Mobile Fluid Power Systems
Design

with a Focus on Energy Efficiency
Mobile Fluid Power Systems Design

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Björn Eriksson
To

my family
Abstract

This work deals with innovative energy efficient fluid power systems for mobile applications. The subjects taken up concern to what extent and how energy losses can be reduced in mobile working hydraulics systems. Various measures are available for increasing energy efficiency in these kinds of systems. Examples include:

**Flow controlled systems** The pump controller is switched from a load sensing to a displacement controlled one. The displacement is controlled in an open loop fashion directly from the operator’s demand signals. This reduces energy consumption at the same time as dynamic issues that are attached to LS systems can be avoided.

**Individual metering valve systems** Flexibility is increased by removing the mechanical coupling between the meter-in and meter-out orifices in directional valves. An overview of this kind of system is given in the thesis. A design proposal that has been implemented is also presented. Initial test results are shown. Patents for this particular system have been applied for.

**Displacement control** Metering losses are reduced by removing the directional valves. One pump is used for each load in such systems. This hardware layout involves considerable changes compared to conventional systems. Displacement controlled systems are not studied in this work.

In mobile applications, overall efficiency is often poor and losses are substantial. The measures listed above can help improve this significantly in such applications. A flow dividing system can decrease energy consumption by about 10% and an individual metering system by about 20%. Losses in pump controlled systems are difficult to give a figure for; the losses are rather attached to the pumps and motors and not to the system layout. However, the losses for these systems are presumably even lower than for individual metering systems. The main focus in this work is on individual metering systems but questions about which components and so on are also treated. For example, the Valvistor valve concept has been studied as part of this work.
Acknowledgements

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Linköping in October 2010

Björn Eriksson

\(^1\)The division name before 2010 was Division of Fluid and Mechanical Engineering Systems
The following seven appended papers will be referred to by their Roman numerals. All papers are printed in their originally published state with the exception of minor errata and changes in text and figure layout.

In all papers the first author is the main author, responsible for the work presented, with additional support from other co-writers. Papers [VII] and [X] are exceptions where the two first authors are the main authors, responsible for the work presented, with additional support from the co-writers.


Patents


Papers not included

The following papers are not included in the thesis but constitute an important part of the background.


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Fluid power is the technology of exploiting the properties of fluids to generate, control and transmit power using flow and pressure of fluids. The fluid can be of any kind. If it is a gas the technology is called pneumatics. If it is a liquid the technology is called hydraulics.

In the 17th century Blaise Pascal formulated Pascal’s law:

“Pressure exerted anywhere in a confined incompressible fluid is transmitted equally in all directions throughout the fluid.”

This was the birth of fluid power. Pascal also invented a hydraulic press which used hydraulic pressure to multiply force.

The first modern fluid power system first appeared in 1795 when the British engineer Joseph Bramah was granted a patent for a press invention [1]. Bramah’s press was first used to compress soft bulky materials and later on it was also used to crush seeds to produce vegetable oil.

William George Armstrong, a British industrialist who invented the hydraulic crane as well as the accumulator, and Joseph Bramah are considered to be the fathers of the modern fluid power engineering field.

Burrows gives an overview of the early development of fluid power in [2].

1.1 Background

Fluid power systems are power generating and/or transmitting subsystems. They are used in a wide range of applications, mobile as well as industrial. In mobile machinery, fluid power is used for propulsion, working hydraulics and auxiliary functions. The main reasons for preferring fluid power systems to other technologies are:
**Power density** Fluid power components are superior in compactness to other technologies, compared for instance with electrical components.

**Robustness** Fluid power systems have the ability to handle force impacts better than, to for example, mechanical transmissions, which often contain fragile gears.

**Cost** For high power applications fluid power components are generally available at lower cost compared to other technologies.

Working hydraulics in mobile fluid power systems often contain several different actuators. In most systems the actuators share one single pump. The need for only one system pump makes the fluid power system compact and cost-effective since the pump is a complex and expensive component.

A hydraulic load often consists of two ports, for example motors and cylinders. Such loads have traditionally been controlled by a directional valve that controls the ports by one single control signal; the position of the main spool. With this kind of directional valve, the inlet (meter-in) and outlet (meter-out) orifices are connected mechanically. The mechanical connection makes the system robust and easy to control. But, at the same time, the system lacks flexibility. For example, there are a number of types of losses attached to this configuration. These include losses due to simultaneously operated functions with different pressure demands. There are also unnecessary losses at meter-out orifices since these are dimensioned for an over-running load. This causes needless pressure losses when handling restrictive loads.

To overcome these shortcomings individual metering systems are an alternative that can be applied. Independent metering is an umbrella term for systems where the meter-in and meter-out orifices are independently controlled. This can be realised in different ways, which is a part of the scope of this work. By decoupling the meter-in and meter-out orifices a range of opportunities open up. For example, the system can change flow paths during operation and recuperation and regeneration can be utilised to reduce energy consumption. Recuperation is when oil is pressurised by external loads and then fed back into the system. Regeneration is when both cylinder chambers (or motor ports) are connected to the pump line and the superfluous pressurised oil at the meter-out orifice is fed back to the pump line.

Energy saving aspects are among the main reasons for research on this kind of systems, but there is also an opportunity for dynamic improvements compared to conventional systems.

This thesis discusses the area of mobile working hydraulic systems with special attention to energy efficiency. Different types of systems are studied with the base in load sensing (LS) systems.
1.2 Limitations

This thesis concerns the energy efficiency and dynamic characteristics of mobile fluid power systems only. The work is also limited to valve controlled working fluid power systems only. For instance, propulsion systems are not treated at all. Other system layouts, machine controlled or displacement controlled systems, among others, are not studied. Aside from these system characteristics, other properties such as production considerations regarding manufacturing and marketing are not treated.

1.3 Contributions

A deeper understanding of mobile fluid power systems, how losses can be reduced in them and how this influences the dynamic properties is the most important contribution of this thesis. A novel system layout utilizing a type of bi-directional proportional poppet valve which is based on the Valvistor principle is proposed. A control strategy with restrictive use of sensors for this system is studied and implemented in a demonstrator. A contribution to a new generation of simulation tool based on transmission line modelling (TLM) that can be used for real-time and optimization applications has also been accomplished.
Energy efficiency has been an active research topic in the field of fluid power for a long time.

The development of both hydraulic components and systems is controlled by demands from the market regarding controllability characteristics, system efficiency and flexibility. In certain applications, authority requirements also drive development. One example is the requirements concerning load holding valves in applications where humans are present. A review of the research and technology in the field of mobile working hydraulic systems is given in this chapter.

Mobile working hydraulic applications are often designed in such a way that more than one function is supplied from one single pump. The total installed power at the consumer side, actuators such as cylinders and motors, is generally considerably higher than the installed power at the delivery side, the system pump. This is feasible because the actuators almost never request their maximum power at the same time.

Demands from the market for better control properties, higher energy efficiency and more flexible systems have pushed the development of mobile working hydraulic systems toward load sensing (LS) systems. This is the state-of-the-art today.

LS-systems are in various aspects often considered to have better control properties than for example open-centre systems, which are considered to be a proven, simple and robust system layout, figure 2.1(a). An LS-valve is often
equipped with a pressure compensator which controls the pressure drop over the control orifice at the main spool, figure 2.1(b). Different loads can thereby be operated almost without cross-talking\(^1\).

Variable pumps are suitable in systems with LS-valves; the pump adopts the pressure level from the highest operated load. The pump is pre-set to maintain a certain pressure margin beyond the highest sensed load pressure. This pre-set margin is necessary to guarantee the pressure drop over the main spool controlled by the pressure compensators; otherwise the pressure compensation of the highest load can not be maintained. In many applications, losses in LS-systems are considerably reduced compared to losses in open centre valve systems with fixed pumps where all excess flow is lost to tank.

There are two major reasons why LS-system design has met with success:

**Energy efficiency** Variable pumps considerably reduce metering losses, especially when systems are operated at part loads.

**Handling qualities** The pressure compensators suppress cross-talk between different functions.

The LS-technique is mostly used in applications where handling qualities and/or energy efficiency are important. An example of such applications is forestry machinery.

One weakness of LS-systems is hydraulic damping. Valves contribute to hydraulic damping through pressure-flow characteristics. To obtain damping from a valve the flow has to decrease when the load pressure increases and vice versa. In LS-systems with pressure compensated valves, the primary design endeavours to achieve low influence on the flow from the load pressure. This decreases the damping capability of the valve. The downside of

\(^1\)Changes in one load influence other loads.
this load independence feature is potential system dynamic issues, especially in closed loop control, see paper [IV].

The pump in the system shown in figure 2.1(b) is controlled in a closed loop, where the highest load pressure is the feedback signal. Two principal weaknesses with LS-systems are:

**Oscillations** Because of the pressure compensators, the system can be relatively undamped; low pressure-flow dependency. In specific points of operation, LS-systems can thereby display oscillatory behaviour.

