 1 - \left(\frac{SOC(t) - SOC_{target}}{(SOC_{max} - SOC_{\min})/2}\right)^{a} \tag{6.7}$$

Figure 6.2 displays how the values of $p(SOC)$ vary with different exponents a.

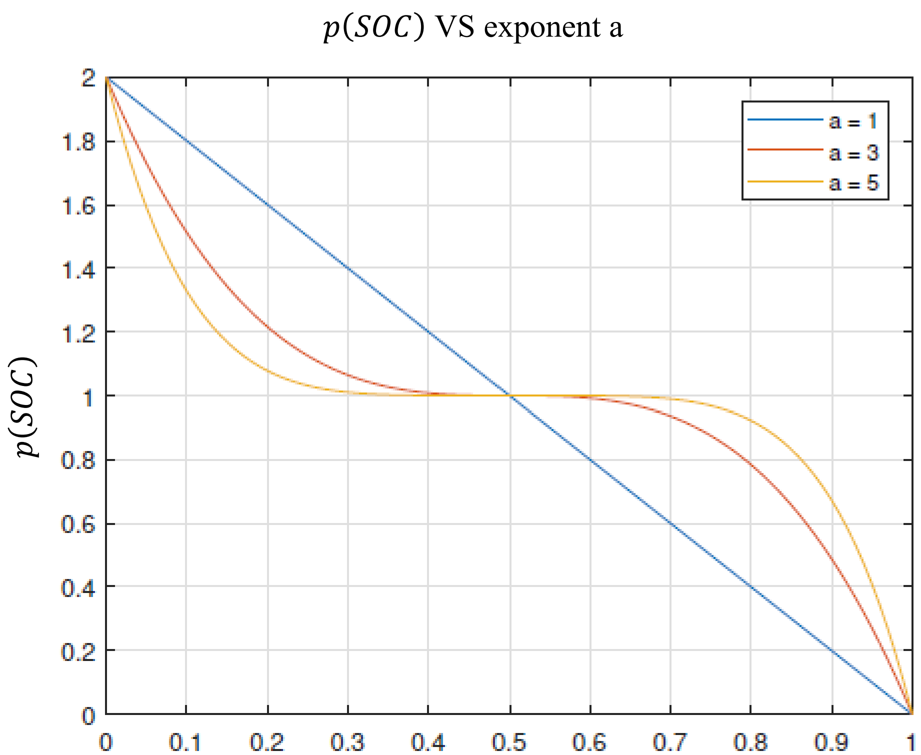

Figure 6.2: Penalty function p(SOC) for different exponent a. SOCmin=0, SOCmax=1, SOCtarget=0.5

As it is indicated in the figure 6.2, $p(SOC)$ is higher than one if the SOC is lower than the target state of charge (0.5) and $p(SOC)$ become lower than one if SOC is higher than the target value SOC_{target}. By doing so the controller decides to use more or less the alternative energy storage. The exponent, a, modifies the state of charge interval surrounding the SOC_{target} with near-unity weighting and acts as an additional deign parameter for the ECMS controller design.

6.1.3 ECMS algorithm

The ECMS algorithm operates based on the following steps:

1. Discretization of the control interval into finite numbers with the same distance.
2. Calculation of the equivalent fuel consumption $\dot{m}_{f,eqv}$ for each control candidate at each time step.
3. Selection of control candidates that lead to the lowest equivalent fuel consumption.

6.1.4 Optimal equivalence factor

As was mentioned before, the equivalence factor is highly cycle-dependent i.e. an equivalence factor that can cause charge sustainability in a drive cycle may result in charge depletion or over charging in another drive cycles. Therefore, the optimal value of the equivalence factor must be determined to ensure that the ECMS achieves charge sustainability. In this thesis the Bisection method is used as a structured search algorithm to obtain the optimal equivalence factor for the driving cycle by successively halving the search interval. Figure 6.3 shows a graphical representation of the bisection method that starts by guessing two points a1 and b1 such that the function values at these points F(a1) and F(b1) have opposite signs. Then by halving the interval and evaluation of F(c), depending on the sign of F(c) either replace a1 or b1 with c to form a new interval. The process is repeated until either the width of the interval or the function value at the midpoint becomes smaller than the predefined tolerance, at which point c is considered the final value.

