Fluid Power Pumps and the Electrification

With a Focus on Discrete Displacement Control
in Load Handling Applications

Samuel Kärnell

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To my opponent

” Man ska vara snäll.

Unknown

Abstract

More and more vehicles are being electrified. Mobile working machines and heavy trucks are not excluded, and these machines are often hydraulically intense. Electrification entails new requirements for the hydraulic system and its components, and these requirements must be taken into consideration.

Hydraulic systems have looked similar for a long time, but now there is an opportunity to advance. Many things change when a diesel engine is replaced with an electric motor. For example, variable-speed control becomes more relevant, electric regeneration becomes possible, and the use of multiple prime movers becomes an attractive alternative. The noise from the hydraulic system will also be more noticeable when the diesel engine is gone. Furthermore, the introduction of batteries to the system makes the energy more valuable, since batteries are heavy and costly compared to a diesel tank. Therefore, it is commercially viable to invest in the hydraulic system.

This thesis revolves around the heart of the hydraulic system, that also is the root of all evil. That is the pump. Traditionally, a pump has had either a fixed displacement or a continuously variable displacement. Here, the focus is on something in between, namely a pump with discrete displacement. The idea of discrete displacement is far from unique, but has not been investigated in detail in combination with variable speed before. In this thesis, a novel design for a quiet pump with discrete displacement is presented and analysed. The results show that discrete displacement is relevant from an energy perspective for machines working extensively at high pressure levels and with low flow rates, and that a few discrete values are enough to make a significant difference. However, for other cycles, the possible energy gains are very limited, but the discrete displacement can be a valuable feature if downsizing the electric machine is of interest.

Acknowledgements

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Tack!

Linköping, May 2020
Samuel Kärnell

Abbreviations

AC	Alternating Current
AIP	Artemis Intelligent Power
DC	Direct Current
DD	Digital Displacement®
DDBC	Direction-dependent Boundary Control
DDM	Digital Displacement Machine
DDP	Digital Displacement Pump
EHA	Electro-hydraulic Actuator
FOC	Field-oriented Control
ICE	Internal Combustion Engine
PMSM	Permanent Magnet Synchronous Machine
PWM	Pulse-width Modulation
WPP	Wobble Plate Pump

Nomenclature

η_{hm}	Hydro-mechanical efficiency	[-]
η_{tot}	Total efficiency	[-]
η_v	Volumetric efficiency	[-]
ω	Angular velocity	[rad/s]
ω_{ref}	Angular velocity reference	[rad/s]
θ	Angular position	[rad]
D	Displacement	[m ³ /rad]
i_a	Current, a-axis	[A]
i_α	Current, α -axis	[A]
i_b	Current, b-axis	[A]
i_β	Current, β -axis	[A]
i_c	Current, c-axis	[A]
i_d	Current, d-axis	[A]
$i_{d,ref}$	Current reference, d-axis	[A]
i_q	Current, q-axis	[A]
$i_{q,ref}$	Current reference, q-axis	[A]
n_{cyl}	Number of cylinders	[-]
n_d	Number of displacement settings	[-]
n_g	Number of pump elements	[-]
$n_{g,min}$	Number of cylinders in the smallest group	[-]
P_1	Power, load 1	[W]

P_1	Power, load 2	[W]
P_{loss}	Power loss	[W]
p_{in}	Inlet pressure	[Pa]
p_{max}	Maximum pressure	[Pa]
p_{out}	Outlet pressure	[Pa]
q_{in}	Inlet flow	[m ³ /s]
q_{max}	Maximum flow	[m ³ /s]
q_{out}	Outlet flow	[m ³ /s]
S	Transistor switching signal	[-]
T	Torque	[Nm]
T_{ref}	Torque reference	[Nm]
V	Voltage, DC-link	[V]
$v_{\alpha,ref}$	Voltage reference, α -axis	[V]
$v_{\beta,ref}$	Voltage reference, β -axis	[V]
$v_{d,ref}$	Voltage reference, d-axis	[V]
$v_{q,ref}$	Voltage reference, q-axis	[V]

Papers

The following publications are included in the thesis and can be regarded as its foundation. They will be referred to by their Roman numerals. Apart from formatting changes and minor errata, they are reproduced in their original form. At the time of publication, Paper [III] had been accepted but not presented and Paper [IV] was under review.

- [I] S. Kärnell, A. Dell’Amico, and L. Ericson, “Simulation and validation of a wobble plate pump with a focus on check valve dynamics,” in *2018 Global Fluid Power Society PhD Symposium (GFPS)*, IEEE, 2018, pp. 1–8. DOI: 10.1109/gfps.2018.8472400.
- [II] S. Kärnell and L. Ericson, “As simple as imaginable - an analysis of novel digital pump concepts,” in *Proceedings of the 16th Scandinavian International Conference on Fluid Power (SICFP19), 22-24 May, Tampere, Finland*, 2019, ISBN: <http://urn.fi/URN:ISBN:978-952-03-1302-9>.
- [III] S. Kärnell, A. Rankka, A. Dell’Amico, and L. Ericson, “Digital pumps in speed-controlled systems - an energy study for a loader crane application,” in *Proceedings of the 12th International Fluid Power Conference (12th IFK), 9-11 March, Dresden, Germany*, 2020.
- [IV] S. Kärnell and L. Ericson, “Why not open-circuit? an analysis of a regenerative speed-controlled hydraulic actuator concept (submitted),” in *Proceedings of the ASME/BATH 2020 Symposium on Fluid Power and Motion Control, FPMC 2020*, American Society of Mechanical Engineers, 2020.

The author of this thesis is the main author of all appended papers and has been responsible for modelling and conducting the experiments behind the results. The co-authors have had a supervisory function.

Additional Publications

The publication below, to which the author of this thesis contributed to the prototype design and measurement data collection, is not included in the thesis but is relevant to the topic.

- [V] L. Ericson, S. Kärnell, and M. Hochwallner, “Experimental investigation of a displacement-controlled hydrostatic pump/motor by means of rotating valve plate,” in *Proceedings of the 15th Scandinavian International Conference on Fluid Power (SICFP17)*, June 7-9, Linköping, Sweden, Linköping University Electronic Press, 2017, pp. 19–27.

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Appended Papers

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1

Introduction

People like to move things. Often heavy things, in which case we tend to be assisted by machines such as cranes, wheel loaders and excavators. However, it is time to start moving things in a more sustainable manner. The consumption of fossil fuels must be reduced, and the electrification of machines is a step towards improved sustainability.

The electrification of passenger cars has come a long way, and the trend is pointing upwards. A similar trend is expected for load handling machines. However, load handling work requires energy, and energy is valuable in mobile electric vehicles since batteries are both costly and heavy. Therefore, it will be an even higher priority than ever before to minimise energy consumption for the load handling work. However, it is not only energy efficiency that is important. Drivability must also be retained, and noise emissions must be considered.

Today, load handling applications rely heavily on hydraulics, and they will continue to do so. However, hydraulics must be adapted in line with electrification, which offers new design possibilities. New solutions at both system level and component level must be developed and evaluated, and this thesis is part of that work.

1.1 Aim and Research Questions

The overall theme of this thesis is pump design considerations when moving towards electric drives in mobile machines. However, the research focuses more specifically on the potential use of digital pumps, which here are defined as pumps with discretely variable displacement. The work intends to answer the following questions:

- **RQ1:** How can a Wobble Plate Pump (WPP) be transformed into a digital pump?
- **RQ2:** What are the pros and cons of using digital pumps in combination with variable speed drive?
- **RQ3:** When is the use of digital pumps of interest in electrified mobile systems?
- **RQ4:** To what extent will multi-quadrant operation of hydraulic machines be desirable when moving towards electric drives?

1.2 Overview of Appended Papers

Paper [I] includes an analysis of a conventional WPP and can be regarded as an initial investigation to address RQ1. In Paper [II], different digital pump concepts – based on the WPP – are presented and analysed. The focus is on flow pulsations. This addresses both RQ1 and RQ2. Paper [III] addresses the potential benefits of using a digital pump instead of a fixed pump in a speed-controlled system, which provides a foundation for answering RQ2 and RQ3. Paper [IV] examines pump-controlled systems and especially an open-circuit architecture. This paper offers insights into the potential applications for open-circuit pumps and closed-circuit pumps, and therefore aims to answer RQ4.

1.3 Methodology

This is applied research. It is based on industrial needs, but the aim is increased knowledge. A general view of the methodology for the research is shown in Fig. 1.1. In summary, established theories are combined in new ways and new knowledge and insights emerge. Validation of models is generally desirable to ensure usability, but non-validated results are also valuable since they still gives insights in the potentials.

Figure 1.1 Block diagram of the general methodology for this research. The grey lines and blocks can often be considered desirable but not required.

1.4 Delimitations

Throughout this work, a loader crane has been considered as the use case. Other applications are, however, also relevant.

The analysis of the digital pump design has revolved around a specific WPP design. There are other pumps to which the same principle can be applied, but these have not been investigated. Furthermore, only Permanent Magnet Synchronous Machines (PMSMs) have been considered as the drive unit. The switching dynamics have been analysed for the digital pump, but switching in combination with variable speed has not yet been analysed due to time limitations.

Furthermore, component costs will be discussed, but no cost analysis has been conducted.

1.5 Thesis Outline

The thesis is built up as follows. Chapters 2, 3 and 4 should be regarded as introductory chapters to the field whilst chapters 5, 6 and 7 focus more on the specific research. In chapter 8, the content of the thesis is merged into a discussion and in chapter 9, the research questions are answered. Comments regarding prospects and future work can be found in chapter 10. Chapter 11 contains short summaries of the appended papers. This is then followed by the appended papers.

2

Hydraulic Systems

Mankind has made use of hydrostatics since ancient times. Some would say that it is in our blood. However, the Greek inventor Ctesibius is often considered to be the father of hydraulics (as well as pneumatics). He lived in Alexandria around 300-200 BCE and invented, among other things, the force pump, which is believed to be the first pump based on the principle we now often use in fluid power. The pump was likely used for firefighting purposes [1]. However, it was not until the mid-17th century that Blaise Pascal formulated the concept of pressure, and it took until the late 18th century before we really started to make practical use of Pascal's theories.

In 1795, Joseph Bramah patented the hydraulic press. This can be considered to be the beginning of hydrostatic engineering. In the following years, he built demonstrators to exemplify the potential of the force amplification. These include a device that is surprisingly similar to a crane with one boom [1]. Nevertheless, Bramah was a diligent inventor and filed many patents. For example, in 1812 he registered a patent for hydraulic power networks [2]. The idea was that a main hydraulic power supply, driven by for example a steam engine, could be used to supply machines in a large area with power. Bramah passed away in 1814, but in the second half of the 19th century, many hydraulic power networks were in fact constructed in cities in the United Kingdom. The network provided power for applications such as cranes on quays and theatre stages. One of the most famous hydraulic networks was that built in London, which around 1930 powered 8000 machines via a 300 km pipe network [1]. However, the hydraulic power networks' power distribution shares decreased when the electric grid became popular at the beginning of the 20th century. Nevertheless, the principle of hydraulic power distribution is still around, but instead of having a centralised power source for a city, there are centralised power sources for individual machines.

2.1 Characteristics of Hydraulic Systems

The hydraulic power networks were outcompeted by the electric power networks. To some extent, this caused a brief interruption to the implementation and development of hydraulic technology. However, in the early 20th century William and Janney came up with a hydrostatic transmission based on an axial pump and motor, in which oil was used as fluid. The use of oil meant great performance improvements and hydraulics was on track again.[3]

Today, we may witness a second competition between the electric and hydraulic domains, with the electrification of vehicles as a driving force. However, electrification does not necessarily mean that electric components will replace hydraulic components. What electrification means is that the energy source is electric. The energy can then be transformed via several domains before it reaches the end consumer. Having fewer energy transformations is not necessarily better, even though it is a good rule of thumb. It is all about using the right technology for the right purpose, and hydraulics has several advantages. Among them, the following should be mentioned:

- **Power density**

High pressures mean high power density. High power density has been one of the main driving factors behind the use of hydraulics in mobile applications.

- **Heat transfer**

The energy losses that are turned into heat are transported via the fluid. A central cooling system can therefore be used.

- **Shock absorption and overload properties**

Overload of hydraulic systems is not crucial. Damage is simply avoided by the use of pressure relief valves.

- **Simple and robust**

Even though electronics are being increasingly integrated into hydraulics, hydraulic systems are essentially just well-designed compositions of common alloys.

- **Load holding properties**

A load can be held at rest simply by closing a valve. No power or additional mechanical brake has to be applied to the system to keep it in position.

- **Flexible power distribution**

Power can be transferred via hoses or pipes. This means far simpler installations compared to mechanical linkages. However, electric components also have this simple installation property.