**Pressure margin** One reason to use a variable pump is to suppress flow and pressure losses. Nonetheless, there is a needless pressure loss in LS-systems, viz. the excessive pressure margin set by the pump controller. This margin is necessary to overcome the throttle losses between the pump and the directional valves. These losses are system dependent and change with external and internal conditions such as temperature, oil properties, hose lengths and so on. The pressure margin is often set substantially higher than necessary to ensure that it is high enough at all operational points.

The weaknesses of LS-systems have been the subject of extensive research. For example, Krus in [3] gives an analytical and systematic description of LS-systems and their dynamics, including pump controllers.

Another research area is displacement controlled systems where one pump is dedicated to each function, either as a transmission in a closed circuit [4] or in an open circuit [5–7]. Displacement controlled systems share the weakness of low dampening with LS-systems since there is no pressure-flow dependency due to the lack of metering valves. This can be handled by adding artificial damping with a pressure feedback, in this case to the electrical pump controller. A practical implication of this is that the response demands on the pumps in these systems can be cumbersome.

Accumulator systems are used when energy is stored over time in hydraulic systems. Liang and Virvalo [8] for example have done research on mobile fluid power systems using accumulators. There are also commercial products utilising accumulators, one of which was invented by Bruun [9].

An early broad overview of load sensing systems was published by Andersson [10] in the early ’80s. An overview of energy efficiency fluid power systems in particular is given by Rydberg in [11].

Most commercial mobile fluid power systems are still operated with open centre valves and fixed pumps. The reason is that these systems have lower initial cost as well as being robust. Those properties are appreciated by industry.

A step forward from the described LS-systems is to use another type of pump controller.

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2 Flow should increase when the pressure drop increases through a valve to contribute additional damping to the system.
The pump controller in an LS-system is designed to control the pressure level despite the flow demanded by the operator. Instead, it seems more natural to control the pump flow rather than the pressure level. This can be done by removing the LS feedback hose in figure 2.1(b) and controlling the pump according to the total flow demand, see [12–16] and [IV, XII]. The challenge in such a system is to match the valve signals to the pump displacement signal precisely. This can be realized by changing the design of the compensators and using, for example, post-compensators. These types of compensator have been studied in [12] and [16].

Since there is no feedback signal to the pump in such systems, it is essential to have knowledge about every flow consumer in the system. An example of unknown loads is when bodybuilders attach hydraulic systems to existing ones. See [15] and [13].

The idea of controlling the pump without load feedback can also be utilized in open centre valve systems with variable pumps, see [17] and [18].

The next step would be to decouple the meter-in and meter-out control orifices in the valves. This has been a research topic since the late ’80s or early ’90s. One of the first publications in the area is Jansson and Palmberg [19] where the idea of implementing a controller for four independent poppet valves electrically was proposed.

In the literature, various terms are used for independent metering, among them programmable valves [20], multifunctional valves [21], individual metering valves [22, 23], and separate meter-in and meter-out control [24–26]. Some of them do not necessarily mean individual metering, but they are often used as synonyms.

Individual metering systems can be split up into at least two categories:

Fixed flow paths Systems where the flow paths are similar to conventional valves. In these systems the main benefit of individual metering control is lower metering losses, for instance at the meter-out element.

Variable flow paths Systems where the flow paths change during operation, for example when different modes are utilised such as “normal mode”, “high side regeneration mode”, “low side regeneration mode” and so on.

Already in the ’70s the former Swedish company Monsun-Tison had a system called moniti, see figure 2.2 and [27–31]. This system was a pioneer in the area of mobile hydraulics. The moniti system was designed according to four basic principles [27]:

1. The valve functions control flow rate independently of pressure and load.

2. The valve units controlling the flow and direction are located on the actuators themselves and contain all the necessary supplementary devices, such as pressure relief, anti-cavitation and hose break valves.
2.2(a) MONTI system layout, distributed valves at the cylinders. (Figure from [27].)

2.2(b) Example of the MONTI system with fixed pump configuration, more detailed than figure 2.2(a). (Simplified figure from [28].)

Figure 2.2 – Layout of the MONTI system. This individual metering system was introduced in the mid-70s.

Valve sizes are related to the individual actuators; different sizes can be used in the same system.

3. The valves control the pressure in the system to correspond to the requirements of the heaviest load. For fixed displacement pumps, this is done by means of a special bypass valve; for variable displacement pumps, by regulation of the displacement. The system thus never operates at a higher pressure than is necessary.

4. All these main valves are remotely controlled, either by a hydraulic pilot system or by an electrohydraulic system.

The system was not able to handle mode switches like modern individual metering systems. Among the stated advantages were good manoeuvre properties, a smoother learning process for operators and no need for hose burst valves thanks to the valve distribution [30].

Another independent metering valve was developed in the U.S. in the ’80s, called CMX [32]. It was also an early independent metering system. It has a spool valve at meter-in and poppet valves at meter-out. The valve includes a load-drop check valve, load sensing valve and relief valves. The CMX is of the sandwich type and sections can be stacked into a bank supplying several loads.
Advanced electrical controllers utilising different metering modes first appeared in the late ’90s. One of the first publications on individual metering systems in that sense is Jansson and Palmberg [19]. Ever since then interest in this topic has been growing in academia as well as in industry.

One of the first modern individual metering systems showed up in the late ’90s [33]. It is one of the first individual metering valve systems to utilise advanced electronics. The system is a valve consisting of two independently controlled main spools; one controls the meter-in flow and the other the meter-out flow of a load, e.g. a cylinder.

At the beginning of the 21st century an individual metering systems based on poppet valves appeared [34]. It is a poppet valve based individual metering system.

Research on independent metering systems has been conducted with various approaches. Jansson and Palmberg [19] and Elfving [35] used a physical approach for decoupling the quantities at the different load ports. Eriks-son [36] used an LQ scheme for the same purpose. Mattila and Virvalo [37] used a feedback linearisation scheme for decoupling. Andersen et al. [24] studied two simple control strategies using PI-controllers. Hu et al. [38] show an implementation where the independently controlled valves are open loop controlled and realise different conventional systems, e.g. open centre, closed centre, tandem centre and so forth. Liu and Yao [20] introduce a non-linear adaptive robust controller (ARC) for controlling an individual metering system. Kong et al. [21] use a similar approach to Hu et al. but also with PID-controllers. Nielsen [26] proposed a decoupling strategy based on pressure feedbacks using phase lead filters. Pfaff [39] focuses on different control mode possibilities, for example regenerative, recuperative modes and so on. Shenouda and Book [22] have also studied different control mode possibilities. In further work by Shenouda he has also presented approaches for dynamic mode switching [40]. Tabor [41] used a quasi-static approach for the control design of an independent metering system. Yuan and Lew [42] present a non-linear decoupling controller scheme based on a sliding surface technique. Yao et al. [43] studied an asymmetric mathematical model of an independent valve system controlled by PID-controllers. Linjama et al. [44] have done work on digital hydraulics. This technique can be used in individual metering systems [45]. Eriksson [23] introduced an individual metering system utilising a proportional type of poppet valve without the need for pressure sensors. The valve type uses the Valvistor concept, see [V, VI]. Linjama et al. [46] presented a P-control based system. Analyses of different operation modes were also performed using the P-control based system. Liu et al. [47] studied a two-layer controller layout in an independent metering system where the back pressure and load speed were the controlled signals.
The principal aim of this thesis is to investigate and propose how to improve energy efficiency in mobile hydraulic systems. The anticipated reduction in energy losses is in the order of 10-30%.

Flow controlled and individual metering systems are primarily studied within the scope of this thesis.

To reach this aim, it is necessary to investigate different valve concepts which can be deployed in different system layouts. These are critical components in valve controlled mobile systems. It has to be verified that their required characteristics can be realised for use in the intended systems. Characteristics mean static and dynamic performance. Of these, flow capacity and robustness are critical.

The aims also include to propose and investigate a design for a flexible, robust proportional bi-directional 2/2 valve.

The hypothesis is that there are valve controlled system concepts that can increase the energy efficiency of mobile working hydraulic systems.

The objective is also to validate the concepts experimentally to verify the expected performance.
In this work the hypothetico-deductive method [48] has influenced the methodology of the research. The typical procedure has been to:

1. Set the requirements for a given system.
2. Gather experimental data from the system.
3. Model the system and validate the model using the collected data to confirm that the model mirrors reality.
4. To meet the requirements, re-design the system by using the model in a simulation environment.
5. Re-build the system to resemble the outcome of simulations. Then collect data from experiments with the re-built system.
6. Validate the simulation model so that it mirrors the re-built system.