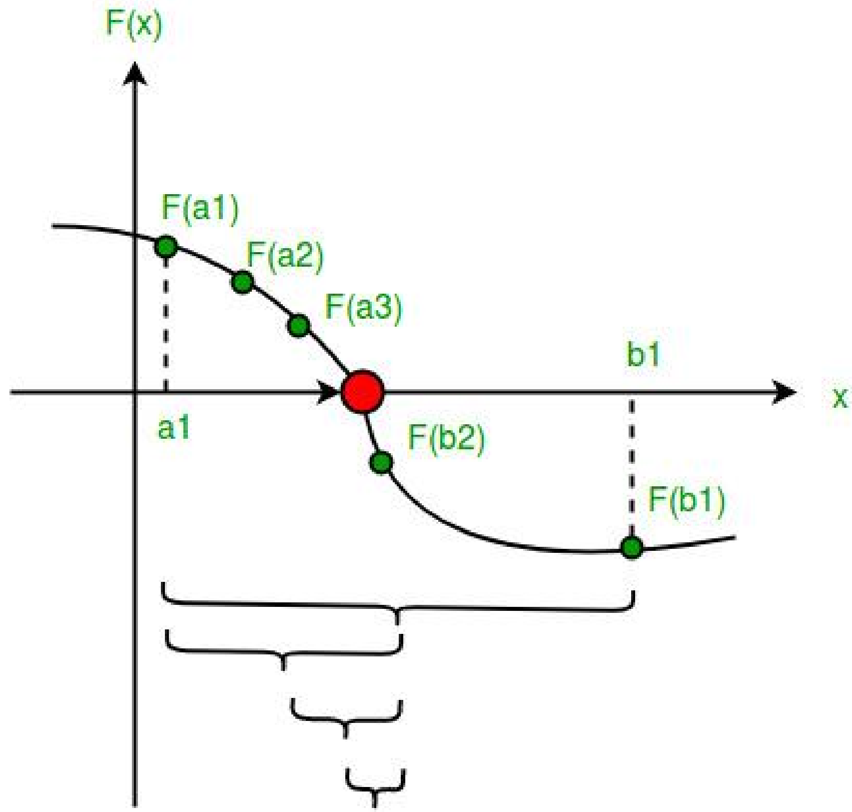

Figure 6.3: Bisection method graphical representation for finding the optimal equivalence factor

7. Results and Strategy analysis

7.1 Analysis

In figure 7.1 it is shown how the hybrid system assists the internal combustion engine by providing torque to the wheels (ps: power split). As was mentioned earlier the energy source in this case acts as a supercapacitor, thus the stored energy is utilized completely as soon as the accumulator is charged. Charging the accumulator (ch: charging) can be done during regenerative braking or by internal combustion engine only if the state of charge is not at its maximum. However, the results show that the accumulator is charged mainly via regenerative braking, since using the internal combustion engine to charge the accumulator can lead to an increase in fuel consumption. Besides that, As the accumulator pressure ranges between 280 bar-400 bar, the torque that the hydraulic pump/motor can provide is limited. Therefore, in a few brief instances the hydraulic pump/motor acts as the sole torque provider, and the internal combustion engine only provides torque to auxiliary systems and not to the wheels (ph: pure hydraulic). Moreover, another color can be present in this graph