On the downside, hydraulics suffer from noise issues and the risk of external leakage is considered to be a disadvantage as well as the high degree of non-linearity. Also, poor efficiency is often mentioned, even though some hydraulic machines can perform with efficiencies of up to around 95 % [4].

2.2 Mobile vs Industrial Hydraulic Systems

It is common to separate mobile hydraulics from industrial hydraulics. Generalised characteristics of mobile and industrial applications are summarised in Table 2.1. What makes mobile hydraulic systems special is their undefined drive cycle and low degree of customisation. Mobile machines are generally built to be versatile, which makes it hard to make general use cases. These machines are also still operated manually to a very high degree. This means that the drive cycle can be different from one operator to another, even when performing the same task. The fact that the machines should be mobile generally also means restrictions on the weight and size of the components. Furthermore, mobile machines have historically been powered by Internal Combustion Engines (ICEs), which is not the case for industrial applications. Industrial applications generally have a well-defined automated drive cycle. This makes it easier to optimise the system and customised system designs can be highly relevant, since the task is very specific. This statement is supported by the results from a report by Oak Ridge National Labs [5], who found the average efficiency of mobile systems to be 21 % whilst the corresponding number for industrial hydraulics was 50 %.

Table 2.1 *Properties of mobile vs industrial hydraulic systems.*

	Mobile	Industrial
Drive cycle	Unknown	Well defined
Customisation	Low	High
Production numbers	High	Low
Power source	ICE	Electrical grid
Weight & size	Important	Less important

This work is oriented towards mobile applications, but the results are relevant for industrial applications as well.

2.3 Load Handling Applications

This thesis includes the term 'load handling applications' in the title. Here, load handling applications are considered to be a branch within mobile working machines. A mobile working machine is a machine with a propulsion unit which has the main purpose of performing a work process. For load handling

applications, the work process should focus on transporting materials. Typical examples include mobile cranes and earth-moving machinery, such as excavators and wheel loaders. In this thesis, a loader crane is used as a reference machine, but most of the content is relevant to other applications as well.

Mobile working machines are generally very hydraulically intense. The propulsion is often performed via a hydrostatic transmission, and the work process is carried out using hydraulic cylinders and motors. However, it should be noted that even though many machines are similar in principle, there can be great differences in load conditions, as well as in safety standards and regulations [6], [7].

Most mobile machines, including the loader crane, are still manually operated to a high degree. The operator controls the movement of each actuator by adjusting valve positions. However, systems with intelligent features are available and are becoming increasingly popular. If we look at a crane, there are features such as active damping as well as crane tip control, where the operator controls the position of the crane tip rather than the position of each actuator. Features such as automatic folding are also available. However, operators are still used to having much control over the machine and the operation requires practice.

2.4 Mobile Hydraulic System Design

Generally, a mobile machine has several working functions. If, for example, we look at a typical loader crane, we will find a swing, two boom cylinders, telescopic extension cylinders and sometimes additional features. We should also remember the extension legs. All these functions are traditionally powered by an ICE and the power is hydraulically distributed, but there are many different ways of distributing power to the different functions.

Hydraulic systems can be separated into systems with fixed and variable flows. In turn, these systems can either work at a constant pressure level or adapt the pressure level to the load. This classification is illustrated in Fig. 2.1. The figure also shows the throttling losses over the valves for the different system types. As seen in the figure, it is beneficial from an energy perspective to work with both variable flow and pressure levels. However, losses can still be substantial when there are large differences in load pressures. This clearly illustrates the problem of unknown drive cycles.

Nevertheless, within the presented system subclasses there are plenty of possible architectures. In the case of variable flow, the flow is generally controlled by a pressure signal that is fed back to the pump control. Another alternative is to use open loop control and base the flow delivery on the operator's command signal. This is beneficial from an energy perspective, since the pressure difference between the load and the pump will be reduced. Open loop flow control has been a research topic for many years [8]–[10], but it is still not common in commercial applications.

Figure 2.1 Classification of hydraulic systems based on how the pump power adapts to the load condition. In the figure, P_1 and P_2 represent the required load power and P_{loss} is the additional power delivered by the pump. In the case of variable pressure, the pump pressure is slightly above the highest load pressure due to resistance in lines and valves.

Traditionally, the pump runs at a constant speed or at a speed set by a pedal or lever. The flow is then controlled by the displacement setting of the pump, which naturally requires a pump with variable displacement. However, with electrification things might be about to change. The question is whether variable displacement will still be of interest in combination with variable speed.

2.4.1 Open vs Closed Circuits

In systems for working hydraulics, where many functions share one pump, open circuits are generally used. This means that the fluid from the actuators is led back to a reservoir, from which the pump takes fluid for the actuators. The reservoir is often vented, but it can also be pressurised. However, since one side of the pump is connected to the reservoir, the pump is only supposed to pressurise the other side. A general open-circuit layout is shown in Fig. 2.2a.

In a closed circuit, most of the fluid is recirculated from the actuator to the pump, without passing a reservoir. In this type of system, the pump should be able to pressurise both sides. An illustration of a closed circuit is shown in Fig. 2.2b. Closed circuits are commonly used to drive hydraulic motors for propulsion purposes, but also for the actuation of symmetric cylinders. Asymmetric cylinders are more problematic due to non-matching flow rates, but

Figure 2.2 General circuit layouts.

there are solutions. This will be discussed further in chapter 5. Nevertheless, auxiliary components are required for sufficient cooling, overload protection, filtering and external leakage compensation.

2.4.2 Load-holding Functionality

The purpose of load-holding valves is to avoid unintentional load movements, and these are installed in many load handling systems. A typical load-holding valve is shown in Fig. 2.3. The valves include several features and can contribute to both improved functional properties and improved safety. The tasks of a load holding valve can be summarised as follows:

- **To hold a load**
The leakage in the valve is negligible and a lifted load will be held in position until it is actuated.
- **To protect against overrunning loads**
The valve ensures that the pressure on the low-pressure side does not become too low and the valve thereby ensures that cavitation will not occur.
- **Hose failure protection**
The valves should be positioned next to the actuator. If a hose breaks, the flow rate from the actuator will be limited since it must pass a restrictor.

These features are in many ways desirable. However, the functionality basically implies that a load will be lowered by throttling, which means losses. In fact, the pump must provide power to the actuator to lower it, even though potential energy could have been harvested. To be able to harvest energy, which

is relevant for electrically powered systems, less conventional load-holding solutions must be considered. Electrically actuated valves instead of hydraulically actuated valves is one solution.

Figure 2.3 *Illustration of a typical load-holding valve. When the cylinder is moving upwards, the flow passes through the check valve. When a downwards movement is requested, pressure has to build up in the upper chamber before the compensator valve opens and the movement can start. Note that the directional control valve between the pump and actuator is omitted in the figure.*

3

Hydraulic Machines

A hydraulic machine refers to either a pump or a motor. A pump transforms mechanical energy into hydraulic energy, and a motor does the opposite. Many machines are capable of both pumping and motoring operation. They might, however, be optimised for one or the other. There are many different kinds of hydraulic machines, but in fluid power - where high pressures are desired - it is almost exclusively positive displacement pumps that are used, and not rotodynamic machines such as centrifugal pumps. Therefore, the focus here is exclusively on positive displacement machines.

3.1 Positive Displacement Machines

A positive displacement machine is a machine that takes a fixed volume of fluid from one place and moves it to another. The oldest known positive displacement machine is the Archimedean screw, which was not in fact invented by Archimedes [11]. This machine dates back to Ancient Egypt and was used to transport water from the River Nile. The pump was basically a single screw in a cylinder. However, about 2300 years ago Ctesibius came up with a different type of positive displacement pump, which is more similar to the ones used in fluid power today. This was the force pump, which was based on pistons sliding in cylinders. It made use of a kind of check valve to direct the flow. This pump was powered by a lever. However, variants with rotational inputs were soon produced. An example is Heron's wind-powered organ [12].

In 1588, Ramelli released a book describing different machines that could be used for e.g. welling applications [13]. The book contains illustrations of screw pump configurations, different piston pump arrangements and use cases for vane pumps. This indicates that there has been some development in the hydraulic field since Ctesibius' time, but applications were still limited to water transportation and more development might have been expected in almost 2000 years. Nevertheless, in around 1600, a man named Johannes Kepler invented

Figure 3.1 Pump classification.

a gear pump to be used for fountains. The pump worked, but did not revolutionise society at the time. Kepler moved on and became famous for something else. However, others continued to develop hydraulic machines and today there are many types of hydraulic pumps and motors. A classification of pumps is shown in Fig. 3.1. The pumps are categorised as either rotational pumps or piston pumps. In rotational pumps, the volume displacement is achieved by a rotating motion whilst the volume displacement comes from a translating motion, i.e. a moving piston, in a piston machine. Piston machines can generally work with high efficiency at high pressures because of their comparatively small sealing surface. However, gear and vane pumps are very common due to their simple construction, compact design, and low flow pulsation properties. Nevertheless, the focus in this thesis is on piston pumps.

Even though pistons have a translating motion, a rotational input to the machine is desired for most applications. Rotational piston machines are then classified according to how the pistons move in relation to the rotational shaft, which can be in a radial or an axial direction. For radial piston pumps, the pistons can have mechanical support from either the inside or the outside, which means either internal or external support. Axial piston pumps are divided into swashplate pumps and bent-axis pumps. Swashplate pumps include pumps where the cylinders rotate with the shaft, which is the most common solution, as

well as pumps where the swashplate is rotated instead. The wobble plate pump, which will be further described in chapter 6, belongs to the latter category.

The conventional swashplate pump was first patented in the US in 1893 [14], and most swashplate pumps are still very similar to this one. However, within this category there are new pumps that are principally different from the conventional solution, such as the floating cup [15], which was recently commercialised [4], and the floating piston pump [16]. This indicates on that there are plenty of possible technological realisations within each pump category and also that there still is room for innovations.

3.2 Commutation Techniques

The classification above does not consider commutation methods. That is, however, an important aspect in the pump design. Commutation refers to switching from high to low pressure or vice versa for an individual cylinder. To achieve operation that is as smooth as possible, the cylinder pressure should match the inlet or outlet pressure during commutation. There are three types of commutation techniques for piston pumps that principally can be applied to all pump classes above. These are:

- Rotation-dependent
- Pressure-dependent
- Actively controlled

Examples of these are shown in Fig. 3.2. Rotation-dependent commutation means that the timing is defined by fixed mechanical connections. An example of this is the valve plate (sometimes called the port plate), which is shown in Fig. 3.2a. This technique is preferably used when the pistons are rotating with the shaft. The problem with this technique is to match the pressure in the cylinder with the system pressure during opening. Pressure-relief grooves are generally used to achieve smooth commutations for a large pressure range.

Fig. 3.2b shows an example of pressure-dependent commutation, where a check valve determines the commutation timing. This means that optimal pressure matching is always achieved. However, note that a pump with check valves cannot run as a motor and also that the flow direction is independent of the shaft's rotational direction. This commutation technique is mainly relevant for pumps where the pistons are not rotating with the drive shaft, which includes radial piston pumps with internal support and axial pumps with a rotating swashplate. However, solutions with inbuilt check valves in the valve plate have been investigated [17], and such a design would also be classified as pressure-dependent.

The check valves can be replaced with actively controlled valves, and actively controlled commutation will then be achieved. This is what is done in machines

Figure 3.2 Two possible valve configurations for a digital pump with two pump elements.

based on Digital Displacement® (DD) technology, which will be described further in section 3.4. However, another type of active control can be achieved by having a rotatable valve plate [V], [17].

3.3 Operating Modes

In section 3.1, pumps were classified by their design. However, a functional classification is often more relevant. From a functional perspective, a hydraulic machine has four driving modes since both the flow direction and the pressure difference over the machine can be varied. Therefore, it is common to divide the operation modes into quadrants, as in Fig. 3.3.

The flow direction can be changed by either changing the sign of the dis-

Figure 3.3 Driving modes in quadrant division.

Figure 3.4 *Driving modes in octant division.*

placement setting or the rotational speed. Therefore, there are in fact eight different driving modes, which are graphically represented in Fig. 3.4. However, it is hard to think of applications where it would make sense to be able to change both the sign of the displacement and the speed, since these cancel each other out. Therefore, the quadrant representation is sufficient, but with either displacement setting or rotational speed as a replacement for the flow direction. Traditionally, the displacement setting is used to define the flow direction since it has not been relevant to change rotational speed in machines driven by combustion engines. But again, electrification changes the conditions.

Machines working in open circuits are generally only working in one quadrant (and exception is secondary controlled systems) whilst multiple quadrant operation is more relevant for closed circuits.

3.4 Displacement Control

In conventional hydraulic systems, where the pump speed is more or less fixed, variable flow can be achieved by using variable displacement pumps. There are many possible ways to control the displacement. For piston pumps, the control possibilities can be classified as in Fig. 3.5.