Here follows an example of how this procedure is typically applied in this thesis work. It begin with something that should be improved in some sense, for example improve energy efficiency of a forwarder mobile working hydraulic system. To be able to improve this system there is a need to understand how the system is influenced by certain parameters and circumstances. Experiments are performed to collect information/data from the system, for example dynamic responses from the different working hydraulic functions\(^1\). A mathematical model is then derived to describe the system; software for dynamic simulations, such as HOPSAN, Matlab/Simulink or AMESim, is used. Now, the same input signals that were used in the experiments are applied

\(^1\)Typical working hydraulic functions in a forwarder are swing, lift, jib, telescope, and grip.
to the simulation model. Unknown parameters are set and uncertain ones are tweaked so that the model’s output agrees with the experiments. Typical examples of such parameters are masses, volumes and bulk modulus. This is an inductive step in the procedure. The next step is where creativity comes into the picture. When the model has been verified to an acceptable degree, the improvement work begins. Here, the model is used as a prototype. Parts of the model may be exchanged and/or existing parameters tuned. For example, valves and control laws could be replaced to minimize losses and thereby increase efficiency. In this step, the model is used to predict how the system will act with the changes applied. This part of the work is then deductive. If the changes to the model introduce unacceptable uncertainties into the model it might be an alternative to go back and revalidate the model or at least parts of it. For example, if a valve is replaced there may be a need to validate this sub-model. Finally, the model hopefully meets the requirements. Then, the system is re-built and a final validation can be performed.

As mentioned above, simulation is an essential ingredient. This work has also included work in the simulation tool area. A simulation tool called HOPSAN NG has been developed and this PhD work has been a part of that. More about this simulation program can be found in chapter 6.
Energy efficiency is an important property of fluid power systems. Dynamic characteristics are another important property. System characteristics, both static and dynamic, often have to fulfil given constraints. For example, a new design often has to yield the same characteristics or better in terms of response and damping.

In some mobile applications the propulsion system is hydraulic. A pump is connected to the combustion engine and a motor at the drive shaft. This kind of system is called hydrostatic transmission. In these applications there are two hydraulic systems present: the propulsion system and the working hydraulics. Then there are opportunities to integrate the systems by means of saving energy. This work is centred on working hydraulics only so this chapter is concentrated on working hydraulic systems. However, most mobile machinery uses torque converters for propulsion.

Displacement controlled systems are not studied in this work. Nonetheless, some remarks regarding their comparison to individual metering systems are important to mention. A displacement controlled system, often referred to as valveless, often needs some valves to meet safety requirements. Since the load functions have their own dedicated pump/motor, each has to be sized for maximum flow. A typical example of a dimensioning motion is lowering a bucket in a wheel loader.

The rest of this chapter gives an overview of studied working hydraulic systems and how their energy consumption can be reduced with maintained or improved performance. First, a study of the potential energy saving possibilities in a wheel loader system is presented. Flow controlled systems and
individual metering systems are then studied more specifically.

## 5.1 Potential Efficiency Improvements – a Wheel Loader Example

In order to give an idea of the possible energy saving in a typical mobile working hydraulic system, an energy study of a wheel loader was performed. The study shows the potential for energy reduction in the working hydraulics (lift, tilt and steering) of a wheel loader by applying different hardware layouts. The calculations are based on real measurements of positions and pressure levels in the different cylinders of the working hydraulics in a wheel loader during a “short loading cycle”, see figure 5.1. The energy consumption and efficiency are determined by the circumstances of the operational conditions of the system. The “short loading cycle” is used in this study because it is a representative working condition for the wheel loader application.

The potential efficiency improvement from changing the system layout can roughly be determined by backward calculation from measured load pressures and piston speeds.

![Figure 5.1](image-url) – The short loading cycle.

The mechanical power that is needed to perform this cycle is the force multiplied by the speed of each load. The force can be estimated approximately by measuring the pressures in the cylinders; then, if the friction is ignored, the force is given by the pressures multiplied by the cylinder areas. The flow is then estimated by measuring the speed of the cylinders, ignoring the leakage. The flow in and out from the cylinder are given by the speed multiplied
by the cylinder areas. The power needed is then approximately the force multiplied by the speed of each cylinder.

In this particular study the working hydraulics are studied. This concerns the boom cylinders, the tilt cylinder and the steering cylinders. The presented figures are thus not valid for a whole vehicle, only the working hydraulics subsystem.

5.1.1 Calculation Cases

This section describes the different cases used when the energy consumption of the short working cycle was calculated. All calculations in this section are only rough estimations, but they give a good overall picture of the difference between the working hydraulics configurations in respect of energy consumption.

Case 1

This case represents the minimum energy consumption that can be achieved. Only pure mechanical energy is considered.

This corresponds to a hydraulic system with an efficiency of 100%. There are no throttling losses in the system. It is possible to store energy over time. This system has to contain some kind of ideal accumulator.

Case 2

Here it is assumed that the system cannot assimilate energy over time. This means, roughly, that the system does not contain any accumulators, but that flow can be transferred between cylinders instantaneously. There are still no throttling losses in the system. Otherwise, conditions are the same as in Case 1.

Case 3

This corresponds to the system of today. Here, the system pressure is assumed to be equal to the LS-pressure, which is the highest sensed load pressure.

Case 4

The last case studied is when independent meter-in and meter-out orifices in the valves are introduced. The LS-pressure is calculated similarly to Case 3. There are two main reasons why this system can save a considerable amount of energy using these kinds of valves. These are the opportunity to use:
**Differential mode** When dealing with small partial loads it is possible to minimize the pressure drop over a cylinder by connecting both cylinder chambers directly to the high pressure line. This is called differential mode.

**Floating mode** When dealing with, for example, lowering loads it is possible to minimize the pressure drop over a cylinder by using the floating mode, which is when the cylinder chambers are directly connected to the tank line.

### 5.1.2 Energy Efficiency Comparison of the Different Cases

Normalized\(^1\) results of the calculations described above are shown in table 5.1 and figure 5.2. In table 5.1 the energy consumed is shown for the four different cases.

<table>
<thead>
<tr>
<th></th>
<th>Case 1</th>
<th>Case 2</th>
<th>Case 3</th>
<th>Case 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lift</td>
<td>0.48</td>
<td>0.70</td>
<td>1.00</td>
<td>0.75</td>
</tr>
<tr>
<td>Tilt</td>
<td>0.15</td>
<td>0.42</td>
<td>1.00</td>
<td>0.68</td>
</tr>
<tr>
<td>Steer</td>
<td>0.40</td>
<td>0.40</td>
<td>1.00</td>
<td>0.96</td>
</tr>
<tr>
<td>Total</td>
<td>0.37</td>
<td>0.60</td>
<td>1.00</td>
<td>0.74</td>
</tr>
</tbody>
</table>

The calculations suggest that there exists good potential to reduce the energy consumption of the working hydraulics in a modern wheel loader. By introducing the split spool concept, Case 4, there is an energy reduction of about 25%.

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\(^1\)The normalization is done because of secrecy reasons from the data provider.
Figure 5.2 – Normalized power consumption during the short loading cycle in the different cases.
5.2 Flow Controlled Systems

In load sensing (LS) systems the actuation of the different loads is carried out by joystick signals. These signals pose either a flow or pressure demand from the operator. LS systems used in mobile applications are commonly equipped with pressure compensators. They maintain a constant pressure drop over the directional valves. This constant pressure drop makes the signals from the operator correspond to flow demands. There are different ways of controlling the pump in this kind of system. The LS system approach shown in figure 5.3(a) uses a pressure controlled pump and flow controlled valves.

Since the operator’s signals correspond to flow demands it seems more natural to control the pump by flow. Traditional pre-compensated valves are not feasible together with this kind of pump control because of their reducing valve characteristics. One flow controlled pump control strategy is to control the pump displacement directly; both the pump and the valves are then flow controlled. The pump flow is set to the sum of the flow demands of the actuators, which are proportional to their velocity demands. However, if they are not perfectly matched problems will arise. Two situations may occur:

1. The pump flow is too high. Both compensator spools will close more and the pump pressure will increase until the system relief valve opens. The throttle losses will be huge and the system will emerge as a constant pressure system.

2. The pump flow is too low. The lightest load will slow down and possibly even stop.

This means that the components must be highly accurate. The reason is that traditional pre-compensators control the absolute pressure drop over the control orifice by reducing the pump pressure relative to the load pressure of its own load. This works fine as long as the pump pressure is actively controlled, with for instance an LS feedback.

There are work-arounds to address this unsatisfactory property. The key is to implicate the highest load pressure into the compensators to avoid this behaviour. This can be done by using compensators placed downstream of the control orifice, so called post-compensators. The compensators then act as relief valves instead of reducing valves and all the valve sections will work against the highest load pressure. The entire pump flow will thus be distributed relative to the individual valve openings. This approach has been studied, for example, in [12] and [16].

It is possible to combine the properties of pre-compensators and post-compensators. One solution is to use the actual pressure difference between the highest load pressure and the pump pressure to replace the fixed spring preload in figure 5.3(a). Such a system can be seen in figure 5.3(b).
Traditional load sensing (LS) system with pressure compensated directional valves.

Flow controlled system with flow-dividing pressure pre-compensators.