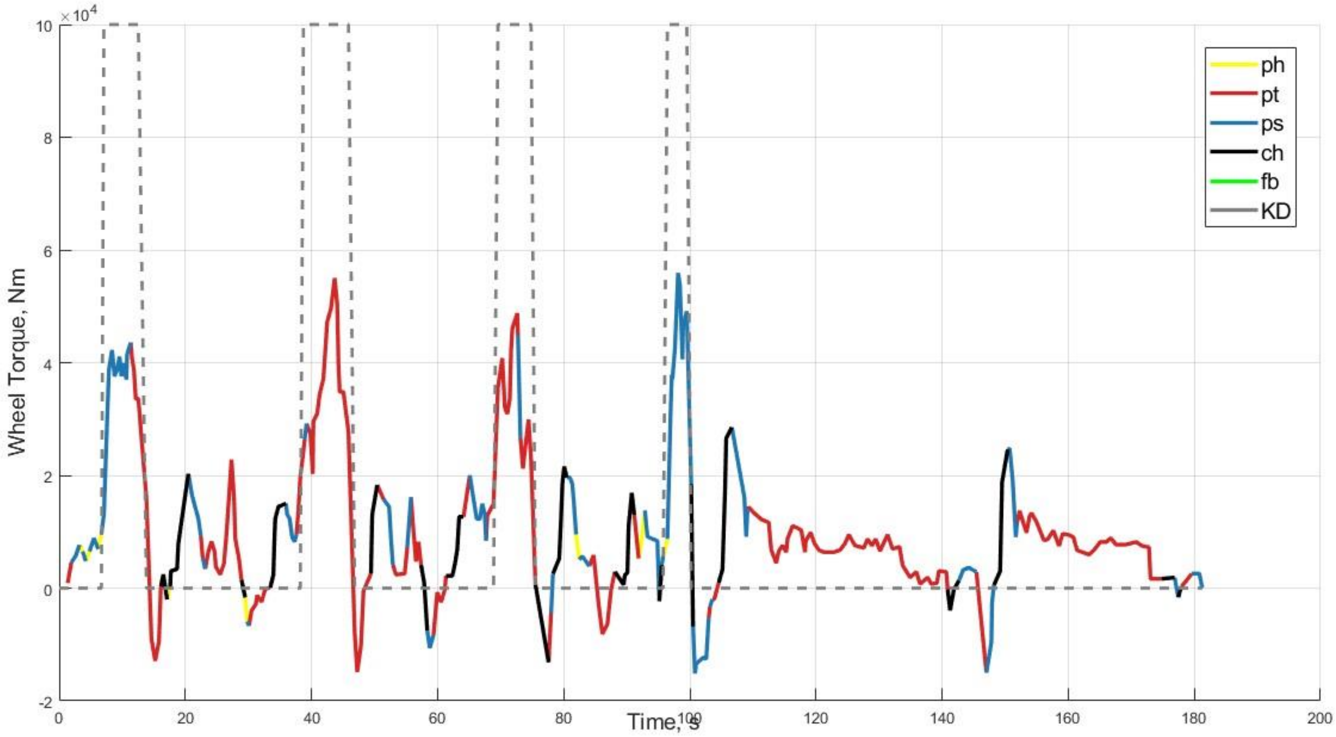

Figure 7.1: wheel torque profile with the torque providers specified at each instance. pt: pure thermal, ph: pure hydraulic, ps: power split, ch: charging, fb: friction brake, KD: kick-down signal

if the regenerative brakes do not operate (fb: friction brakes). i.e. if the hydraulic pump/motor does not come into action to charge the accumulator during braking phases. In addition, the gray dashed line indicates Kick-down signal activation -when the highest amount of torque is demanded-, but as can be seen the hydraulic pump/motor is utilized in only two of the four kick-down phases. This comes from the fact that the ECMS controller tries to reduce fuel consumption at each timestep without predicting future timesteps, thus the controller utilizes the accumulator whenever it is charged and ready to use. This is the reason why the accumulator can be at its minimum pressure when the kick-down phases happen.

Likewise, figure 7.2 represents the vehicle speed profile with different power providers specified. During acceleration phases whether in forward or backward motion, the controller tries to exploit hydraulic pump/motor as long as it is allowed by the energy stored in the accumulator. Whereas, in the load-carry cycle, the hybrid system comes into action during acceleration in both cycles, as the accumulator was charged during the braking phases before the start of the load-carry cycle. It is evident that before the end of the driving cycle, the internal combustion engine takes over the responsibility of charging the accumulator (indicated by the red circle in figure 7.2). By doing so, the driving cycle ends with the final state of charge close to the initial state of charge, ensuring charge sustainability (shown in figure 7.4).