The control can be on either piston or pump level. On piston level, one can either control the stroke length or the usage of the stroke. The conventional method is to vary the stroke length. In a swashplate pump, this is done by changing the swashplate angle. However, there are also novel methods such as the 'adjustable linkage' developed at the University of Minnesota [18]. This method is claimed to be efficient at part displacement since the pump does not rely on hydro-dynamic bearings to the same extent as conventional pumps do. When it comes to varying the usage of the stroke lengths, there are sev-

Figure 3.5 Displacement control classification.

eral proposed methods. The most well-known is probably the DD technology, where each piston is individually controlled. This technology enables both part-stroke control and discrete control where an arbitrary number of pistons can be turned off for each revolution. The DD technology is further described in section 3.4.2. Nevertheless, the usage of the stroke can also be controlled by means of a rotatable valve plate [V] alternatively by using mechanically or hydraulically manipulated opening of check valves in e.g. WPPs [19]. Another innovative solution is the 'alternating flow hydraulic pump', presented in [20]. The alternating flow hydraulic pump has pistons connected in pairs and the usage of the stroke (i.e. the displacement setting) is controlled by the phase angle between the pistons. The phase angle is controlled by the pump case angle.

Control performed on pump level is based on bypassing fluid. This means that the pump flow is either directed to the reservoir (i.e. bypassing the flow) or to the system. There are two principally different approaches. One is to bypass the entire pump flow and match the flow rate by means of Pulse-width Modulation (PWM) control. This requires a fast switching valve and the technique has been studied in e.g. [21]–[23]. It is sometimes referred to as virtually variable displacement. This valve control is analogous to how transistors are controlled in electric circuits. However, hydraulic valves have switching times in the range of milliseconds, which is considered as ages from a transistor's perspective. Slow switching means considerable throttling losses. The success for virtually variable displacement is therefore limited to date.

The other bypass-technique is based on dividing the pump into groups, called pump elements. The pump elements can be groups of pistons in a pump or individual pumps connected to the same shaft. The flow from each pump element can be directed in different ways. This control implies a discrete number of displacement settings and is sometimes referred to as 'digital pump control'.

It is the type of control that has been investigated in Papers [II, III] and it will be further described in chapter 6.

Today, it is almost exclusively conventional stroke length control that is used, even though Digital Displacement Pumps (DDPs), are approaching the commercial market. The conventional control and the DD technology are described in more detail here below.

3.4.1 Conventional Control

In most variable-pump applications, the displacement setting is varied by changing the stroke length of the pistons. The stroke length is generally continuously variable and controlled with either a hydro-mechanical or electro-hydraulic controller. Due to the high internal forces in the pump, purely electrical solutions are rare. However, there are still many different kinds of controllers and the best controller choice is highly dependent on the application.

Fig. 3.6 shows how the stroke length can be varied on a swashplate pump by changing the angle of the swashplate. The main differences between different stroke controllers is how the pressure in the control piston is varied. Many controllers use the internal pump pressure for control. Therefore, a bias spring is required to ensure that pressure can be built up. This also means that the pump requires a minimum working pressure level, typically around 2-3 MPa. However, note that pumps used in closed-circuit applications generally use their charge pressure for displacement control.

The response of hydraulically actuated controllers is limited by the flow provided to the control piston and is therefore pressure-dependent. In practice, the bandwidth does usually not exceed 25 Hz [24].

Hydro-mechanical Controllers

In a hydro-mechanical controller, pressure signals are used to control the displacement setting. The controllers are usually pilot operated, which means that pressure signals control the position of a controller valve, which in turn controls the pressure on the control piston. A typical hydro-mechanical pressure controlled controller is shown in Fig. 3.7.

It can be seen how the control valve, often called the pressure compensator, compares the pump pressure with the spring setting and the tank pressure. This means that the reference pressure is set by adjusting the pre-compression of the spring. When the pressure is too low, the valve will move towards the right which in turn causes increased displacement. Analogously, when the pressure becomes too high, the valve will move towards the left and the displacement will decrease. Another important component in the controller is the orifice that connects the tank with the control piston. The task of this orifice is to increase the damping in the pressure control loop. A large opening area means high damping, which is preferable. However, a large opening area also means significant flow through the orifice and this implies high losses. In some

Figure 3.6 Example of swashplate control. Often, there is also a bias piston that is working against the control piston.

Figure 3.7 *Hydro-mechanical pressure control.*

controllers the orifice is adjustable, while in others it is not. Nevertheless, there are more control possibilities than just the pressure control described above. Common hydro-mechanical controller types are:

- **Pressure control**

The compensator strives to keep the pump pressure constant. The reference pressure is set by a spring in the controller. This type of controller is shown in Fig. 3.7.

- **Load sensing control**

A load sensing compensator strives to keep the pressure drop over an external orifice (usually a control valve) constant. This can be regarded as a special case of pressure control.

- **Flow control**

The controller aims to keep the flow constant independent of the drive speed. Similarly to load sensing control, flow controllers strive to keep the pressure drop over an orifice constant.

- **Torque limit control**

This type of controller limits the maximum pump torque by reducing the displacement when the pressure increases. The control is sometimes called power control which makes sense for constant speed applications.

- **Dual-displacement control**

The displacement setting can be varied between two pre-set values. If a pressure signal crosses a limit value, the displacement will change. Another name for this control is two-point control.

These are just examples of basic control principles. There is a wide variety within these types and they are often combined. A load sensing compensator is, for example, generally used together with a pressure compensator. This means many different products in the catalogue.

Electro-hydraulic Controllers

Hydro-mechanical controllers are useful because they are simple and robust. However, their flexibility is very limited and stability issues are common. Problems with hydro-mechanical controllers are discussed in [8], and the main disadvantages can be summarised as followings.

- Limited signal filtering possibilities
- High losses due to damping orifices
- Temperature-dependent characteristics
- Time delays due to pressure build up in hoses

Figure 3.8 *Electro-hydraulic controller.*

- Difficulties with automating processes
- Bulkiness of signal lines

Electric control can eliminate most of these problems, which was concluded in 1990 when an electrical version of a conversional load sensing system was presented [25]. Furthermore, electric control offers new control possibilities such as flow-on-demand systems, where the pump displacement is based on the operator command rather than pressure signals. This can increase energy efficiency compared to load sensing systems, since pressure drops are reduced [9], [26]. Furthermore, since the control is carried out via software, parameters can be tuned on-the-fly and the functionality of the controller can basically be changed without having to change any physical parts. This means that one electrically controlled controller can replace several other hydro-mechanical controllers. Nevertheless, bear in mind that the features that can be achieved with an electric displacement control can principally also be achieved with a variable-speed drive.

Electro-hydraulic controllers work very similarly to hydro-mechanical controllers. The difference is basically that the control valve is electrically actuated. Naturally, there are also many variations of control valves and circuit layouts here. There are simple alternatives with dual-displacement control and advanced multi-stage servo controllers. However, a typical controller is shown in Fig. 3.8. Note that the displacement angle is not measured in most mobile controllers. Instead, mechanical feedback is used to achieve control precision. This also applies to hydro-mechanical actuators.

Electro-mechanical Controllers

There are commercial swashplate controllers that do not rely on hydraulics [27]. The displacement can be controlled by an electric motor that drives a ball

screw, which in turn is connected to the swashplate. This control is, however, very slow and bulky compared to the abovementioned alternatives.

Manually Controlled Controllers

The controllers described above are controlled by either a pressure signal or an electric signal. However, there are also controllers where the control signal is given by the operator, for example by means of a lever connected to a control valve. In other machines, the displacement can be controlled by a hand crank mechanism [3]. However, these solutions are not of great interest in modern applications, where automation is desired.

3.4.2 Digital Displacement

A Digital Displacement Machine (DDM) is a machine in which each piston is individually controlled. This control relies on controlling the commutation rather than the stroke length, which the previously described controllers do. By using active valves, a piston can be deactivated by simply keeping it connected to the reservoir. When each piston can be controlled for every revolution, the resolution of the discrete control is very high.

Swashplate machines with an adjustable swashplate angle have been around for more than 100 years. DDMs are more recent. Their history dates back to the 80's. In 1989, Salter and Rampen filed a patent regarding electronically controlled poppet valves for the inlet in multi-piston pumps [28]. They mention that mechanically controlled valves had been proposed before, but that such systems are slow and noisy. They consider their electronic solution to be more suitable. In the application, they propose a solenoid-based valve design and different control strategies (flow and pressure control), where the controller decides the number of active pistons for each rotation. An accumulator is included in the design to compensate for the pulsating characteristics.

The following year, in 1990, they filed a new patent application [29]. The application related to a machine similar to the previous one, but that can work as both a pump and a motor since both the inlet and outlets are electrically controlled. In their application, they clarify that the valve timing is of great importance for the motor mode since large forces are required if the movement is not performed when the pressures on both sides of the valve are equal.

The story continued in 1992, when Rampen presented his PhD thesis [30]. There, he explicates the ideas above. The focus is, however, on the pump mode and the development of a prototype, based on a WPP. In the end of the thesis, he pinpoints required future work on valve characteristics and noise. Since then, many people have been working on that.

The work on the DD technology resulted in the formation of the company Artemis Intelligent Power (AIP) in 1994 [31]. Since then, they have worked on developing the technology and have been granted many patents along the way, from detailed control [32] to system implementations [33]. However, DD

technology has not been neglected by academia. The technology has been explored at Purdue [34]–[36], in Aalborg [37], [38], in Minnesota [39] and in Tampere [40], [41], for example.

AIP have implemented their technology in many different applications. The DD technology is for example used in a wind power plant in the megawatt scale. However, in 2016, they implemented a DDP in a full size excavator with substantial energy savings as a result [42]. AIP are now, together with Danfoss Power Solutions, working to commercialise the pump [31]. The pump uses active valves for the inlet and passive valves for the outlet. The main arguments for the pump are:

- **High efficiency at part displacement**
The conventional controller losses are eliminated.
- **Pressure independent control**
In conventional controllers, the displacement is dependent on the pressure since the pressure determines the flow rate to the control piston. This dependency is eliminated. Furthermore, the requirement for a minimum pressure level for control is also eliminated.
- **Fast response**
The displacement can be changed within tens of milliseconds (depending on the rotational speed) [43].
- **No hysteresis**
Conventional controllers often suffer from hysteresis, which the DDP does not.

Furthermore, the advantages of electro-hydraulic control also apply to this control method. The disadvantage is the high degree of complexity, regarding both control and physical implementation.

3.5 Losses in Hydraulic Machines

In simple terms, there are leakage losses and there are frictional losses, and these counteract each other in many ways. Leakage losses come from gaps between sliding surfaces. Large gaps reduce the friction but increase the leakage. Furthermore, the viscosity, which is temperature dependant, has a big influence here. This means that the performance of a hydraulic machine will vary during operation. Also, the rotational speed has a high impact on frictional losses, but also on wear. Speeds that are too low mean that lubrication films cannot be established properly and dry or mixed lubrication occurs. To avoid this, many pumps have a minimum speed, typically around 500 rpm. This is a huge limitation when variable-speed drives are used.

Nevertheless, there are also losses due to compression of the oil, churning losses, bearing losses and internal fluid friction.

3.5.1 Loss Models

Throughout the years, many models of different complexity have been presented. There are simple physical models and models where each interface is considered. However, the losses are generally divided into flow losses and torque losses. An overview of different common models is presented in [44]. From this overview, it can be seen that losses roughly can be scaled by the displacement to some extent, which is convenient during component sizing in system design.

3.5.2 Efficiency Definitions

The total efficiency of hydraulic machines is often divided into hydro-mechanical and volumetric efficiency. Hydro-mechanical efficiency is supposed to represent frictional losses and volumetric efficiency should correspond to leakage and compression losses. There are ISO standards in which these efficiencies are defined, but hydro-mechanical efficiency is in fact only defined for motors and not pumps. Nevertheless, there is an ongoing discussion on how the efficiency and loss definitions for hydraulic machines should be defined and how measurements should be conducted [45]–[47].

Today's ISO standards for measuring the performance of positive displacement machines are described in ISO 4409:2019 [48]. The procedure is described for both pumps and motors. For pumps, the total efficiency is defined as:

$$\eta_{tot} = \frac{q_{out}p_{out} - q_{in}p_{in}}{2\pi\omega T}. \quad (3.1)$$

Furthermore, the volumetric efficiency is defined as:

$$\eta_v = \frac{q_{out}}{D\omega} \quad (3.2)$$

D is the pump's displacement. According to ISO 8426:2008 [49], the displacement is experimentally measured. The principle of the displacement measurement is to measure the flow for different pressure levels and thereafter linearly extrapolate the results to zero pressure differential over the pump. The accuracy of this method is, however, disputed [46].

As mentioned, there is no statement on hydro-mechanical efficiency for pumps. However, it is common to regard the difference between total losses and the volumetric losses as hydro-mechanical losses, which means:

$$\eta_{tot} = \eta_v\eta_{hm} \quad (3.3)$$

This is, however, a ventured assumption since it means that all losses are strictly volumetric or hydro-mechanical and also that they are appearing in series.