Figure 5.3 – Two different approaches to system design where the control valves are pressure compensated, both using pre-compensators.
More on flow controlled systems can be found in paper [IV].

5.2.1 Hardware

The hardware in this kind of system is similar to a traditional LS valve system. To achieve the same system capacity the same magnitude of pump size is used; only the pump controller needs to be different. Instead of actively controlling the pump pressure above a certain pressure margin with a highest load pressure hardware arrangement, the pump displacement is controlled directly from the operator demand signal. This yields a system where the load sensing hose to the pump controller can be removed and replaced with an electrically controlled displacement controller. Both systems can be seen in figure 5.3.

Pre-compensators have the advantages of simpler installation and that most existing systems already use pre-compensators. In some systems, pre-compensators can be more or less replaced with flow sharing pre-compensators without even replacing the valve housing.

The flow paths in this system are equal to the flow paths in a conventional LS system, see figure 5.3. The difference in energy consumption is primarily a consequence of the possibility of reducing the pressure drop over the compensators and the directional valve sections. The compensators can be designed in such a way that the pressure drop over it is minimized. This is described in detail in [IV]. Also, the pressure margin pre-set by the pump controller is eliminated. Instead of the prescribed pressure margin, the pressure drop is given by the resistance in the hoses and the compensator restriction.

There is no need for sensors to achieve the desired functionality with this solution. However, it could be beneficial to use sensors to, for example, determine when end stops are reached. It is then possible to adjust the pump flow and avoid unnecessary energy losses.

5.2.2 A Demonstrator System

To verify the properties of flow controlled systems measurements were performed. The application was a wheel loader with an operational weight of 6900 kg, figure 5.4. The machine was equipped with a pump that could be operated both in pressure and displacement control modes. The used valve was prepared for use with both types of compensator described in section 5.2.1. This simplified the tests since the same pump and directional valve could be used in both test cases. A short working cycle was used for comparison tests between conventional LS operation and flow control operation when the measurements were performed. It is the dynamic properties and the reduction of energy consumption that are of interest to look at.
In figure 5.5(b) a reduction of the pressure drop between the pump outlet and the highest load pressure can be seen. The actuator speed is similar in both test runs, see figure 5.5(a). The energy consumption of the working hydraulics was reduced by about 14% in this measurement. Keep in mind that this is a figure specific to this particular application and operational conditions. However, the result shows that there is a potential for energy savings in this kind of system. These measurements are described in more detail in [XII].

Also, the system with flow sharing pressure compensators and flow controlled pump is less oscillative. The controller in load sensing pumps often reduces the damping in systems. This has been studied in previous research, for example see [3].
5.3 Individual Metering Systems

Individual metering systems are a class of fluid power systems where the meter-in and meter-out control valves are individually controlled. This section gives an overview of how individual systems can be designed and their properties.

There is also an example of a specific system which was designed and tested. This system uses proportional poppet valves, which make use of the Valvistor principle.

5.3.1 Hardware

The hardware in individual metering systems can be designed in various ways. Different kinds of valves and different layouts can be utilised. In this section different hardware layouts, how different connection schemes result in different feasibility for energy recovery and so on are presented.

The main characteristic features of independent metering systems are:

Metering losses reduction When controlling a meter-in orifice in a conventional system the meter-out orifice opening is determined by the spool position. For an individual metering system the meter-out orifice can be independently controlled from the meter-in orifice with the objective of, for example, reducing metering losses.

Dynamic response improvements Individual metering systems have more input signals to the plant system compared to a conventional system, which means that a controller has superior opportunities for sophisticated control.

General hardware Unlike conventional valves where a valve can be specified in an enormous number of different spools or configurations, independent metering valves can be designed more generically. For example, software parameter changes can be made instead of different grindings of spools. Product variety can be decreased by a change-over to independent metering valves. This gives the opportunity for an OEM to choose among different component suppliers due to general hardware with no application-dependent notches and so on.

Productivity improvements With the same system power supply\(^2\), an individual metering system is able to increase the speed of a load under certain conditions. For example, while lifting a light load the cylinder can be regeneratively controlled due to the independently controlled meter-in and meter-out valves. A higher force can also be applied on the load due to lower pressure losses, for instance lower meter-out pressure drops.

\(^2\)In practice this means with the same pump size.
Stability improvements Increased control possibilities open for more efficient damping measures to be implemented. For example, the cylinder chamber pressures can be individually controlled in an individual metering system.

Customisable system characteristics System characteristics such as spool notches no longer have to be established by hardware. In individual metering systems it would even be possible to let the operator change the system characteristics on-line.

Float functionality Desirable functionality like floating operation, for example a non-powered motion of a cylinder. This is a common function for the bucket in wheel loader applications. Floating operation also saves energy compared to forced motion since no pump flow is needed.

Recuperation/regeneration functionality Mobile systems operate in a wide range of operation. There is often also one single pump supplying a number of different functions. These things together often cause mobile systems to generate significant losses. These losses can be efficiently reduced by using the cylinders at part load as discrete transformers. This is done by connecting both cylinder chambers to the pump line to transform the pressure up and transform the flow down. Then the pressure demand at part load endeavours to approach the governing system pressure and the throttling losses can thereby be reduced. Individual metering also enables recuperation of load energy, during lowering for instance, to the pump line and other loads.

Given a specific design pattern there often exists a set of different subcategories. This is also true for individual metering systems. All systems in figure 5.6 are able to meet the requirements for enabling the features listed above. The different layouts all have their pros and cons.

The concepts can also be combinations of each other. One example is a valve system presented by Andersson in [32].

There also exist solutions where additional valves are presented. For example, Book and Goering in [50] propose a valve connection between the pump and tank port. This valve is not necessary to fulfil the flexibility stated above, but there may be other reasons to add a valve like this. One is to substitute basic features such as relief functionality. In [38] Hu and Zhang show different implementations of conventional systems, for example open-center, closed-center, and “regeneration” functions with a five-valve solution. In [20] Liu and Yao present a system also using five metering elements; for example they use the cross-load-port valve to obtain regeneration in situations where the load is used as the power source for the motion. In [21] Kong et al. present an implementation of independent metering valves in a five-valve configuration to achieve matching of asymmetric loads. In [26] Nielsen gives an overview of different valve concepts, some of them using
extra valves. In \[45, 46\] Linjama et al. discuss dynamic benefits of using an extra valve that connects both load ports. Yao works with a poppet valve based individual metering system utilizing five valves \[51\].

More on the configurations can be found in \[I\]. A broad overview of different valve configurations and their characteristics is given by Backé in \[52\].

**Valves**

In a chosen system layout the choice of components, especially valves and their types, are nested and coupled with the chosen system layout itself. The different systems in figure 5.6 are best suited to different types of valves. Generally speaking, all valves can be realized as spool valves. In contrast to, for example, 3/3-valves, 2/2-valves can be realized as poppet valves as well.

Spool valves have a number of advantages. The technique with notches for small valve displacements enables accurate valve opening control. Yuan et al. \[53\] have for example worked with accurate flow control in spool valves for individual metering systems. Spool valves are easily made pressure balanced. The resulting force at the spool is less dependent on the
pressure drop over the spool compared to the force at the poppet in poppet valves. This is why spool valves have been widely used as directional proportional valves in motion control in both the mobile and industrial hydraulic fields. Examples of works where spool valves have been studied as components in individual metering systems are [42] by Yuan and Lew, [26] by Nielsen and [54] by Liu et al. Lin and Akers [55] analysed a servo valve with decoupled spools and concluded that from a dynamic perspective the two-spool configuration is slower but manufacture can be made more cost-effective. Anderson and Li have also studied a two-spool valve type servo valve in [56] where a mathematical model is also presented.

Poppet valves also have a number of advantages. Poppet valves can be designed to have an extremely low leakage when closed. It is possible to integrate other functions, for example chock releases, check valve behaviour, and so on in poppet valves. They also require less precise machining. The design also makes them capable of adjusting themselves as they wear. Since the seat seals the valve, the poppet can be machined with lower diametrical tolerances. The valve can then also be made less sensitive to contamination. Examples of works where poppet valves are used as components in individual metering systems are [39] by Pfaff, [23] and [II] by Eriksson and [57] by Xu et al.

The low leakage makes poppet valves appropriate as meter-out elements. The check valve behaviour can be used to advantage in the system design, for example as anti-cavitation functionality. Generally, poppet valves have a strong pressure dependency. This makes them suitable for pressure control. However, if a position feedback from the poppet is introduced it allows for reduced pressure dependency. In the late ’70s and early ’80s several proportionally controlled poppet valve concepts were presented. There were research activities on different kinds of feedback mechanisms, for example electrical feedback by using position sensors, force feedback using springs, follow-up servo mechanisms, and also combinations of these. See section 5.4 for more on different poppet valve concepts. A survey of poppet valve concepts can be found in Andersson [58] and Backé [59].