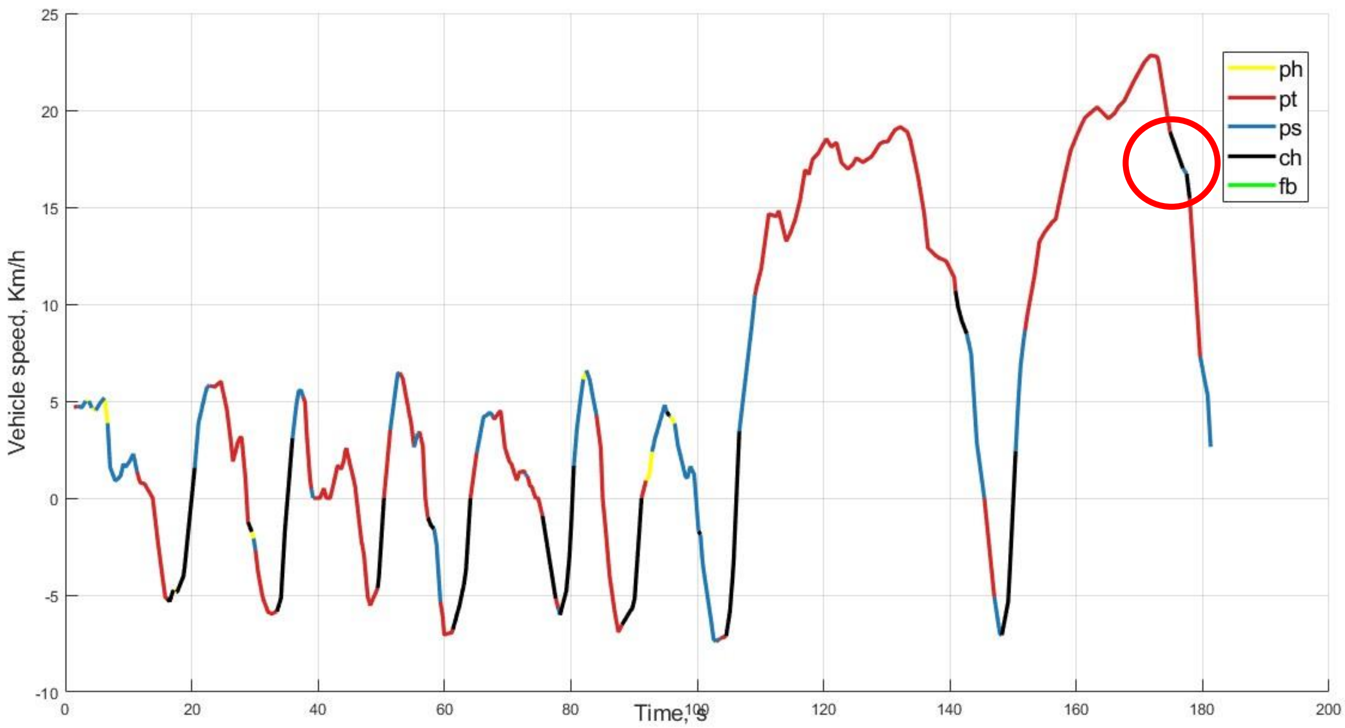

Figure 7.2: Vehicle speed profile with the power providers specified at each instance. pt: pure thermal, ph: pure hydraulic, ps: power split, ch: charging, fb: friction brake

Figure 7.3 indicates traction force of the vehicle and the torque provided by the hydraulic pump/motor. As was mentioned before, only in the first and the last kick-down phases the hydraulic pump/motor is activated not in all four. All in all, hydraulic pump/motor provides torque when the traction force at the wheels is high only if the pressure in the accumulator is higher than the minimum pressure; otherwise, the total traction force should be provided by the internal combustion engine. Due to the limited size, and pressure range of the accumulator the maximum torque provided by the hydraulic pump/motor is 717Nm which is not too high compared to the wheel torques. During forward movements of the wheel loader, regenerative braking is utilized when the traction force or wheel torque is negative therefore, the hydraulic pump/motor's torque should be negative. In the backward movements, however, a positive traction force or wheel torque indicates braking, thus positive hydraulic pump/motor's torque enables regenerative braking.