Figure 3.9 *Simplified representation of dead volumes.*

Critical Views on the Standards

One of the problems with the above definitions is that compression of the fluid (i.e. increase of internal fluid energy) will be regarded as a loss. This means that the volumetric efficiency cannot reach 100 %, even without any leakage, since the volumetric outlet flow will always be smaller than the inlet flow for the same mass flow (there are examples with volumetric efficiencies greater than 100 %, but this is due to the problems with measuring the displacement [46]). By not considering the internal fluid energy, the pump losses are overestimated and motor losses are underestimated. To come away from this problem, a new definition of total efficiency is proposed in [45]. A factor that compensates for compression is introduced in that definition. This factor is dependent on the pressure differential, the bulk modulus and the ratio between the dead volume within the pump and the displacement volume. A representation of the dead volume and the displacement volume is shown in Fig. 3.9. The proposed definition also suggests to differentiate between volumetric and hydro-mechanical losses based on hydro-mechanical efficiency rather than the volumetric.

This approach is thermodynamically motivated, and it gives a better view of the actual losses in the system compared to the ISO standards. However, the dependency on the dead-volume ratio is problematic since it can be hard to determine experimentally. Furthermore, engineers are used to using the volumetric efficiency to calculate flow rates rather than power losses. Therefore, the question is: What should the volumetric efficiency be used for? If one is interested in a factor that describes the delivered flow rate, the current definition is appropriate. However, if one wants to describe and segregate losses, something similar to the newly proposed method, where the theoretical displacement, D , is pressure-dependent, is preferable. This becomes evident when measuring the efficiencies of pumps with pressure-dependent or active commutation, since these can achieve hydro-mechanical efficiencies over 100 %

[50]. This is due to the fact that the compression of the dead volume will be counted as a volumetric loss during upstroke, but this volume will then help to drive the pump during downstroke. This behaviour has also been pinpointed by Caldwell [51].

3.5.3 Controller Losses

A type of loss that is often overlooked is controller losses. The standards do not describe how the controller should be considered in the measurements. As seen earlier, most controllers require a bypass flow to achieve satisfactory damping performance and for cavitation-related reasons [52]. This introduces losses, and these losses make the efficiency of the machines poor at low displacement settings. For example, the required bypass flow can account for as much as 19 % of the total outlet flow at 50 % displacement [52].

The literature focusing on controller losses is limited but in [53], it was shown that controller losses account for a considerable share of total losses, especially in part load conditions. They compared measurement results with simulations and concluded that the damping orifice is the major loss contributor. However, they also stated that an electrically controlled controller could reduce the controller losses by 30-60 % since the damping orifice can be smaller or even removed.

Nevertheless, there are more downsides related to the conventional swashplate control. Problems regarding swashplate oscillations have been reported [52], [54]. Here, it is claimed that measurements performed with mechanical end stops for the controller can be misleading because this prevents the swashplate from moving, as it would in a real situation. In a real situation, oscillations arise due to varying forces on the swashplate and these oscillations will cause losses [55].

Furthermore, losses related to the actuation of solenoids in electrically controlled controllers can also be regarded as losses related to the machine. They might be small, but for a complete analysis they should be included. One problem is, however, that they cannot be categorised as hydro-mechanical or volumetric losses. A third component is therefore necessary.

3.6 Noise

Many studies on how noise pollution affects human health have been conducted. The focus is often on noise from traffic. However, it is clear that noise pollution can cause non-auditory effects on health, such as sleep disturbance, heart disease and cognitive impairment, as well as auditory effects on health, such as hearing impairment and tinnitus [56], [57]. Noise emissions are therefore an important parameter when designing hydraulic systems, and not at least the pump, which in many ways can be considered as the source of noise.

(a) Equal-loudness contours according to ISO 226:2003 [58]. This shows that we are most sensitive to frequencies in the region 2-5 kHz, but our hearing range spans between 20 and 20 000 Hz.

(b) Weighting filters used to compensate for the equal-loudness curve in measurements. Different filters are used depending on noise level, but A-filters are most commonly used.

Figure 3.10 Perceived noise and compensating filters.

How we perceive noise is individual, and we also perceive noise differently depending on whether or not we are prepared. Furthermore, the perceived loudness is dependent not only on the sound pressure level but also on the frequency, which can be seen in Fig. 3.10a. To compensate for this, weighting filters are often used when measuring noise levels. The choice of filter depends on the noise level. The most common filters are shown in Fig. 3.10b. However, there are many other effects that make the analysis of noise perception complex, such as the masking effect, which means that a frequency can suppress other adjacent frequencies, and the harmonisation of frequencies. Nevertheless, a low noise level can always be considered preferable.

3.6.1 Noise From Hydraulic Machines

It is common to separate the sources of noise into fluid-borne and structure-borne noise. Fluid-borne noise has its origin in flow variations whilst structure-borne noise comes from force variations within the pump.

Fluid Bourne Noise

As mentioned, fluid-borne noise comes from flow variations. The flow variations cause pressure variations, which make components in the system vibrate. This in turn causes airborne noise that will reach our eardrums. This means that the noise caused by the pump can appear in a completely different part of the hydraulic system if the system has low damping. This also means that the properties of the system are important for the amplification of the pump noise. However, when analysing pump noise, it is desirable to isolate the pump from the system to make the results general. In a lab environment there are several ways of dealing with this, but one of the simplest is the anechoic method in which pressure wave reflections are rejected using an orifice. For details about this and other methods, see e.g. [59].

Nevertheless, pressure variations arise due to non-continuous flow provided by the pump. Note that screw pumps do not suffer from these flow variations. However, they are rarely used in fluid power systems due to their poor efficiency [3].

The flow pulsations that cause the fluid-borne noise are usually separated into two parts: a kinematic and a compressible part [60]. The kinematic part is strictly geometrical and the compressible part accounts for the disruptions in the commutation stages. This division is usually done for machines with rotation-dependent commutation. In such machines, the pre-compression angle is geometrically defined which means that it will affect the kinematic flow pulsations. However, for pressure-dependent commutation, the pre-compression is not geometrically defined which means that the pre-compression should be added to the compressible part, according to the above definition.

Nevertheless, to reduce kinematic pulsations, it is intuitive to increase the number of displacement elements (i.e. the pistons in a piston machine). However, note that an odd number will give lower kinematic flow pulsations than an adjacent even number, which can be seen in Fig. 3.11. Regarding the compressible pulsation amplitudes, their amplitude can be minimised by making sure that the pressure levels in the system and the cylinders are matched during commutation. This minimisation is always ensured for pressure-dependent commutation techniques, and the pressure matching can also be realised in actively controlled machines. However, for rotation-dependent machines the compressible flow is more problematic since the cylinder pressure during commutation is principally defined by the pre- and de-compression angles. Pressure-relief grooves are generally used to smoothen the commutation.

Structure Bourne Noise

Structure-borne noise comes from variations in forces within the pump. This makes the pump structure and its attachment vibrate, causing airborne noise. The force variations have their origin in the pressurisation of displacement elements. Note that an even number of pistons is advantageous from a force

Figure 3.11 Kinematic flow for odd and even number of pistons.

variation point of view. However, it is not only the force that matters but also the moments, and the moments are larger for evenly numbered configurations [61].

Both fluid- and structure-borne noise are important. It is also clear that the system in which the pump is working has a great influence of the amplification of the noise, so the most important source cannot be stated without having the system at hand.

Noise and Variable Speed

In traditional systems where the pump speed is pre-set, the fundamental frequency from the flow pulsations is well defined. The hydraulic system and the mechanical structure can then be designed with this frequency in mind, and undesired resonance can be avoided. However, the fundamental frequency varies when using a variable-speed drive, and the risk of excitation and undesired modal oscillations is increased. It is, however, likely that an operator will notice this and then change the flow demand by intuition. Still, it is undesired.

Nevertheless, speed-controlled systems are generally perceived as quiet compared to displacement-controlled systems, since the noise from the prime mover is significantly reduced at no load or part flow delivery.

Cavitation

Another noise-related phenomenon is cavitation. That happens when the static pressure level becomes too low and causes a high frequency sound. This is, however, generally a consequence of a poorly designed system and can therefore be avoided. Cavitation is also undesired since it can damage components.

4

Electrification of Mobile Machines

Electromobility is nothing new. Electric cars have existed since the 19th century, even though they were soon outcompeted by the combustion-driven alternatives. However, we are now witnessing a turning point. And electrification is affecting not only cars but all kinds of vehicles – from baby strollers to wheel loaders.

For most vehicles, this electrification is mainly driven by environmental arguments and emission regulations. Still, electrification means more than just a substitution of power generation. It can add additional values to the product, and this applies particularly to mobile working machines. When a combustion engine is replaced by an electric alternative, the local emissions from the machine are more or less eliminated. This means that the machine can work in places where it has not been allowed to before, such as indoor environments and poorly vented areas. Furthermore, the noise from an electric drive is lower than from a combustion engine, which means improved comfort, but can also result in opportunities for extended working hours and operation in highly populated areas.

4.1 Commercial Trends for Mobile Machines

Different combinations of the words 'first' and 'electric' have often appeared in press releases from manufacturers and trade magazines in recent years. There are hybrid solutions and battery-powered fully electric alternatives. Mini excavators in sizes below 4 tonnes account for a large proportion of the latter, but wheel loaders and backhoes are also included. Nevertheless, the general trend for these machines is to use one electric motor for the propulsion and one for the hydraulics. Apart from that, these machines are similar to their combustion-driven counterparts. However, their low noise level is a frequently mentioned

advantage as are the reduced operating costs. Case claims that operating costs have been reduced by 90 % with its new 580 EV backhoe [62]. This is partly due to reduced energy consumption and partly due to reduced maintenance. The trend indicates an optimistic view of electrification for smaller machines, and Volvo CE has stated that it will phase out combustion-driven machines in those sizes where it offers electric alternatives, starting this year [63].

There are also examples of full-electric alternatives for mid-size and large machines. One example is the 12-tonne wheeled excavator from Mecalac [64]. Larger electric machines are available from Epiroc, which is offering battery-driven underground loaders and mining trucks [65]. The loaders can carry the Mecalac excavator in their buckets. There are also fully electric solutions for surface-mining. One example is the 210-tonne excavator from Liebherr [66]. This is, however, powered by cable.

Among mid and large-size machines, hybrid solutions are dominant. Komatsu released a mid-size excavator with electric swing already in 2008 [67]. This machine used capacitors instead of batteries. Huddig has released the Tigon, which is a large articulated backhoe that can run on diesel, electricity, or both [68]. Volvo CE has been working on the prototype LX1, in which hybrid technology has made downsizing possible and reduced fuel consumption by 50 % [69]. Hybrid solutions can also be found among large machines in the mining sector [66].

Heavy road trucks are also being electrified. In 2016 Mercedes-Benz presented its Urban eTruck concept [70], and since 2018 has been testing its eActros in the field [71]. One of Mercedes' competitors is Volvo, and in 2019 Volvo opened its order books for the electric versions of its FL and FE models [72]. Furthermore, crane manufacturers are striving towards electrification, and for example Hiab offers electric power pack solutions which can be used to power its cranes [73].

These are just a few examples from a long list of electrified mobile machines. However, they show the general trend.

4.2 Electric Machines

Electric machines convert power from the mechanical domain (most often the rotational) to the electric, or vice versa. When the power is converted from mechanic to electric, the machine is working as a generator. When the power flow is in the other direction, the machine is working as a motor. Most electric machines can perform both tasks.

There are many types of electric machines to choose from and, as always, they all have their pros and cons. However, in medium to high power applications, Alternating Current (AC) machines are most often used. AC machines can be either synchronous or asynchronous. Asynchronous machines are induction machines and they are most often of squirrel cage type. They are very simple and reliable. They have therefore been very common in industrial applications

in the last century and they remain so, but they have also found a place among electric vehicles. However, in highly demanding mobile applications, there is a trend towards synchronous machines, and in particular PMSMs [74], [75]. The advantages of these machines include high efficiency, high power density and low inertia [76].

4.3 Permanent Magnet Synchronous Machines

PMSMs are synchronous AC machines in which the rotor does not need external excitation. It might not come as a surprise that there are many variants of these. However, they are all based on a rotor with permanent magnets and a stator with windings. AC current in the stator creates a rotating magnetic field that interacts with the permanent magnet field. Thereby, the rotor speed is controlled by the frequency of the rotating field, without any slip. The general construction of a PMSM is illustrated in Fig. 4.1.

Figure 4.1 General construction of a PMSM. A three-phase AC current in the stator windings creates a rotating magnetic field that interacts with the permanent magnet's field.