Andersson [58] also introduced a proportional controlled poppet valve concept called the Valvistor principle. This valve concept is nowadays marketed and sold by several companies. It is a poppet valve concept with hydraulic position feedback, see section 5.4.1. The feedback mechanism is a slot in the cylindrical surface of the poppet. The pilot flow goes through this slot in series to the pilot valve. When the valve opens up, and the poppet lifts, the slot opens up its flow area correspondingly. It is analogous to an electrical potentiometer. The Valvistor valve as a whole can be compared to an electric bipolar transistor component. It amplifies flow, analogous to how a bipolar transistor amplifies current. The Valvistor concept has been studied in [V] as a component in independent metering systems.

Among the disadvantages of poppet valves are stability issues. The
topic of poppet valve stability has been studied by Funk [60], McCloy and McGuigan [61], Shin [62], Hayashi [63], Muller and Fales [64] and Eriksen et al. [VI] among others.

5.3.2 Control

In contrast to conventional mobile systems with 4/3-valves, individual metering systems have to include some kind of controller for the valves. Conventional systems have one input signal for each load, a spool position. By means of this signal an operator is able to control one output signal in an open loop manner, for example speed or force/torque. An individual metering system incorporates at least two input signals, a meter-in signal and a meter-out signal. To manage the manoeuvring of these systems, a controller needs to control at least all but one input signal. The last one can be controlled by an operator in an open loop manner. For example, the operator controls the meter-in orifice directly and the controller controls the meter-out orifice, or the other way around. Of course, all valves can be controlled in closed loops where the operator generates reference signals to closed loop controllers.

There are a number of parameters/variables to consider when designing an individual metering control system. These include:

Control output variables Different sets of state variables can be chosen as control output. Load speed or actuator force are common output variables. In individual metering systems it is possible to control more than one output signal. For example, load speed and a pressure level in the actuator can be chosen as control signals. Another example of chosen output signals is [II] where valve positions are controlled to try to achieve an open position for energy saving purposes. Mode switching controllers are also commonly used to increase the flexibility of independent metering systems. The set of output variables and/or input signals can then be allowed to change between different modes during operation. Since this kind of system is often considered for its energy-saving capability, energy or power itself could be an interesting control output variable.

Sensor signals (feedback signals) Computer based control systems need sensors to provide feedback signals. A failure, hydraulic or otherwise, can always occur. It is the question of probability and effect of a failure that is interesting. Consequently, the number of sensors should be kept to a minimum and when a sensor breaks it should not cause a hazardous situation. There are both safety and production availability aspects of failures such as a sensor failure. It can also be critical if sensors are inaccurate, for example if a small pressure drop is measured by two pressure sensors. The direction of a pressure drop can then be
misjudged. Such errors can cause a load to drop into the ground for example. It is important to be aware of the risks in complex systems; a tool to use in the design is for example FMEA. Most independent metering systems incorporate pressure sensors.

**Control strategy for supply (pump)** The pump/pumps in a system have to be controlled somehow, as well as the valves. Variable pump systems of today, in particular LS-systems, control the pump hydro-mechanically and separate from the valve control. Independent metering systems utilizing a complex control structure can however take advantage of incorporating the pump control into the system control. This of course requires an electrically controlled pump.

Various control approaches have been implemented and tested in different research works. One approach is to manually decouple the output variables through physical decoupling [35, 65–67]. There have also been efforts to use LQ-techniques [36] and [III]. In [III] the LQ-technique was used to design a state feedback. The strong coupling between the valve signals and the flows were used to simplify the state estimations. Traditional PID-controllers have also been investigated in independent metering systems [46].

A mobile fluid power system is considered to be a tough control application. This is due to all the non-linearities present in fluid power systems. Common non-linearities include flow/pressure characteristics in orifices, valve hysteresis, dead-band and saturation. Fluid power systems also frequently change parameters during operation. In [68] Burrows emphasises the importance of development in the areas of robust control techniques capable of dealing with model uncertainty and parameter variations. He also mentions that non-model-based methods are also interesting for fluid power applications for the same reason.

There are different approaches to handle these issues. One is to design the controller conservatively and assume a worst case operational point. Another is to utilise an adaptive controller scheme to estimate varying system parameters on-line. Stoten and Bulut [69] have implemented the adaptive algorithm MCS (“minimal control synthesis”) in an electro-hydraulic position servo. Plummer and Vaughan [70] have implemented a pole placement control method for an electro-hydraulic positioning system using an adaptive recursive least square scheme to estimate system parameters. Yao [51, 71, 72] has worked on an implementation of the ARC (“adaptive robust control”) strategy for individual metering systems.

Overviews of the research in the area of control of hydraulic systems have been presented by Edge [73], Murrenhoff [74] and Burrows [68] among others.

How to implement the controllers in the sense of hardware and software is also of interest. A hardware and software architecture of a distributed embedded electro-hydraulic system in a telehandler is presented by Yuan et al.
Mode Switching

A two-ported hydraulic load with connections to pump supply and tank needs two orifices to be operated, one to control the fluid at each port. In individual metering systems with multiple choices of possible valves and flow paths for supply and drain of a load, a supervisor controller is needed.

In an early publication in this area Jansson and Palmberg [19] discuss different modes of operation. Liu and Yao [20, 76] split up the control in to a “valve level” controller and a “task level” controller. The operation mode is chosen in the task level controller using the desired load speed and force together with the actual load pressures. Often the reference from the operator is one signal, for example the load speed, then the desired force needs to be calculated. Liu and Yao use an adaptive control scheme.

Pfaff [39] and Tabor [41, 77] both use a telehandler as the application. In that particular application three different modes of operation are used. Also here, the controller task is split up into a “valve controller” and a “mode selector”. It is stated that the mode selector switches mode on the basis of pressure sensors and joystick signals.

In [78] Shenouda and Book show how to find an optimal switching point for a four-valve configuration. They show that the optimal switching point for their application is the intersection of the capability curves, which is the crossover point of the “normal operation” and “regenerative operation” in a force-speed diagram.

The different modes can be classified in several ways. For example, this work has a different classification of modes compared to Pfaff [39] and Tabor [41, 77], see section 5.3.3, [23] and [II]. However, there are a number of operational conditions which recur. One example is when the meter-in load port is connected to pump and the meter-out port is connected to tank, often referred to as “normal mode”. Another is when both mentioned ports are connected to pump, “high-side regeneration”. For the same load, the absolute pressure levels in the chambers are divergent from each other depending on how the ports are connected. The pressure difference for the meter-in chamber in a cylinder between these two modes for the same load is:

\[
\frac{p_{\text{regen}}}{p_{\text{normal}}} = \frac{1}{1 - \kappa}
\]

where \(\kappa\) is the area ratio of the cylinder, \(p_{\text{regen}}\) is the cylinder pressure in regenerative mode, and \(p_{\text{normal}}\) is the cylinder pressure in the piston chamber in normal mode. The pressure in the piston rod chamber is assumed to be tank pressure in “normal mode”. As an example, if the area ratio is 0.5 the pressure difference of the piston chamber is about 50%. Similarly,
there will be pressure differences at the meter-out chamber. A challenge is to avoid pressure peaks and achieve a smooth transition between such metering modes.

Shenouda [40] uses an analogy to automatic transmission gear shifting. He has also done research on a continuous mode switching as a measure for smoother transients during mode switches. This continuous mode switch operation involves operation by three valves simultaneously.

This work proposes a strategy of avoiding the pressure transients by meeting the “high-side regeneration” from “normal mode” through the meter-out orifice increasing the pressure in the meter-out chamber, see section 5.3.3, [II] and [23]. Further, there is a check valve connection from the pump to the cylinder rod chamber, which means that when the pressure reaches the pump pressure, flow will go back to the pump and the mode has switched with no significant pressure peaks.

**Dynamic Considerations**

System dynamics are often crucial in fluid power system design. Even in systems with open loop control dynamic issues can be present, due for example to volumes, flow-pressure characteristics and masses, see Merritt [79].

Modern individual metering systems often contain sensors that feed back signals to the valves via complex controllers. The dynamic system characteristics then become an issue for stability and not only for performance which is the case with open loop controlled systems. In modern electro-hydraulic systems there are extensive possibilities for sophisticated manipulation of feedback signals by computer controllers. This enables the controller to introduce stabilising measures, e.g. dynamic pressure feedbacks.

Typical of individual metering systems is that both flow and pressure control are present simultaneously in closed loops, for instance in cylinder drives there may be pressure control at meter-out chambers and flow control at meter-in chambers. In some cases the circumstances may resemble the over-centre valve configuration, for example when the pressure at one of the load ports is controlled by the flow at the other load port. Over-centre valves have been studied by Persson [80], Andersen et al. [81] and Pedersen et al. [82] among others.

**5.3.3 A Demonstrator System**

This work introduces a novel system design utilizing independent meter-in and meter-out valves that increase energy efficiency in a system that consists of a pump connected to more than one hydraulic actuator, see [II] and [23]. Patents for this particular system have been applied for, see appendix A.