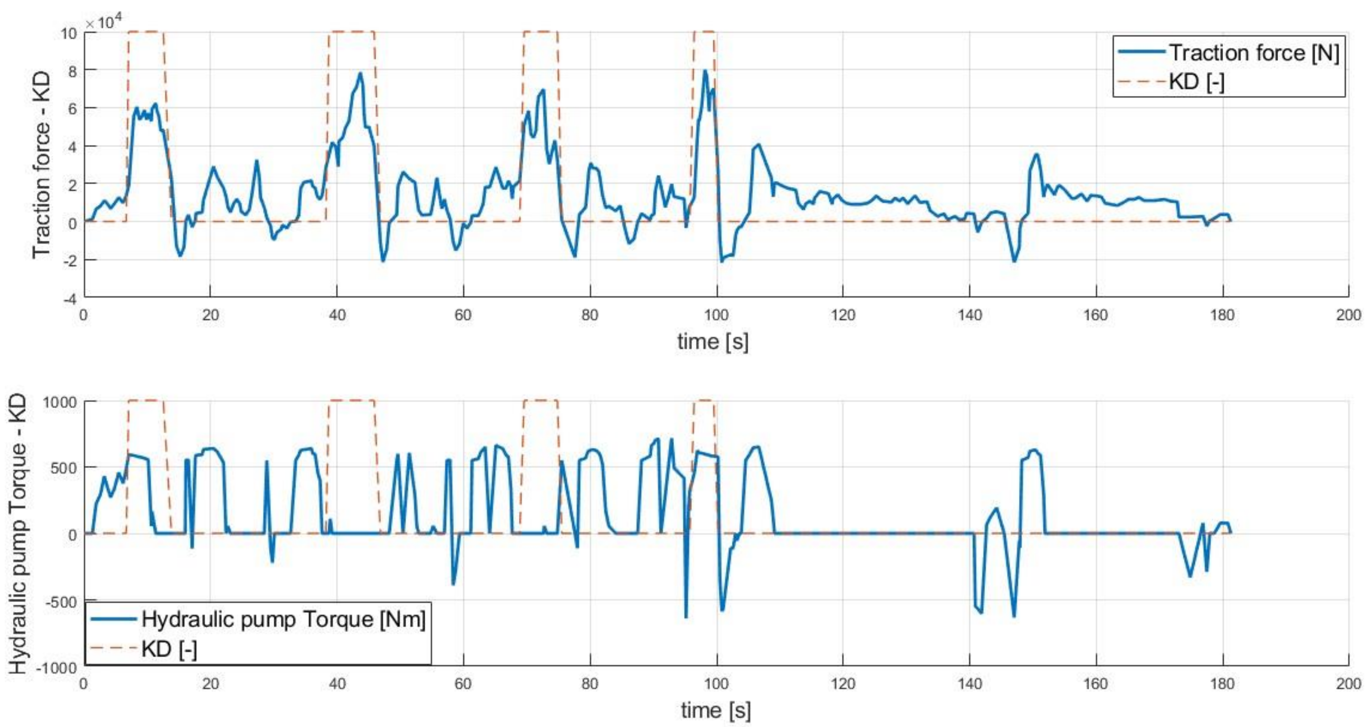

Figure 7.3: Traction force at the wheels and hydraulic pump/motor torque

The volume of the accumulator in this hydraulic hybrid system is changed through the hydraulic pump/motor flow rate. The flow rate of the hydraulic pump/motor is directly proportional to the wheel loader's speed. Therefore, by considering the fact that the most charging phases occur during braking, the wheel loader's speed is relatively low compared to that of a passenger vehicle, and the braking phases are short, the accumulator is typically not fully charged during regenerative braking. Figure 7.4 exhibits normalized accumulator pressure profile (SOC) in the driving cycle along with the KD signal. Moreover, it is clear that by employing the bisection algorithm, the equivalence factor can be determined, enabling the final state of charge to be as close as possible to the initial one, thereby enabling charge sustainability.