4.3.1 Losses

The low losses are a major advantage of PMSMs. One of the reasons behind the high efficiency is that rotor flux is obtained without power losses, which is not the case for induction machines. This also means that cooling is easier, and the machines can therefore allow a larger current density in the stator. This is a reason for the higher power density. A permanent magnet motor is typically 20-30 % smaller than an induction machine with equivalent performance. [76]

A PMSM can cover large operating areas with efficiencies over 95 % [74]. However, there are losses and they are usually categorised as:

- **Copper losses**

These are losses due to resistance in the stator windings. They account for a large part of the losses and are proportional to the square of the torque. However, they also have a temperature dependency due to the fact that resistance increases with temperature.

- **Iron losses**

Iron losses are sometimes also called core losses. It is common to divide the iron losses into hysteresis losses and eddy current losses. Eddy current losses are said to be proportional to the square of the frequency whilst hysteresis losses are proportional to the frequency [76]. However, this division is disputed and it is argued that these losses have the same origin. A common approach is therefore to use variants of the empirical Steinmetz equation instead. When doing so, the iron losses are proportional to the frequency to the power of 1-3. A short summary of iron loss models can be found in [77].

- **Mechanical losses**

These losses are represented by friction in bearings and but also air resistance. They are, however, generally relatively small.

Sometimes, stray load losses are also mentioned. They are load-dependent losses that appear at various places in the machine. Nevertheless, the total losses are highly dependent on the winding configuration and cooling properties as well as number of poles and magnetic circuit design. Loss modelling is therefore complex and finite element methods are often applied. However, the efficiency is generally worst at low speed rates and high torques, since this means low output power but high copper losses. On the other hand, mechanical losses and iron losses generally increase with the speed. Still, the sweet spot is typically at high speeds and low to medium torque levels.

High losses mean high temperatures. To avoid overheating, electric motors are not allowed to run too much at high torque. The working region is therefore divided into an intermittent and a continuous region. As a rule of thumb, the root mean square of the drive cycle should be within the continuous region [74]. However, the division of regions is highly dependent on the cooling capacity.

4.4 Frequency Control

In AC machines, the rotational speed is controlled by the frequency and the frequency is controlled by an inverter. The inverter transforms Direct Current (DC) power from e.g. a battery into AC power at a demanded frequency and current levels corresponding to the required torque. It basically consists of PWM-controlled transistor power switches, which is illustrated in Fig. 4.2.

Inverters can reach efficiencies of 97 % or even higher, but the efficiency drops with decreased power usage [78]. The losses in an inverter are mainly switching

Figure 4.2 General construction of an inverter.

Figure 4.3 Overview of FOC for a PMSM.

and conduction losses. However, proper inverter control is important for the efficiency of the system. A common control approach is Field-oriented Control (FOC), which is a method where the phase currents are transformed into two vectors that are rotating with the rotor. The transformation decouples the currents into a torque-producing component, i_q , and a magnetising component, i_d . This is advantageous since the produced torque can be maximised with minimum input power. The general control structure is shown in Fig. 4.3.

4.4.1 Flux Weakening

The rotating magnetic field induces an electromotive voltage, which is proportional to the speed. This opposes the current in the stator windings, and this voltage is not allowed to exceed the phase voltage. This means that the maximum speed is limited by the electromotive voltage. However, the voltage can be reduced by decreasing the magnetic flux. This reduction is called flux weakening, and is achieved by applying current to the magnetisation component, i_d . However, since current on this component does not result in any torque, flux weakening leads to reduced efficiency. It is therefore reasonable to avoid extensive operation in the flux weakening region. The rated speed of a synchronous motor determines where flux weakening starts.

4.5 Electric Motors vs Diesel Engines

Apart from having the ability to make things rotate, there are not many similarities between electric motors and combustion engines. Therefore, it is not optimal to perform electrification by simply replacing the diesel engine with an electric motor. If comparing a synchronous motor with a diesel engine, the following differences should be pinpointed:

- **Multi-mode operation**

An electric motor can run in both directions and has the ability to run as a generator, whilst a diesel engine is not designed for bi-directional operation. It would also have a hard time trying to reproduce diesel from mechanical input power.

- **Minimum speed**

The diesel engine has an idle speed which must be exceeded, whilst the PMSM can go to all the way zero speed (and below). This means that it only has to run when required and this is one of the reasons why electric systems are considered to be quiet.

- **Efficiency**

The efficiency of a diesel engine generally peaks around 40 % whilst synchronous motors together with inverters cover very large operating areas with 90-95 % efficiency.

- **Response**

The synchronous motor has a faster response than a diesel engine.

- **Torque curve**

Unlike a diesel engine, a synchronous motor can provide constant torque, independent of the speed.

- **Complexity**

The electric motor consists of considerably fewer moving parts, which means less maintenance. It also offers a greater installation freedom.

- **Air pollution**

There is no local pollution from electric machines.

With these aspects in mind, the PMSM has a great advantage over the diesel engine and this also applies to most other electric motors. The benefits allow for new system architectures and improved performance. However, the disadvantage is energy storage, since batteries are more expensive and bulkier than a fuel tank.

5

Pump-controlled Systems

To obtain system efficiency that is as high as possible, throttling losses due to different pressure levels on simultaneously driven loads must be eliminated. For rotational actuators, i.e. motors, this is not a major problem since variable displacement can be used. However, the situation is different for linear actuators. Multi-chamber cylinders are one solution and hydraulic transformers are another. Such solutions are evaluated in e.g. [79]. However, the easiest solution might be to separate the loads. It is then time to consider pump-controlled systems.

In a pump-controlled system, each actuator has its own drive unit and the pump flow determines the actuator speed. Other names for such systems include throttle-less hydraulics, valve-less hydraulics and direct-drive hydraulics. Here, however, we stick to the term pump-controlled actuators or pump-controlled systems. Note that Electro-hydraulic Actuators (EHAs) is another common term for such systems. However, EHAs are a subclass within pump-controlled actuators, where the aim is to provide compact products including the drive unit, the actuator itself and auxiliary systems. This means that all components will be positioned adjacent to the load. That might not be ideal in load handling applications. However, the technical solutions for EHAs are also applicable for systems with spread-out architecture.

Nevertheless, electrification makes energy efficiency more important since the energy storage accounts for a large part of the total system cost. This makes it more relevant to invest in individual drive units for high-power consumers. It is also more interesting to regenerate energy from aiding loads. Pump-controlled systems can be either displacement-controlled, speed-controlled or both. Electrification moves the focus towards speed control, but from a high-level system perspective the control choice is not crucial.

5.1 System Architectures

In this section, basic pump-controlled architectures will be presented. Note that additional valves will be required for proper load-holding functionality. Nevertheless, controlling a cylinder with a pump might seem like a fairly straightforward task. However, there are two things that make this problematic:

1. **Asymmetric cylinders**

Most mobile applications use asymmetric cylinders due to size considerations. This means that the flow that enters the cylinder is different from the flow that leaves it, and the difference must be considered.

2. **Regeneration**

It is desirable to run the hydraulic machine as a motor for aiding loads, and the mode switch between pumping and motoring or vice versa can cause problems.

Due to these complications, there is a wide variety of pump-controlled architectures. It is reasonable to divide the architectures into closed and open circuits, which is done here. In closed circuits, both sides of the pump are allowed to act as the high pressure side and the fluid is recirculated to the greatest possible extent. However, it will soon become clear that this division is not optimal since there are architectures that contain both open- and closed-circuit pumps.

5.1.1 Open-circuit Architectures

Different open-circuit architectures are shown in Fig. 5.1. Fig. 5.1a shows the simplest possible pump-controlled architecture. This is a single acting system, which makes its usefulness limited. Fig. 5.1b shows how the single acting system can be modified to become double acting. A slightly modified version of this system is investigated in Paper [IV].

Fig. 5.1c shows a refined version of Fig. 5.1b. This architecture was presented and demonstrated by Heybroek [80]. The circuit offers plenty of control opportunities such as floating and differential modes.

Fig. 5.1d is principally different from those presented previously. This circuit has two hydraulic machines and the displacement ratio corresponds to the area ratio of the cylinder. However, flow matching is still a problem with this circuit architecture. Flow matching is problematic partly because it can be difficult to find the correct pump sizes and partly because of pressure-dependent leakage. Pressure relief valves and anti-cavitation valves are therefore considered to be required [81]. However, one alternative is to solve this problem using two separate drive units.

Figure 5.1 Pump-controlled open circuit architectures.

5.1.2 Closed-circuit Architectures

Different closed-circuit architectures are shown in Fig. 5.2. Fig. 5.2a represents a class of circuit design that makes use of one closed-circuit machine together with valves that take care of the differential flow. The figure shows a circuit with actively controlled valves, but most circuits of this type use pilot-operated check valves or inverse shuttle valves instead. However, these systems suffer from mode switch oscillations [82], but active valves can be used to enhance mode switch stability [83]. This class of circuit can be regarded as the simplest closed-circuit alternative, since it only requires a single conventional hydraulic machine.

Fig. 5.2b shows how a three-port machine can be used to compensate for

the differential flow. This is considered to be a compact solution. The ports of the machine are designed to match the area ratio of the cylinder. This solution was patented in 1990 [84], and has been investigated in e.g. [85], [86]. It has been proven to work, but the valve plate design is challenging.

An illustration of a circuit using a more conventional transformer, compared to the three-port machine, is shown in Fig. 5.2c. Note that the transformer can be placed on either side of the cylinder.

In Fig. 5.2d a closed circuit pump is used together with an open circuit pump that is dimensioned to match the differential flow. This type of circuit suffers from the same flow-matching problems as the circuit in Fig. 5.1d. However, the solution to use two separate drive units can also be applied here.

5.1.3 Reservoir Pressure Aspects

It might be tempting to choose an open-circuit solution with one pump because it has a similar architecture to a conventional system, and therefore to assume that a vented tank can be used. However, a vented tank is not recommended in most systems. In fact, it is preferable to have a pressurised reservoir in all presented architectures, but primarily in those where reservoir pressure defines the actuator's low pressure level. The argument is that the stiffness of the actuator will be increased with the pressure. Another argument for having high reservoir pressure is that the risk of cavitation is reduced. This is especially important in architectures where fluid is sucked through valves, but it is also beneficial for the pump performance. Furthermore, in Paper [IV] it was concluded that a high reservoir pressure can be used to reduce problems with mode switch oscillations since the increased pressure increases the controllable transition region, as illustrated in Fig. 9.

5.1.4 Comparison Between Open and Closed Circuits

In Paper [IV], the characteristics of variants of the circuit in Fig. 5.1b and Fig. 5.2a are compared. It was concluded that they suffer from similar problems, but that the open-circuit solution has one additional issue. That is that the sign of the rotational speed must change with the load. The advantage of the open circuit is that the pump can be designed for open-circuit use, which amongst other means better suction performance. However, with pressurised reservoirs, the suction performance aspect might not be of the highest importance.

(a) Single closed-circuit machine.

(b) Three-port machine control.

(c) Closed-circuit machine with transformer.

(d) Closed circuit with open circuit machine.

Figure 5.2 Pump-controlled closed circuit architectures.

Figure 5.3 *Controllable switching region.*

5.2 Commercial Products

Applications for pump-controlled systems for symmetrical cylinders include flight control in aircraft [87], and can be considered to be established in the market for high-end products. These are basically built as rotational hydraulic transmissions, which have been around for more than 100 years. However, some suppliers now also offer solutions for asymmetric cylinders [88]–[91]. These are generally compact EHA solutions.

6

The Digital Pump

There is no established definition of a digital pump, but the subject has been debated [92]. However, a digital pump is defined here as a pump with discretely variable displacement. Several types of pumps meet this definition, for example the dual-displacement pumps and the DDPs described in section 3.4. However, several pump element could also be connected in parallel, where each pump element can be bypassed. This is the principle that has been further investigated in Paper [II] and will be presented here.

6.1 The Wobble Plate Pump

The concept that will be presented is based on a WPP. The working principle of a WPP is shown in Fig. 6.1. The WPP is an inline piston pump with a rotating swashplate. This means that the pistons are translating. Therefore, check valves can be used for connecting the cylinders to either the outlet or inlet line. This means that the flow pulsations due to compression are very small and basically independent on operating conditions because the check valves will always open at the right pressure level. However, the check valves must be designed with care to avoid instability. This is investigated in Paper [I] and it concludes that there are many parameters that affects the stability performance.

Note that the WPP with check valves cannot operate as a motor and that the flow direction is not dependent on the rotational speed direction. In other words, it can only work in one quadrant.

Figure 6.1 Working principle of a two-piston WPP.

6.2 The Digital Pump Concept

In the presented digital pump concept, which is further described and analysed in Paper [II], the pistons within the WPP are divided into groups and each group can be activated or deactivated individually. A patent from Sauer-Danfoss (now Danfoss Power Solutions) for this idea was granted in 2013 [93]. By having groups of different sizes, few groups are needed for a comparatively large number of displacement settings. The following analysis is based on so-called binary scaling, which means that the displacement ratio between two adjacent groups is two, which is illustrated in Fig. 6.2. This gives an equally spread displacement distribution, with the highest number of displacement settings to number of groups ratio possible.