In other proposed solutions, pressure sensors play a key role as regards controllability, see chapter 2. Here, another solution is proposed that is not
dependent on pressure sensors for either flow control or mode selection. The main difference between the work presented in this work and the work done by others mentioned above is the control strategy. The choice of output signals in the closed loops is new. Some functionality is kept in hardware to avoid critical sensor dependencies.

The presented system uses pressure compensators to achieve desired flows. The controller is fed with a speed reference signal by the operator for each load. It then decides which valves to use and controls them in an open loop manner in respect of the flows. When deciding which valves to use the controller will try to minimize the total throttle losses in the system. If possible, the load will recuperate high pressure oil back to the pump, which then can be operated as a motor.

The system will automatically choose from the following operating cases:

**Recuperative operation** In the case of a lowering load, for example, the oil into the cylinder is withdrawn from the tank. The load will then itself pressurize the oil in the opposite cylinder chamber and pump it into the pump line of the system. The cylinder thus works as a pump and can be used to drive other loads or operate the system pump as a motor.

**Energy neutral operation** When the load is not large enough to pressurize the oil above the pump pressure, the system will instead leave it for the tank. As in the “recuperation mode”, the oil into the load is withdrawn from the tank. No power is taken from the system pump. This operating case is also referred to as “floating mode”.

**Regenerative operation** The oil into the load is withdrawn from the pump. If the pressure in the other load port exceeds the pump pressure, the returning oil from the load is fed back to the pump line. If the load is a cylinder load, the cylinder will act as a transformer in this operating mode.

**Normal mode** The highest load in the system will be operated as in a conventional system. The oil is withdrawn from the pump and the return oil is fed to the tank.

**System Description**

This section describes the novel proposed system. Both the hardware and the control laws are shown. Pressure compensated proportionally controlled 2/2 poppet valves of a modified Valvistor type are used, see section 5.4 and also [V], [23] and [58]. The sensors that are used in the control loop are position sensors that measure the position of the compensator spools. No pressure sensors are needed.

The system is designed to meet the flexibility requirements for utilizing energy efficient operation, for example power recuperation and regenerative drive. The different operation cases are shown in figure 5.7 below.
The system has to be able to connect pump or tank to each cylinder chamber independently of each other. The possible configurations are more in detail described in section 5.3.1 and [I].

Systems that utilize independently controlled meter-in and meter-out valves often have most of their functionality, such as pressure compensation, moved into software. In such hardware configurations, the sensors are essential to enable functionality as flow control. See for example [53] and [33].

In this system the ideas are different. The most important functionality is kept in hardware. Pressure compensation, for instance, is kept in the hardware in this system, see figure 5.8. The position of the pressure compensator spools is measured and used both for mode selection and to control feedback signals. The control loops are discussed later. Details of the bi-directional pressure compensated valves can be found in [V].

**Control**

There are two control aims in this novel system:

1. Follow the reference speed given by the operator.

2. Use as little energy as possible from the pump to utilize the desired motion specified by the operator.

The controller is split into two different parts:

1. The operator who controls the speed of the cylinder. The speed control loop is of an open loop control type; the operator himself closes the loop. Because of the pressure compensation of the valves there is no or weak load dependence and cross-talk\(^3\) between the loads.

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\(^3\)Cross-talk in this sense means when changes in one load influences other loads in terms of speed.
2. Choice of which valves to use to achieve the energy saving ability.

There are up to four different choices of which valves to use depending on the load. These choices are listed below and are intuitively ranked from an efficiency point of view, starting with the most suitable:

1. Recuperative operation; energy is gathered from the load and can be used at other actuators in the system or to operate the pump as a motor.

2. Energy neutral operation; no energy is needed to perform the operation.

3. Regenerative operation; a good option if the pump pressure is high at the moment (another load has a higher pressure demand) so that less flow is then used from the pump.

4. Normal mode (The conventional way); has to be used at the load with the highest pressure demand. By using independent meter-in and meter-out valves the meter-out pressure drop can be smaller than usual. In this work meter-out always denotes where flow leaves the cylinder and meter-in always relates to flow entering the cylinder.

The condition that has to be fulfilled to be able to operate in the recuperation operation case is that the load is large enough to build up a higher pressure in the cylinder chamber than the pump pressure, and of course the load has to act in the desired motion direction. Since pressure compensators are used for flow control the position of the spools in the compensators pro-

![Image](image_url)
vides information about whether a valve is capable of delivering flow in a certain flow direction or not.

If the condition to operate in the recuperation operation case is not fulfilled, the system will try to operate in the energy neutral operation case. This operation case can be analogously described as the recuperation operation case; the difference is that the oil is delivered to the tank instead of the pump.

These two operation cases, recuperation and energy neutral operation, are here called meter-out control mode since the speed (or flow) control is effected at a meter-out valve. This is visualised in the left part of figure 5.7.

When neither of these operation cases can be used, the system will start using oil from the pump. The speed control is now effected at the meter-in valve, see the right part of figure 5.7. As mentioned previously, the regenerative operation case is preferred to the conventional operation case, at least when the present load is not the heaviest one. This can be realized by controlling the position of the compensator spool of the meter-in valve with the meter-out valves. If the compensator at the meter-in valve is controlled to be held in a relatively open position, this is the same thing as saying that the pressure drop over the same valve is controlled to be held low. The pressure level in the cylinder chambers is then as high as possible for the given pump pressure. If the load is small enough the pressure in the meter-out chamber of the cylinder is higher then the pump pressure and the regenerative operation is automatically enabled; otherwise the meter-out valve to tank has to be used.

Both meter-out valves, the one from the meter-out cylinder chamber to tank as well as the one connected from the meter-out cylinder chamber to pump, are used to control the position of the compensator at the meter-in valve in a closed loop manner.

By considering the choice of reference positions of the control loops of the compensator positions the regeneration operation case can be prioritized over the conventional operation case. This is explained in more detail in paper [II].

The idea of this system’s control system can be represented by a state machine describing how modes are selected. See figure 5.9.

The structures of the control loops are shown in figure 5.10. The mode selector chooses which valves are to be active according to figure 5.9. The different control algorithms are activated as shown in the software box in figure 5.10. The closed loops in the software block are the control of the compensator positions that are fed into the software from the compensators in the hardware. The joystick signal controls one of the valves as described above in an open loop manner.

So far, pump control has not been discussed. It is desirable that the pump should be controlled only by the load with the highest pressure demand. Analogous with the meter-in control mode, the pump controller is set to

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4The position of the compensator spool gives information about the direction of the pressure drop.
Too low pressure drop over meter-out tank valve

Initial state, \( \dot{x}_{\text{ref}} = 0 \) (no motion)

Too low pressure drop over meter-out tank valve

**Figure 5.9** – Principal state machine description of the control scheme. (Faded valves are inactive.)
control the compensator position of the meter-in valve. To get the right prioritization, the reference value is set so that the pump “kicks in” after the meter-out is fully open. To get better response it is also desirable to feed the flow needs forward to the pump controller. The flow needs can be estimated from the joystick signals and the system controller.

**Measurements**

A proof of concept has been carried out. The control strategy described in the previous sections has been implemented in a real-world application, a forwarder machine. Fast spool valves were used to emulate the valves in a final system.

One of the functions in the forwarder is replaced with individual metering valves. Both the demonstrator and the valve configuration are shown in figure 5.11. The jib function is chosen in this case. Six proportional valves

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*Figure 5.10 – This is an overview of the control loops in the system. The electrical feedback signals are the position of the compensator spools, which are the signals from the hardware back into the software block in the figure. The inner flow control loop is kept in the hardware by the compensators. The software is split into two parts; it chooses which metering valve to use and also controls the meter-out valves in the meter-in mode in a closed loop manner. The signal from the operator controls the flow magnitude in an open loop manner; the loop is of course closed by the operator. $\Delta p_{\text{ref}}$ is a constant and is related to the pre-load of the spring in the compensators.*
with additional pressure compensators were used to enable the functional
demands of the system. LVDT sensors were used to determine the positions
of the compensator spools.

![Diagram](image)

**Figure 5.11** – The demonstrator and the system layout used for the initial test of the
system schematically described.

In order to test the valve system ability, the jib function was selected when
it runs over centre. This allows the valves to use most of the possible modes
shown in figure 5.7. However, the modes where the flow demand from tank
exceeds the return flow to tank were disabled due to the lack of pressurized
oil in the return line. The pump in the tested system was a conventional
pump and could thus not be used as a motor. This restricted the modes
where oil is fed back to pump. In the test the pump was set constant to
emulate pressure requirements set by other, heavier, loads at other functions.
The pressure level was set to enable interesting switching between different
modes.