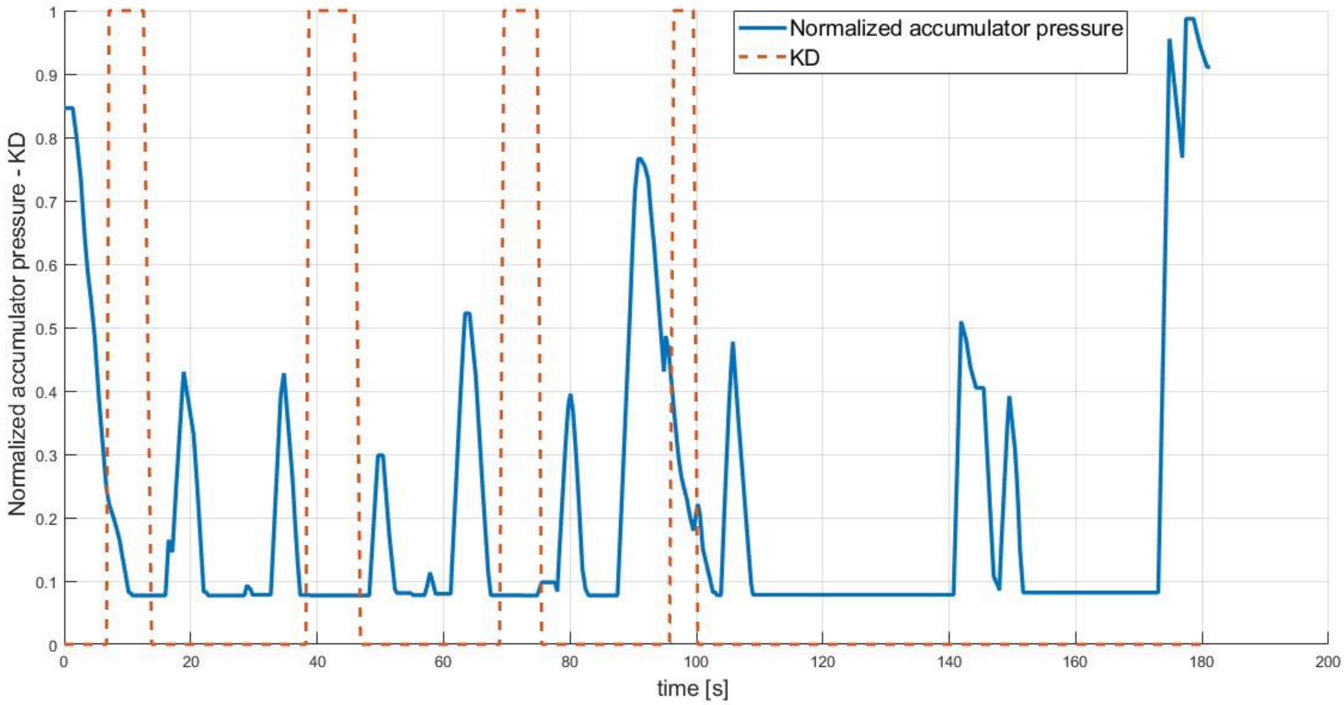

Figure 7.4: Normalized state of charge of the accumulator

7.2 Fuel consumption

By selecting the optimal equivalence factor - considering its strong dependence on the driving cycle in ECMS - and applying the bisection algorithm to ensure the charge sustaining mode, the ECMS reduces fuel consumption by 8% compared to the conventional wheel loader. In table 7.1 the values of fuel consumption for conventional and hydraulic hybrid wheel loader along with the optimal equivalence factor are specified.