Figure 6.2 Control of a binary scaled digital pump with three pump elements.

Figure 6.3 Optimal grouping of different configurations. Each group is colour coded. The number in the cylinders is the total number of cylinders in the group it belongs to.

The number of available displacement settings, n_d , as a function of the number of groups, n_g , is described by:

$$n_d = 2^{n_g}. \quad (6.1)$$

Assuming that all cylinders are placed at the same radius and have the same diameter, the number of required cylinders is:

$$n_{cyl} = n_{g,min} (2^{n_g} - 1), \quad (6.2)$$

where $n_{g,min}$ is the number of cylinders in the smallest displacement group.

6.2.1 Valve Configuration

In Paper [II], two principally different valve configurations with directly actuated valves were analysed. These configurations are shown in Fig. 3.

In both configurations, the flow only has to pass through one component. Therefore, their pressure drop performance can be considered as equal. However, it is only the three-way valve that allows for motor operation. When

(a) Digital pump with three-way bypass valves.

(b) Digital pump with two-way bypass valves.

Figure 6.4 Two possible valve configurations for a digital pump with two pump elements.

using a WPP, this is not relevant since the WPP cannot run as a motor, but if other kinds of pumps are used it may be of interest. The two-way configuration requires two components (the two-way valve and a check valve) per pump element, whilst the three-way valve can manage the task by itself. The two-way valve has a switching behaviour that can be interpreted as unpredictable, if it is not compensated for in the valve timing, which the three-way valve does not suffer from. However, when switching with the three-way valve, there is a risk of either noticeable backflow or high pressure build-up if the valve is not fast enough. Therefore, the valve design is more crucial. These aspects are summarised in Table 6.1. There are also circuits that use a three-way valve in combination with an additional check valve. However, that configuration is considered to be uninteresting since it will basically give the same performance as the two-way configuration, but with additional pressure drops.

Table 6.1 Properties of different valve configurations.

	Two-way valve	Three-way valve
Pressure drop	+	+
Motor operation	-	+
Number of components	-	+
Switching timing	-	+
Valve design sensitivity	+	-

6.3 Noise

In chapter 3, it was stated that the noise generated from a pump depends on the flow pulsations and the forces and moments within the pump. It is obvious that some of the configurations presented in Fig. 2b will give odd flow ripple due to the distribution of the pistons. However, a quantification of this oddity is shown in Fig. 5, where the amplitudes of the flow ripple for different configurations of displacement settings are shown.

As expected, the amplitudes are low when the number of pistons in the smallest group is odd and larger than one (i.e. configuration 3+6 and 3+6+12). However, the difference between configuration 3+6+12 and 2+4+8 is not that

(a) Normalised flow pulsation amplitudes for various cylinder configurations and different displacement settings. The shown amplitudes are for three-way by-pass simulations at 1000 rpm and 35 MPa.

(b) Normalised kinematic flow pulsation amplitudes for various cylinder configurations and different displacement settings.

Figure 6.5 Simulated flow pulsations versus kinematic flow pulsations. The amplitudes are normalised to the ideal flow at full displacement. The observed configurations are the ones within the green dotted line in Fig. 2.

Figure 6.6 Forces and levers from the pistons within the pump.

big when the pre-compression is considered. This means that the 2+4+8 configuration could be an interesting alternative since it has fewer pistons, which is preferable from a manufacturing point of view.

Nevertheless, regarding the forces, they are also expected to vary more than in conventional pumps. An example of the forces and lever arms for the 2+4+8 configuration is shown in Fig. 6.6. It can be seen how the effective force moves along a wide area at small displacement settings, but that the forces are also small at low displacement settings. However, the amplitude in force variations is still relatively large.

6.4 Comparison With Digital Displacement

In the DDP, the inlet check valves are replaced with electronically controlled on/off valves. This control acts closer to the source, and the losses can be smaller since the pressure drop over the outlet check valve and the bypass valve is eliminated. Furthermore, transients can be minimised since each piston can change mode exactly at the time when it does not deliver any flow. This allows for smooth operation. Since each piston is controlled individually, the resolution of the discrete control is high. However, a large number of fast but small valves are required as well as sophisticated closed-loop control. The presented digital pump concept strives for the opposite, i.e. few and simple valves with simple and robust control. They should therefore not be considered as competitors, even though it is easy to mix them up by their names.

Nevertheless, Danfoss' version of the DDP has three separated outlets. These outlets can be used in a similar manner as the pump elements in a digital pump. However, it does not really make sense to bypass them.

7

Digital Pumps in Speed-controlled Systems

This chapter focuses on the opportunities that come from using discrete displacement in variable speed drives. However, first it is reasonable to clarify why variable displacement should be considered in combination with variable-speed drives. There are three main reasons. These are:

- **Increased energy efficiency**

The additional degree of freedom makes it possible to optimise control for energy efficiency. This means that the operating conditions for electric and hydraulic machines can be optimised based on their efficiency properties. An example of such control is described in [94].

- **Downsizing possibilities**

By reducing the displacement at high pressure levels, the torque requirements for the electric motor are reduced, which allows for downsizing if maximum flow is not required at maximum pressure.

- **Improved dynamic performance**

Improved dynamic performance means faster response. A faster response can be achieved since the contribution from the electric and hydraulic machine are combined. This has been investigated in [95].

The first two arguments will be analysed further in the next section. The third argument has not been analysed since it is not considered as an opportunity when digital pumps are used.

7.1 Case Study: The Loader Crane Application

In this section, the results from a simulation study of the possibilities for a specific loader crane application are presented. The general idea was to quantify the potential gains with a digital pump. The system that was considered is shown in Fig. 7.1. The analysis is based on a given drive cycle (e.i. pressure and flow) and the required input energy to the inverter is then calculated for different system configurations. More details can be found in Paper [III].

Figure 7.1 System boundary for the analysis.

7.1.1 The Drive Cycle

Knowledge about the drive cycle is of immense importance when evaluating the possible gains with a digital pump. This cycle is developed from long-term measurements of a crane in the field. The development process is described in [96]. The drive cycle is represented in Fig. 7.2. It can be seen that the crane mainly works with low flow rates and medium pressure levels, which means generally low power.

7.1.2 Power Losses

Fig. 7.3 shows how the power losses are distributed between the different components for the analysed drive cycle. These results can also be found in Paper [III]. However, results from simulations of a variable pump are also added here as a reference.

The simulations are based on efficiency maps, from which losses have been calculated and scaled to represent the simulated machine sizes. The efficiency maps for the pumps and motors are provided by the manufacturers. Note that controller losses were included in the measurements for the variable pump. Nevertheless, it can be seen that the potential energy gains with using digital or variable pumps are small. It can also be seen that the inverter losses

Figure 7.2 Example of a drive cycle in a pressure-flow diagram. The difference between the maximum cycle power and the corner power corresponds to the potential torque downsizing.

Figure 7.3 Power loss distribution for a fixed pump, digital pumps with 2 and 3 pump elements, and a continuously variable pump. The electric motor and inverter were dimensioned for the corner power in all configurations.

are surprisingly substantial. One explanation to that is that it is last in the power chain, which means that it must have a higher efficiency than the other components if the losses should be the same. However, the inverter losses are based on data from [78] and it is uncertain if this data is representative for the commercial inverters that are available for mobile machines today.

7.1.3 Downsizing Possibilities

Whether or not downsizing is possible depends on the machine's drive cycle. If the machine must be able to work at maximum pressure levels with maximum flow rates (i.e. at corner power), downsizing is not possible. However, for a drive cycle similar to the one presented in Fig. 7.2, the required torque rating for the electric motor can be substantially reduced. The downsizing possibilities for the current drive cycle are shown in Fig. 11.

Figure 7.4 Drivable fraction of the drive cycle as a function of motor torque for different number of binary scaled pump elements. 'Inf' corresponds to a continuously variable pump. The vertical lines are placed at the torque level where 100% of the cycle can be performed.

7.2 Generalisation

The analysis above was carried out for a specific drive cycle. However, by comparing the energy consumptions for a fixed pump and a digital pump in the entire operational region, a visual representation of when digital pumps are of interest can be provided. This is what Fig. 7.5 shows. It can be seen that the highest energy gains are in the region when high pressure but low flow rates are demanded.

Figure 7.5 Potential power gains when using a digital pump with two pump elements compared to using a fixed pump.

8

Discussion

Most linear hydraulic actuators will most likely not be replaced by electric in the foreseeable future. Therefore, the development of hydraulic machines and systems is, and will continue to be highly relevant. However, the digital pump will not revolutionise the fluid power industry. It should be regarded as a simple and compact alternative to fixed or variable pumps. The presented WPP-based digital pump can, however, only be relevant in systems where electric regeneration is not of interest, and in the long run, electric regeneration is likely to be desirable in many applications. Nevertheless, the general bypass principle can be applied to other machines as well, such as machines with cylinders placed at different radii [97] or simply conventional stacked machines. With a bypass configuration that allows for motor operation, the digital pump design is more future proof. However, it will not be as quiet as the WPP-based design.

The check valves in the WPP make it quiet, and the check valves mean that there are no compression losses due to the dead volumes in the cylinder. This does not apply to conventional valve plate pumps, and especially not at low displacement settings when the dead volume accounts for a considerable proportion. To minimise compression losses, conventional piston pumps are generally designed with dead volumes that are as small as possible. However, small dead volumes mean fast pressure build-up in the cylinders, which leads to a great deal of structure-borne noise [59]. WPPs can have a large dead volume, without having to compromise on total efficiency. Therefore, they can be even quieter. However, bear in mind that a large dead volume will still result in a low volumetric efficiency according to today's standards.

The DDP also has the advantages that come with the WPP-based digital pump, but it is far more complex. However, the commercial version of the DDP cannot currently operate as a motor since it uses check valves for the outlet in the same way as the WPP does. Extensive research has been carried out on replacing these valves with active valves to allow for motor operation, which would make this version more suitable for future use.

When it comes to pumps and electrification in general, pumps must be better adapted for variable speed drive, which primarily means that they must perform better at low speeds. Furthermore, they should be designed for bidirectional operation and have regeneration abilities.

Regarding variable displacement in combination with variable-speed drives, some might consider this to be unnecessary. However, in chapter 4 it was stated that variable displacement in combination with variable speed can make sense in three respects: response, efficiency and downsizing. However, displacement controllers have downsides. To adapt to the aims of energy efficiency, the controller losses must be reduced, and should be considered within the efficiency measurements for pumps. Furthermore, the fact that conventional controllers require a minimum pressure to be able to work is probably undesirable in future system designs. Digital displacement technology as well as the digital pump proposed here, is therefore of interest since it does not require any control pressure. Regarding the response, however, a low number of discrete displacement settings would be inappropriate since the response gain cannot be utilised for fine control. The DDP or a continuously variable pump is therefore advantageous from this point of view. Regarding efficiency, it has been shown that the potential energy gains are small if the cycle is not working much at high pressure and low flow rates, but that digital pumps still can be of interest due to the downsizing possibilities. However, downsizing is not necessarily relevant either. For example, consider a loader crane propelled by a large electric truck. In that case, the electric motor is dimensioned for the propulsion of the truck and not the crane application, which is generally less power-demanding.

Lastly, it should be mentioned that a shiftable gearbox will have the same influence as a digital pump in many ways. The gearbox can be designed so that the operational region becomes identical to the region for the digital pump. However, the difference is the speed of the hydraulic machine. A digital pump will run at high speed and low displacement setting, while the pump in a geared solution will run at low speeds. Generally, hydraulic pumps have poor performance at low speeds. Therefore, digital pumps are considered to be preferable. However, solutions with shiftable gears in combination with electric motors are now available for heavier electric vehicles [98] and passenger cars [99], [100].

9

Conclusions

- **RQ1: How can a wobble plate pump be transformed into a digital pump?**

A digital pump is defined here as a pump with discrete displacement. A simple and compact type of digital pump can be created by grouping pistons and integrating bypass valves for each group. Either three-way or two-way valves can be used. Regarding the grouping, binary scaling is considered to be a reasonable grouping method since it minimises the number of valves and offers a uniform displacement distribution. However, the grouping can also be optimised for a specific drive cycle.

- **RQ2: What are the pros and cons of using digital pumps in combination with variable speed drive?**

There are two main reasons for considering a digital pump:

1. The speed of the machines can be kept high, even for low flow rates. This can improve the efficiency, since neither the pump nor the motor are efficient at low speeds.
2. The electric motor can be downsized since the displacement setting can be reduced at high pressure levels.

The downside with the digital pump is the switching behaviour. Switching behaviour depends on the type of bypass valve chosen. However, there are ways to obtain satisfactory behaviour, even though it might affect the control effort or valve complexity in an undesired direction.