The signals displayed in the graph, figure 5.12, are the valve displacement
signal, the pump flow, the absolute value of the flow in/out of piston chamber A, and the angle of the jib arm. The first part of the graph illustrates jib
out motion\(^5\). The motion starts in energy neutral operation, since the press-
ure drop from chamber A to tank is higher than a predefined level\(^6\). When
the pressure drop is too low, indicated by the compensator, pump flow is
added to chamber B and the system passes into normal operation. Observe
that the initial part of the motion does not consume any pressurized oil from
the pump.

The second part of the graph, figure 5.12, illustrates jib in motion\(^7\). The
motion starts in regenerative operation, since the pump pressure\(^8\) is high

\(^5\)Equivalent to cylinder retraction.

\(^6\)This level corresponds to a compensator position on the AT valve.

\(^7\)Equivalent to cylinder extension.

\(^8\)The pump pressure level is decided by another, heavier, fictive load.
enough to use the cylinder as a discrete transformer. When the pressure drop over the meter-in valve becomes too low, indicated by the compensator spool position, the operation is changed continuously from regenerative to normal. Observe that the regenerative part\textsuperscript{9} of the motion consumes only about one third as much pressurized oil from the pump as in normal operation.

\textbf{Figure 5.12} – Measurement results from the demonstrator. The jib cylinder extends between 0 and 15 s, then the cylinder is stopped and retracts between 20 s and 35 s.

\textsuperscript{9}From 30 s to 32 s in figure 5.12(b).
5.4 Poppet Valves

Energy efficient systems, such as those described earlier in this chapter, need a larger number of valves than conventional systems. On the other hand, simpler valves can be used. 2/2 proportionally controlled valves meet the requirements of this system.

The most common proportional valves are spool valves. Spool valves are suitable for traditional systems because of the simplicity of control compared to poppet valves. More than one orifice is, by the mechanical connection, often controlled by only one actuator. In this new kind of system the different orifices are individually controlled. Therefore, the mechanical connection in spool valves at the orifices can not be utilized as an advantage.

Poppet valves are a suitable choice for individual metering systems since their design is often simpler than a spool valve design and is of the 2/2 type. They also often preferably have sealing properties. Unfortunately, they are also often of the on/off type. However, there are a number of designs that allow proportional control. Some examples are shown in figure 5.13 and are briefly described below.

**Electric feedback servo** This principle uses an electrical signal from a position sensor as feedback. The properties of such a design are mainly determined by the electrohydraulic actuator, statically and dynamically. It is a flexible configuration. A drawback is that it is totally dependent on electronics. See figure 5.13(a).

**Force feedback servo** The position control loop in this design is closed by the spring arrangement between the pilot valve and the main poppet itself. When the main poppet lifts, the pre-load of the spring between the poppet and the pilot is increased. Depending on the applied external force, $F_{input}$, the steady state occurs at different main poppet openings. The input force is often realized by a current-controlled solenoid. See figure 5.13(b).

**Mechanical feedback servo (Follow-up servo)** In this design the pilot orifice is located directly in the main poppet. A rod then controls the opening degree of the pilot orifice; it is a needle orifice. Moving the needle causes the pilot orifice to open. The pressure then decreases in the control chamber above the poppet and the poppet lifts and closes the needle orifice. The main poppet then follows the pilot rod. The position of the rod can for example be achieved by using a step motor, a solenoid, a hydraulic pilot arrangement, etc. See figure 5.13(c).

**Combined force feedback and follow-up servo** There are solutions that utilize both the follow-up and the force feedback mechanisms, for example to increase the stiffness in the position control loop of the main poppet.
Figure 5.13 – Different designs of proportionally controlled poppet valves. $p_{\text{low}}$ is the low pressure side, e.g. tank/load port, $p_{\text{high}}$ is the high pressure side, e.g. load/pump port. $y_{\text{ref}}$ is the desired poppet position and $F_{\text{input}}$ is a pilot force.
Hydraulic position feedback (Valvistor principle) The valvistor principle utilizes the position of the main poppet to control an orifice. By force equilibrium of the main poppet, the opening of the orifice mimics the pilot valve opening. The result of the mimicking behavior is that the flow through the main poppet is an amplification of the pilot flow. See figure 5.13(d).

In this work the Valvistor concept has been studied and used as the valve element in the individual metering system in section 5.3.3. The main reason for using this valve is its ability to keep functionality in small valves in the pilot circuits. Another reason is that it is relatively simple to make bi-directional, see section 5.4.1.

5.4.1 The Valvistor Valve Principle

The Valvistor valve was originally developed at Linköping University in the early 1980s by Andersson, see [58]. The Valvistor principle was first produced by Hydrauto, a Swedish valve manufacturer. Other research on the Valvistor valve has been conducted after Andersson, e.g. by Pettersson, who developed the “twin Valvistor” which is a design that increases the Valvistor’s bandwidth by putting a Valvistor valve in the pilot of another Valvistor [83] in [84]. There have also been interest in the valve principle from different companies over the years.

Working Principle

The Valvistor principle is a proportional control principle for poppet valves. Characteristic of the Valvistor principle is the variable orifice that connects one port to the control chamber above the poppet, see figure 5.14. This orifice is often a rectangular slot; the opening area of the slot is then proportional to the opening stroke of the poppet itself. Figure 5.14 shows a Valvistor in the standard design. The slot closes a hydraulic control loop for the position of the main poppet. The force equilibrium yields the pressure in the chamber above the poppet. The slot orifice then assumes the position that corresponds to the opening area so that the right pressure drop is met.

The pilot flow drains the control chamber of oil at the same time as the slot orifice delivers oil into it. To fulfil the force equilibrium the pilot valve opening and the slot orifice opening are equal, assuming that the same flow regime is present at the different orifices. This means that the opening of the slot orifice mimics the pilot valve opening. The main poppet position is controlled by the opening of the pilot valve. The valve acts as a flow amplifier, similar to the transistor component in the electrical world. The ideal flow gain is described by equation (5.2)

$$g_{\text{ideal}} = 1 + \frac{w_m}{w_s} \frac{1}{\sqrt{1 - \kappa}}$$  \hspace{1cm} (5.2)
Figure 5.14 – The Valvistor valve.

were \( w_m \) is the area gradient of the main orifice\(^{10}\) and \( w_s \) is the area gradient\(^{11}\) of the slot in the main poppet. \( \kappa \) is the area ratio of the inlet area of the poppet and the control chamber area, \( A_m \), of the poppet. A determining factor that disturbs the ideal flow gain is the underlap in the slot. The underlap in the slot means that the slot is already slightly open when the main poppet is in its closed position. This underlap is necessary for stability reasons, see paper [VI]. It adds a negative contribution to the ideal gain in equation (5.2) that is dependent on the main poppet position. However, there are ways to deal with this phenomenon. For example, the slot can be designed in a non-rectangular shape that compensates the non-linear relationship due to the underlap. It is also possible to add a non-compensated flow through the pilot valve that cancels out the flow through the underlap during operation of the valve. This non-linear characteristics due to the underlap is studied in paper [VI].

Since the main poppet mimics the pilot valve a pressure compensated pilot valve results in a pressure compensated main stage. In such a design care to the non-linear characteristics discussed above need to be taken into consideration. This valve is able to handle substantial flows due to the size of the actuated pilot valve. In some commercial valves, flow forces are utilized at the pilot valve to make it pressure compensated.

The description of the Valvistor principle so far is true only for one flow direction. In the opposite flow direction the valve acts as a check valve. This feature can be utilized as an anti-cavitation function for example. In the

\(^{10}\)If no soft opening arrangements are present at the poppet, the area gradient will be equal to the diameter.

\(^{11}\)If the slot is rectangular, the area gradient is equal to the width of the slot.
application of an individual metering system, the valves need to be proportionally operable in both flow directions. The next section describes how the Valvistor principle can be extended to handle the bi-directional issue.

The dynamic properties of this type of valve are dominated by a first order effect related to the exchange of oil in the control volume above the poppet through the slot orifice and the pilot valve. This break frequency is described approximately by equation (5.3).

\[
\omega_b = \frac{C_q w_s}{A_m} \sqrt{\frac{2}{\rho} (1 - \kappa) (p_B - p_A)}
\]  

(5.3)

More details on the dynamics of the Valvistor principle can be found in [VI]. An overview of general valve modelling can be found in [79].

Compared to other similar valve solutions, such as poppet valves utilizing follow-up mechanisms, the Valvistor principle has the advantage of high loop gain in the inner hydraulic control loop. This feature is enabled by the characteristic variable slot orifice in the surface area of the poppet itself.

**Bi-Directional Valvistor**

To make the Valvistor principle suitable for the kinds of system described in this work it has to be able to be proportionally controlled in both flow directions.

By merging the A- and B-types, figure 5.14, and adding check valve mechanisms in the main poppet that connect the port with highest pressure to the control chamber, and also merge the pilot circuits from both A- and B-types, the valve turns into a B-type when the pressure drop is positive in the B-to A-port and to an A-type when the pressure drop is positive in the A- to B-port. The result can be seen in figure 5.15. Pressure compensators are also added at the pilots in this figure.