	Optimal Equivalence factor	Traveled distance [km]	Fuel consumption [kg]	Fuel economy [l/km]	Fuel consumption [l/h]	Initial SOC	Final SOC
Conventional	-	0.40	0.47	2.22	11.13	-	-
ECMS	-0.0490	0.40	0.43	2	10.24	380	388
% Reduction	-	-	8.5	9.9	7.9		

Table 7.1 Fuel consumption comparison

In the table below the effect of the initial value of state of charge on fuel consumption is also considered. The improvements in the table are based on the first fuel consumption value.

Initial state of charge	Final state of charge	SOC deviation	Fuel consumption [l/h]	% Improvement
400	398.956	-1.04	10.43	0
380	388.409	+8.40	10.24	1.8
340	310.083	-29.91	10.18	2.3
320	310.101	-9.89	10.21	2.1
300	306.171	+6.17	10.24	1.8
280	280	0	11.13	-6.7

Table 7.2 :The effect of initial value of state of charge on fuel consumption

Table 7.2 acknowledges that the two initial state-of-charges, 380 and 300, result in the maximum fuel consumption reduction and the best charge sustainability outcomes. However, since the simulated driving cycle lasts only 3 minutes, certain initial state-of-charge values may not allow for precise charge sustainability behavior.

8.Conclusions

All in all, from the results it is obvious that the ECMS is effective in fuel consumption reduction if an appropriate equivalence factor is chosen. However, by replacing the accumulator with a bigger one the fuel consumption would be further reduced, as more torque for a longer time can be provided by the hydraulic pump/motor to assist the internal combustion engine or alternatively, the hydraulic pump/motor may be the only source to fulfill the required traction force for a longer time.

In addition, providing a part of a required torque by hydraulic pump/motor would increase the efficiency of the torque converter since lower amount of torque passes through the torque converter.

Although the ECMS control strategy needs prior knowledge of the drive cycle, it is efficient for hydraulic hybrid wheel loaders as their operational cycles are typically repetitive and can be defined in advance, unlike the hybrid electric vehicles, which often have long and unpredictable drive cycles.

9. Future works

To improve the control strategy and make the results more realistic, more developments must be made. In this paper some physical effects that are neglected or simplified can be taken into account in order to have a model that is closer to the real wheel loader. For instance, in this model gear shifting is considered instantly and only is based on the vehicle speed and not the position of the accelerator pedal. To make the model more realistic a delay time can be considered.

The accumulator considered in this thesis as secondary energy storage has a relatively low volume which results in low amount of stored energy. In theory a larger accumulator would be beneficial in terms of additional fuel consumption reduction. A suggestion is to compare how this ECMS strategy behaves with larger accumulators.

In this thesis one unique equivalence factor is considered during charging and discharging. To further improve the ECMS strategy these two factors can be different. However, the bisection algorithm should be modified since the algorithm is designed to find one equivalence factor.

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Appendix

The parameter values used in this thesis are presented in the table below.

parameters	symbols	values
Pump/motor displacement	D_{pm}	70 cc/rev
Pump/motor gear ratio	i_{pm}	1.76
Pump/motor max speed	n_{max}	4000 rpm
Accumulator max pressure	p_{max}	400 bar
Accumulator pre-charge pressure	p_0	270 bar
Accumulator volume	V_0	15 L
Wheel radius	r_{tire}	0.70 m
Axle ratio	i_0	0.050
Axle efficiency	η_{axle}	0.92
Transmission efficiency	η_{tra}	0.97
Work hydraulic pump	D_{wH}	90 cc/rev
Auxiliary power	P_{aux}	4.0 KW
Forward ratios	$[i_{F1}, i_{F2}, i_{F3}, i_{F4}]$	[4.0,2.0,1.0,0.5]
Backward ratios	$[i_{R1}, i_{R2}, i_{R3}, i_{R4}]$	-[4.0,2.0,1.0,0.5]
Shift speeds	$[v_{23}, v_{24}]$	[3.1,6.2] m/s

Table 3: parameters