- **RQ3: When is the use of digital pumps of interest in electrified mobile systems?** The usefulness is strongly dependent on the drive cycle. Compared to a fixed pump, the energy efficiency can be substantially increased for a drive cycle when the machine is working at high pressures and low flow rates. Otherwise, it is not very reasonable to use a digital pump from an energy perspective. However, if downsizing is

of interest, a digital pump is still relevant as long as the machine is not supposed to work at corner power. Only a few displacement settings are required to be able to benefit from these gains. Nevertheless, if a digital pump is to be relevant, it must be less costly and/or more efficient than a conventional variable pump, which it has the potential to be.

- **RQ4: To what extent will multi-quadrant operation of hydraulic machines be desirable when moving towards electric drives?**

In pump-controlled systems, closed circuits with a single hydraulic machine have a major advantage over equivalent open circuits, in that the sign of the rotational speed is not dependent on the pressure difference. It is therefore relevant to focus on hydraulic machines that can run in four quadrants. With electrification in mind, these four quadrants will likely be defined by the signs of the pressure and speed rather than the pressure and displacement setting, which has previously been common.

10

Outlook

The presented analysis of digital pumps focuses on their potential usefulness. However, work has to be done to achieve smooth switching performance. Otherwise it will not be useful in load handling applications, where precision is critical. A demonstrator is currently under development. This prototype will not be able to run as a motor, but designs that allow motor operation are also relevant for further investigation.

11

Review of Papers

Paper I

Simulation and Validation of a Wobble Plate Pump With a Focus on Check Valve Dynamics

In this paper, the performance of a wobble plate pump is analysed. The focus is on check valves. It describes a non-linear simulation model of the pump that has been developed. The model is then compared with measurements performed with a high-dynamic pressure transducer placed in a cylinder within the pump. The results show that the model captures the general behaviour, but also that the measurements contain some frequency content that the model did not capture. However, the model could predict performance parameters such as pressure peaks and cylinder pressure rates well.

The paper also contains a stability analysis based on a linearised model. Using this model, a straightforward stability criterion for check valve design could be formulated, and this stability criterion correspond well with the non-linear simulation model. However, according to the models, the check valves of the analysed pumps should be unstable at low rotational speeds, which was not seen in the measurements. Possible reasons for the deviation include the neglected friction or inertia of the fluid.

Paper II

As Simple as Imaginable - An Analysis of Novel Digital Pump Concepts

Here, concepts on how to transform a wobble plate pump into a digital pump are presented and analysed by computer simulations based on the model developed in Paper I. As the title implies, the concepts are very simple in principle, and

rely on the grouping of pistons within one pump. The analysis revolves around flow pulsations, and different piston configurations are investigated together with two principally different solutions for bypassing the groups individually. The results show that having more pistons is better, but the conclusion is that a configuration of either 3+9 or 2+4+8 is mostly relevant since a high number of pistons implies a high production cost.

Paper III

Digital Pumps in Speed-controlled Systems – An Energy Study for a Loader Crane Application

This paper focuses on the potential benefits that come from a discrete number of displacement settings compared to a fixed displacement. In the analysis, a drive cycle for a loader crane is used. The results show that the potential energy savings are relatively small, but the downsizing possibilities for the electric motor are substantial. Furthermore, the energy savings will be much higher for a drive cycle that operates extensively at low flow rates and high pressures. A further conclusion is that a small number of discrete displacement settings (e.g. two or three) can be enough to achieve significant improvements regarding energy savings and downsizing possibilities.

Paper IV

Why Not Open-circuit? An Analysis of a Regenerative Speed-Controlled Hydraulic Actuator Concept

Using an open-circuit pump together with a directional valve is one of the most intuitive solutions for controlling a cylinder. Still, there are very few publications on this type of system that also can work in regenerative mode. This paper intends to find out why. The focus is on the problems and design considerations for this open-circuit architecture. It turns out that many problems are similar to the ones the closed-circuit systems suffer from. However, there is an additional problem in that the sign of the directional speed is dependent on the external force. This is considered to be the main disadvantage of the open-circuit system. Another conclusion of the paper is that it is advantageous to have a high reservoir pressure and this pressure can be used to enhance mode switch stability, which also applies to closed circuits.

Bibliography

- [1] S. Skinner, *Hydraulic fluid power - a historical timeline*. lulu.com, 2014, ISBN: 9781291676891.
- [2] J. Bramah, "Method of organizing and constructing water mains and other pipes," GB Patent: GB-181,203,611, Oct. 1812.
- [3] J. Ivantysyn and M. Ivantysynova, *Hydrostatic pumps and motors*, 1. English ed. New Delhi: Tech Books International, 2003, ISBN: 81-88305-08-1.
- [4] *Ax series - piston pumps and motors*, version 200-FL-991636-EN-00, Bucher Hydraulics, 2019.
- [5] L. J. Love, "Estimating the impact (energy, emissions and economics) of the us fluid power industry," Dec. 2012. DOI: 10.2172/1061537.
- [6] "ISO 8643: Earth-moving machinery — Hydraulic excavator and backhoe loader lowering control device — Requirements and tests," International Organization for Standardization, Geneva, CH, Standard, Feb. 2017.
- [7] "ISO 15442: Cranes — Safety requirements for loader cranes," International Organization for Standardization, Geneva, CH, Standard, Nov. 2012.
- [8] M. Djurovic, "Energiesparende antriebssysteme für die arbeitshydraulik mobiler arbeitsmaschinen elektrohydraulisches flow matching," Ph.D. dissertation, 2007, ISBN: 978-3-8322-6361-4.
- [9] M. Scherer, "Beitrag zur effizienzsteigerung mobiler arbeitsmaschinen: Entwicklung einer elektrohydraulischen bedarfsstromsteuerung mit aufgeprägtem volumenstrom," Ph.D. dissertation, 2015, 168 pp., ISBN: 978-3-7315-0339-2. DOI: 10.5445/KSP/1000045508.
- [10] M. Axin, "Mobile working hydraulic system dynamics," Ph.D. dissertation, Linköping University, Fluid and Mechatronic Systems, 2015, ISBN: 978-91-7685-971-1. DOI: 10.3384/diss.diva-121070.
- [11] B. A. Stewart and T. Howell, *Encyclopedia of water science*. CRC press, 2003.

- [12] J. P. Oleson, *The Oxford handbook of engineering and technology in the classical world*. Oxford University Press, 2008.
- [13] A. Ramelli, *Le diverse et artificiose machine del capitano Agostino Ramelli*. Paris, France, 1588.
- [14] N. Manring, *Fluid power pumps and motors: analysis, design and control*. McGraw Hill Professional, 2013, ISBN: 9780071812214.
- [15] P. A. J. Achten, T. L. van den Brink, T. Paardenkooper, *et al.*, “Design and testing of an axial piston pump based on the floating cup principle,” in *Proceedings of the 8th Scandinavian International Conference on Fluid Power (SICFP03), Finland, Tampere*, 2003, p. 805.
- [16] L. Ericson and J. Forssell, “A novel axial piston pump/motor principle with floating pistons: Design and testing,” in *Proceedings of the ASME/BATH 2018 Symposium on Fluid Power and Motion Control, FPMC2018*, American Society of Mechanical Engineers Digital Collection, 2018. DOI: 10.1115/fpmc2018-8937.
- [17] T. Grahl, “Untersuchung des geräuschverhaltens von axialkolbenmaschinen,” 1987.
- [18] S. R. Wilhelm and J. D. Van de Ven, “Design and testing of an adjustable linkage for a variable displacement pump,” *Journal of Mechanisms and Robotics*, vol. 5, no. 4, Sep. 2013, ISSN: 1942-4302. DOI: 10.1115/1.4025122.
- [19] *Checkball piston pumps*, version PES.CB-0310, Dynex Hydraulics, 2011.
- [20] R. J. Foss, M. Li, E. J. Barth, *et al.*, “Experimental Studies of a Novel Alternating Flow (AF) Hydraulic Pump,” ser. Fluid Power Systems Technology, vol. ASME/BATH 2017 Symposium on Fluid Power and Motion Control, Oct. 2017. DOI: 10.1115/FPMC2017-4315.
- [21] P. Y. Li, C. Y. Li, and T. R. Chase, “Software enabled variable displacement pumps,” ser. ASME International Mechanical Engineering Congress and Exposition, vol. Fluid Power Systems and Technology, Nov. 2005, pp. 63–72. DOI: 10.1115/IMECE2005-81376.
- [22] J. Lumkes Jr., M. A. Batdorff, and J. R. Mahrenholz, “Model development and experimental analysis of a virtually variable displacement pump system,” *International Journal of Fluid Power*, vol. 10, no. 3, pp. 17–27, 2009. DOI: 10.1080/14399776.2009.10780985.
- [23] H. Tu, “High speed rotary pwm on/off valves for digital control of hydraulic pumps and motors,” Ph.D. dissertation, University of Minnesota, 2014.
- [24] N. D. Manring and V. S. Mehta, “Physical limitations for the bandwidth frequency of a pressure controlled, axial-piston pump,” *Journal of Dynamic Systems, Measurement, and Control*, vol. 133, no. 6, Sep. 2011, ISSN: 0022-0434. DOI: 10.1115/1.4004056.

- [25] W. Backe and H. J. Feigel, "Neue moeglichkeiten beim elektrohydraulischen load-sensing," *Ö+P; Ölhydraulik und Pneumatik*, vol. 34, no. 2, pp. 106–114, 1990, ISSN: 0029-8697.
- [26] C. Latour. (2007). "Electrohydraulic flow matching: The next generation of load-sensing controls," [Online]. Available: <https://www.machinedesign.com/archive/electrohydraulic-flow-matching-next-generation-load-sensing-controls> (visited on 05/10/2020).
- [27] *Control devices ma and ema*, version RE 92072/02.2019, Bosch Rexroth AG, 2019.
- [28] S. H. Salter and W. H. S. Rampen, "Pump control method and poppet valve therefor," US Patent 5,190,446, Mar. 1993.
- [29] —, "Improved fluid-working machine," WO Patent App. PC-T/GB1990/001,478, Apr. 1991.
- [30] W. H. S. Rampen, "The digital displacement hydraulic piston pump," Ph.D. dissertation, University of Edinburgh, 1992.
- [31] Artemis Intelligent Power. (2020). "History," [Online]. Available: <http://www.artemisip.com/company/history> (visited on 05/10/2020).
- [32] U. B. P. Stein, N. J. Caldwell, W. H. S. Rampen, and J. Almond, "Fluid-working machine and operating method," US Patent App. 10/526,444, Feb. 2006.
- [33] W. H. S. Rampen, N. J. Caldwell, U. B. P. Stein, *et al.*, "Fluid power distribution and control system," US Patent App. 12/374,454, Feb. 2010.
- [34] M. A. Holland, "Design of digital pump/motors and experimental validation of operating strategies," Ph.D. dissertation, Purdue University, 2012.
- [35] K. J. Merrill, "Modeling and analysis of active valve control of a digital pump-motor," Ph.D. dissertation, Purdue University, 2012.
- [36] F. E. Breidi, "Investigation of digital pump/motor control strategies," Ph.D. dissertation, Purdue University, 2016.
- [37] D. B. Rømer, "Design and optimization of fast switching valves for large scale digital hydraulic motors," Ph.D. dissertation, Aalborg University, Nov. 2014, ISBN: 978-87-92846-58-7.
- [38] N. H. Pedersen, P. Johansen, and T. O. Andersen, "Event-driven control of a speed varying digital displacement machine," in *Proceedings of the ASME/BATH 2017 Symposium on Fluid Power and Motion Control, FPMC2017*, American Society of Mechanical Engineers, 2017, V001T01A029–V001T01A029. DOI: 10.1115/fpmc2017-4260.
- [39] H. Tian, P. Y. Li, and J. D. Van de Ven, "Using an angle domain repetitive control to achieve variable valve timing for a digital displacement hydraulic motor," in *Proc. of The Ninth workshop on digital fluid power, Aalborg, Denmark*, 2017.

- [40] J. Tammisto, M. Huova, M. Heikkilä, *et al.*, “Measured characteristics of an in-line pump with independently controlled pistons,” in *Proceedings of the 7th International Fluid Power Conference (7th IFK)*, 22-24 March, Aachen, Germany, 2010.
- [41] M. Heikkilä, J. Tammisto, M. Huova, *et al.*, “Experimental evaluation of a digital hydraulic power management system,” in *The Third Workshop on Digital Fluid Power*, Tampere, Finland, 2010, pp. 129–142.
- [42] M. Green, J. Macpherson, N. Caldwell, and W. H. S. Rampen, “Dexter - the application of a digital displacement pump to a 16 tonne excavator,” 2018. DOI: 10.1115/fpmc2018-8894.
- [43] *Data sheet - digital displacement pump*, version AI332064420016en-000102, Danfoss, 2020.
- [44] K. Huhtala and M. Vilenius, “Comparison of steady state models of hydraulic pump,” in *Proceedings of the 5th Scandinavian International Conference on Fluid Power (SICFP97)*, Linköping, Sweden, 1997.
- [45] P. A. J. Achten, R. Mommers, T. Nishiumi, *et al.*, “Measuring the losses of hydrostatic pumps and motors: A critical review of iso4409: 2007,” in *Proceedings of the ASME/BATH 2019 Symposium on Fluid Power and Motion Control, FPMC2019*, American Society of Mechanical Engineers Digital Collection, 2019. DOI: 10.1115/fpmc2019-1615.
- [46] G. Toet, J. Johnson, J. Montague, *et al.*, “The determination of the theoretical stroke volume of hydrostatic positive displacement pumps and motors from volumetric measurements,” *Energies*, vol. 12, no. 3, p. 415, Jan. 2019, ISSN: 1996-1073. DOI: 10.3390/en12030415.
- [47] N. Manring and C. Williamson, “The theoretical volumetric displacement of a check-valve type, digital displacement pump,” *Journal of Dynamic Systems, Measurement, and Control*, vol. 141, no. 3, Nov. 2018, ISSN: 0022-0434. DOI: 10.1115/1.4041713.
- [48] “ISO 4409: Hydraulic fluid power — Positive-displacement pumps, motors and integral transmissions — Methods of testing and presenting basic steady state performance,” International Organization for Standardization, Geneva, CH, Standard, Jun. 2019.
- [49] “ISO 8426: Hydraulic fluid power — Positive displacement pumps and motors — Determination of derived capacity,” International Organization for Standardization, Geneva, CH, Standard, Feb. 2008.
- [50] N. Manring and C. Williamson, “Calculating the mechanical and volumetric efficiencies for check-valve type, digital displacement pumps,” in *Proceedings of the ASME/BATH 2018 Symposium on Fluid Power and Motion Control, FPMC2018*, American Society of Mechanical Engineers Digital Collection, 2018. DOI: 10.1115/fpmc2018-8834.

- [51] N. J. Caldwell, "Digital displacement hydrostatic transmission systems," Ph.D. dissertation, University of Edinburgh, 2007.
- [52] P. Achten, "Dynamic high-frequency behaviour of the swash plate in a variable displacement axial piston pump," *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering*, vol. 227, no. 6, pp. 529–540, 2013. DOI: 10.1177/0959651813483419.
- [53] J. Lux and H. Murrenhoff, "Experimental loss analysis of displacement controlled pumps," in *Proceedings of the 10th International Fluid Power Conference (10th IFK), Dresden, Germany*, 2016, pp. 8–10.
- [54] M. K. Bahr, J. Svoboda, and R. B. Bhat, "Vibration analysis of constant power regulated swash plate axial piston pumps," *Journal of sound and vibration*, vol. 259, no. 5, pp. 1225–1236, 2003. DOI: 10.1006/jsvi.2002.5231.
- [55] S. Achten P. A. J. and Eggenkamp and H. W. Potma, "Swash plate oscillation in a variable displacement floating cup pump," in *Proceedings of the 13th Scandinavian International Conference on Fluid Power (SICFP13), June 3-5, Linköping, Sweden*, Linköping University Electronic Press, 2013, pp. 163–176. DOI: 10.3384/ecp1392a17.
- [56] World Health Organization and others, *Burden of disease from environmental noise: Quantification of healthy life years lost in Europe*. World Health Organization. Regional Office for Europe, 2011.
- [57] —, "Environmental noise guidelines for the european region," 2018.
- [58] "ISO 226: Acoustics — Normal equal-loudness-level contours," International Organization for Standardization, Geneva, CH, Standard, Aug. 2003.
- [59] A. Johansson, "Design principles for noise reduction in hydraulic piston pumps : Simulation, optimisation and experimental verification," Ph.D. dissertation, Linköping University, Fluid and Mechanical Engineering Systems, 2005, ISBN: 91-85457-15-9.
- [60] J.-O. Palmberg, "Modelling of flow ripple from fluid power piston pumps," in *2nd Bath International Power Workshop, University of Bath, UK*, 1989.
- [61] M. Pettersson, "On fluid power piston pumps: With special reference to flow ripple reduction," Ph.D. dissertation, Linköpings universitet, 1993.
- [62] "Case Construction". (2020). "CASE Unveils "Project Zeus" - the all-new 580 EV - the Industry's First Fully Electric Backhoe Loader," [Online]. Available: <https://www.casece.com/northamerica/en-us/resources/articles/media-case-unveils-project-zeus-580ev-first-fully-electric-backhoe-loader> (visited on 05/10/2020).

- [63] "Volvo Construction Equipment". (2019). "Volvo goes electric on smaller machines," [Online]. Available: <https://www.volvoce.com/global/en/news-and-events/press-releases/2019/volvo-ce-goes-electric-on-smaller-machines> (visited on 05/10/2020).
- [64] "Mecalac". (2020). "Mecalac e12: a 100%-electric excavator for urban building sites," [Online]. Available: <https://www.mecalac.com/en/e12-electric-wheel-excavator.html> (visited on 05/10/2020).
- [65] "Epiroc". (2018). "Epiroc launches new generation battery-driven machines for sustainable mining," [Online]. Available: <https://www.epirocgroup.com/en/media/corporate-press-releases/2018/20181114-epiroc-launches-new-generation-battery-driven-machines-for-sustainable-mining> (visited on 05/10/2020).
- [66] "Liebherr". (2019). "Liebherr Mining Goes Electric," [Online]. Available: <https://www.liebherr.com/en/int/latest-news/news-press-releases/detail/liebherr-mining-goes-electric.html> (visited on 05/10/2020).
- [67] Y. Nishida and T. Nihei, "Introducing the hb335/hb365-1 hybrid hydraulic excavators," *J Komatsu Technical Rep*, vol. 59, pp. 31–37, 2013.
- [68] "Huddig". (2020). "Tigon - Taking charge," [Online]. Available: <https://www.huddig.com/sv/produkter/maskiner/tigon> (visited on 05/10/2020).
- [69] "Volvo Construction Equipment". (2017). "LX01 prototype hybrid wheel loader delivers around 50 % fuel efficiency improvement during customer testing," [Online]. Available: <https://www.volvoce.com/global/en/news-and-events/press-releases/2017/lx01-prototype-hybrid-wheel-loader-delivers-50-percent-fuel-efficiency-improvement> (visited on 05/10/2020).
- [70] "Daimler AG". (2016). "World premiere: the all-electric and networked Mercedes-Benz Urban eTruck," [Online]. Available: <https://media.daimler.com/marsMediaSite/ko/en/13669478> (visited on 05/10/2020).
- [71] —, (2019). "Progress report from Mercedes-Benz Trucks: eActros electric truck successfully tested by customers for over a year," [Online]. Available: <https://media.daimler.com/marsMediaSite/ko/en/44842543> (visited on 05/10/2020).
- [72] "Volvo". (2019). "Volvo Trucks launches sales of electric trucks for urban transport," [Online]. Available: <https://www.volvogroup.com/en-en/news/2019/nov/news-3471774.html> (visited on 05/10/2020).
- [73] *Epto*, version PB-ePTO-EN-EU 160205, Hiab, 2016.
- [74] *Global vehicle motors gvm series*, version 18-06-08 PVD3666 GB GVM, Parker Hannifin Corporation, 2018.
- [75] *Electric drive for off-highway applications*, version 292000P19K-C/CCA-201406-En, Bosch Engineering, 2014.

- [76] S. N. Vukosavic, *Electrical machines*. Springer Science & Business Media, 2012. DOI: 10.1007/978-1-4614-0400-2.
- [77] J. Reinert, A. Brockmeyer, and R. W. De Doncker, “Calculation of losses in ferro-and ferrimagnetic materials based on the modified steinmetz equation,” *IEEE Transactions on Industry applications*, vol. 37, no. 4, pp. 1055–1061, 2001. DOI: 10.1109/28.936396.
- [78] *Adjustable speed drive part-load efficiency*, version DOE/GO-102012-3730, U.S. Department of Energy, 2012.
- [79] K. Heybroek, “On energy efficient mobile hydraulic systems: With focus on linear actuation,” Ph.D. dissertation, Linköping University Electronic Press, 2017.
- [80] —, “Saving energy in construction machinery using displacement control hydraulics: Concept realization and validation,” 2008.
- [81] H. C. Pedersen, L. Schmidt, T. O. Andersen, and M. H. Brask, “Investigation of new servo drive concept utilizing two fixed displacement units,” *JFPS International Journal of Fluid Power System*, vol. 8, no. 1, pp. 1–9, 2014. DOI: 10.5739/jfpsij.8.1.
- [82] C. Williamson and M. Ivantysynova, “Pump mode prediction for four-quadrant velocity control of valueless hydraulic actuators,” in *Proceedings of the JFPS International Symposium on Fluid Power*, The Japan Fluid Power System Society, vol. 2008, 2008, pp. 323–328. DOI: 10.5739/isfp.2008.323.
- [83] P. H. Gøyttil, D. Padovani, and M. R. Hansen, “A novel solution for the elimination of mode switching in pump-controlled single-rod cylinders,” *Actuators*, vol. 9, no. 1, p. 20, Mar. 2020, ISSN: 2076-0825. DOI: 10.3390/act9010020.
- [84] R. L. Kenyon, D. Scanderbeg, M. E. Nolan, and W. D. Wilkerson, “Electro-hydraulic actuator,” European patent: EP0395420A2, Oct. 1990.
- [85] Q. Long, Y. Yang, and H. Xuwei, “Simulation and experimental research on the axial piston pump with series three-windows in valve plate,” in *Proceedings of 2011 International Conference on Fluid Power and Mechatronics*, IEEE, 2011, pp. 71–76. DOI: 10.1109/fpm.2011.6045732.
- [86] J. Huang, H. Zhao, L. Quan, and X. Zhang, “Development of an asymmetric axial piston pump for displacement-controlled system,” *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, vol. 228, no. 8, pp. 1418–1430, 2014.
- [87] D. Van Den Bossche, “More electric control surface actuation; a standard for the next generation of transport aircraft,” in *Proceedings of the 10th European Conference on Power Electronics and Applications*, 2003, pp. 2–4.

- [88] *Smart hydraulic actuators (sha)*, version January 2020, Kyntronics, 2020.
- [89] *Servo drive cldp*, version 8110-12 en VTHL 2012-02, Voith, 2012.
- [90] *Compact cha*, version HY22-3101E 7/13, Parker Hannifin Corporation, 2013.
- [91] *Servo-hydraulic actuator - sha fields of application*, version RE 08137, Edition: 2020-02, Bosch Rexroth AG, 2020.
- [92] P. Achten, M. Linjama, R. Scheidl, and S. Schmidt, "Discussion: Is the future of fluid power digital?" *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering*, vol. 226, no. 6, pp. 724–727, 2012. DOI: 10.1177/0959651812443935.
- [93] M. D. Gandrud, "Check valve pump with electric bypass valve," US Patent 8,469,677, Jun. 2013.
- [94] J. Willkomm, M. Wahler, and J. Weber, "Model predictive control of speed-variable variable-displacement pumps to optimize energy efficiency," in *Proceedings of the 9th International Fluid Power Conference (9th IFK), Aachen, Germany*, vol. 1, 2014, pp. 372–385.
- [95] J. Willkomm, M. Wahler, and J. Weber, "Process-adapted control to maximize dynamics of speed-and displacement-variable pumps," in *Proceedings of the ASME/BATH 2014 Symposium on Fluid Power and Motion Control, FPMC2014*, American Society of Mechanical Engineers Digital Collection, 2014. DOI: 10.1115/fpmc2014-7821.
- [96] A. Rankka, A. Dell'Amico, and P. Krus, "Drive cycle generation for a hydraulic loader crane," in *Proceedings of the 16th Scandinavian International Conference on Fluid Power (SICFP19), 22-24 May, Tampere, Finland*, 2019.
- [97] "Sunfab Hydraulics AB". (2018). "Dual flow pumps," [Online]. Available: <https://www.sunfab.com/se/products/dual-flow-pumps/?lang=eng> (visited on 05/10/2020).
- [98] *Ev transmissions - efficient technology for electric commercial vehicles*, version EMOB0003, Eaton, 2019.
- [99] "ZF Friedrichshafen AG". (2016). "Paradigm Shift for Electromobility: ZF Presents New Electric 2-Speed Drive for Passenger Cars," [Online]. Available: https://press.zf.com/press/en/releases/release__10181.html (visited on 05/14/2020).
- [100] *The new taycan - soul, electrified*. Version WSLT1901000220 EN/WW, Dr. Ing. h.c. F. Porsche AG, 2019.

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Anders Johansson, 2003

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