The valve in figure 5.15(a) is a bi-directional, proportional flow controlled pressure compensated poppet valve. The valve has another attractive feature as well: there are two parallel pilot valves. The left one in figure 5.15(a) is used to control flow in the A to B direction. The other one is used to control flow from B to A. If the pressure is higher at B than at A and the left pilot is used, no flow will occur and vice versa. As a consequence, no flow will ever occur in an undesired direction, such as a falling load. This can be a crucial detail of an independent metering system.

Let us look at an example where high pressure is present at the B-port and the pressure at the A-port is lower. If the right-hand side pilot valve in figure 5.15(a) is actuated, a flow will occur over the poppet from the B- to the A-port. The left check valve in the poppet will be held open at the same time as the right one will remain closed. If the left pilot valve is actuated instead, the check valves in the poppet will remain the same and the opening of the left pilot valve will not change anything. It will just connect the B-port to
Energy Efficient Mobile Systems

5.15(a) The bi-directional Valvistor valve.

5.15(b) Schematic view.

Figure 5.15 – The modified Valvistor valve.
the control chamber in the same way as the channel in the poppet already
does. This feature is usable in a system to prevent, for example, falling loads
because flow in undesired directions will not be present.

The orientation of the bi-directional Valvistor valve also has to be consid-
erned. It has not exactly the same characteristics in both flow directions. The
leakage is different in the different flow directions. In flow direction B to A
in figure 5.15(a), the valve is almost leakage-free. When pressure is higher
at B than at A, the pressure at B is then connected to the chamber above the
poppet through the check valve. Since there is no pressure drop over the
leakage path around the poppet there is no leakage flow. However, in the
other flow direction, from A to B, the pressure drop over the poppet surface
will be the pressure difference between the A-port and B-port pressures and
leakage will occur. Often it is sufficient to have a valve leakage-free in just
one direction. For example in a crane, the load almost always acts in the
same direction. Then the B-port should be facing the load.

The valve in figure 5.15(a) can be described schematically by the valve
symbol in figure 5.15(b).

5.4.2 A Demonstrator Valve

As proof of concept, measurements were made at the pressure compensated
valve using the Valvistor principle. The results are shown in figure 5.16. The
agreement of the measurements and the calculations are rather good.

![Image](image.png)

**Figure 5.16** – Measurements at the compensated Valvistor valve compared with a non-
linear static model.
Simulation is an important tool that is used in the work of discovering and exploring new ideas and concepts. In this work, primarily Matlab/Simulink™ and AMESim™ have been used. To some extent the simulation package HOPSAN has also been used. This part of the work is to some extent parallel to the work described in the previous chapters. The part of this work which is described in this chapter concerns a simulation tool itself. This is motivated because the need for a proper simulation tool becomes obvious when working on fluid power systems such as those described in chapter 5.

6.1 HOPSAN

The HOPSAN simulation package, used primarily for hydro-mechanical simulation, has a long history and was first released in 1977.

From the beginning, HOPSAN used a central solver, as most other simulation tools did, and still does. The HOPSAN versions with a central solver were inspired by the csmp/360 software from IBM. In the early ’90s, transmission line modelling, TLM, was adopted. TLM, or bilateral delay line modelling as it was originally referred to as, can be traced back to the sixties, where it was used for simulation of distributed systems, [85]. This method means that solvers can be distributed\(^1\) and also come with some other benefits, for example improved scalability\(^2\) and higher numerical robustness, see for example [86] and [87]. One consequence of the distributed nature of TLM is

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\(^1\)This means that individual component models are isolated from each other so that their equations can be solved independently.

\(^2\)With TLM, the computational cost scales linearly to the size of the problem.
that the equations are not flattened, as is the case in for instance the Modelica language. The initial TLM implementation was influenced by HyTRAN, [88].

HOPSAN has been successful ever since it was first released in 1977. It has mainly served as a research tool at the division of Fluid and Mechatronic Systems, but has also been used by a number of companies for high performance simulations.

HOPSAN’s source code is rather complex and old-fashioned. The non-GUI code, the core, is written in FORTRAN and C while the GUI code is written in Visual Basic and has been characterized by organic growth over several decades. The ageing code makes it difficult to maintain. It is also an obstacle to further development. Because of the need for improvements, work has begun on a new version of HOPSAN. This work has been a part of that. At the time of writing, the new version, HOPSAN NG is almost ready for users. HOPSAN NG is still based on TLM and retains all of the benefits that come with this method.

The largest difference compared to the current version of HOPSAN is that all component models are now written in C++ and are pre-compiled before being added to a simulation system model. In earlier versions, models were written in FORTRAN and converted to C before compilation at every simulation run. The GUI application itself is more coherent and contains all HOPSAN tools within one application.

6.1.1 C++ and the Qt Framework

In an attempt to avoid the complexity and limitations of the old HOPSAN code-base, development of the new version primarily uses open-source tools and only cross-platform supported languages. The new HOPSAN package is still divided into a core and a GUI. The core is written in standard C++ and the use of external third party dependencies is avoided. The GUI is also written in C++, but uses the Qt cross-platform application and UI framework, see [89]. The Qt framework, released under the LGPL license, is easy to use, well documented, and offers plenty of features useful for GUI creation.

6.1.2 Hopsan Core

The HOPSAN Core is an object-oriented C++ library containing the basic HOPSAN classes and functions and a set of basic component libraries. It also includes utility function libraries that can be of use when users are writing their own components. A main program, for instance HOPSAN GUI or some other application, loads the library and instantiates an API class through which all core communication is handled. The object-oriented approach gives a natural containment of equations and variables into independent objects for each implemented component model.
As the core is a library, it can be used in any “main program”. This means that it is possible to run simulation models created in the GUI using only the core and a small dedicated application. This may be useful for simulation in embedded systems where a graphical environment may not be present. It also means that the core can be included in other applications to simulate HOPSAN models.

**User Defined Components**

The HOPSAN Core comes with a basic component library useful for building systems. For special applications and simulation of non-standard components, the user must be able to create additional components. In HOPSAN NG, external components are shared libraries which can be dynamically linked in at runtime. Components must be written in C++ and are compiled separately. This means that the models can be distributed as black box models. In the case of a company they can distribute simulation models for their products to customers without revealing the underlying mathematical description (or maybe controller code and similar). In the current development version, new components are created by following a rather simplistic template. The user only needs to specify a few things:

1. The type of component, whether to inherit the C, Q or S component class.\(^3\)

2. A unique component type name, for instance “MySpecialOrifice”.

3. Local variables and which of them should be registered as component parameters.

4. The kind of ports that should be available and what type of nodes they require.

5. What the component must do at the initial time-step to properly initialize the connected nodes.

6. The equations for simulating one time step.

7. A line of code that registers the component with the global component class factory in the core.

As component creation follows a simple template, it is possible to write an automatic component generator to assist the user in the modelling process. A component generator for transforming a sub-set of Modelica code via Mathematica into HOPSAN components exists, [90]. Based on this a Mathematica\(^\text{TM}\) to HOPSAN NG component generator is already under development.

\(^3\)The C, Q or S types are related to TLM, see paper [VII].
6.1.3 Hopsan GUI

The Hopsan GUI is based on the Qt cross-platform application and UI framework. The cross-platform support in Qt means that the drawback with the current Hopsan version, which is only available for Windows, is avoided. The GUI basically consists of graphical representations of the most important core objects, components, and system components. These can be moved by dragging-and-dropping and connected to each other with lines. A screen-shot of an example model and the Hopsan GUI interface is shown in figure 6.1. The user interface and the possible actions are similar to those in the current version. The look and feel of the GUI application are also similar to those of many other modern graphical simulation tools.

The graphical appearance of the modelled components are loaded from external plain text files and scalable vector graphics (SVG). The actual model code, however, is handled by the Hopsan Core library.
6.1.4 Scripting and Optimization

An important feature in a simulation program like HOPSAN NG is the ability to optimize parameters. For this the Python\(^4\) language is used. HOPSAN NG interacts with Python via a console integrated in the HOPSAN GUI. This enables the user to script his/her models freely. For example, the users can use their own or third party modules for optimization. This is where the scalability and numerical stability of TLM comes into its own.

References to work on optimization where Hopsan simulation models were included in the optimization loop can be found from as long ago as the ’90s, for example [91].

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\(^4\)Python is a high level programming language. There are also free mathematical packages available that have similar functionality to Matlab.
Discussion and Conclusions

Energy efficiency in mobile fluid power working hydraulic systems can be improved. There are a number of measures which can be applied to reduce losses in these kinds of system. This work studies flow controlled systems and independent metering systems. They are compared with conventional load sensing (LS) systems which are state-of-the-art in industry today.

Displacement controlled systems are not included in this work. However, some remarks regarding their comparison to individual metering systems are necessary nonetheless. Often, when comparing displacement controlled systems with various valve controlled systems one thing is rarely mentioned: the cost. There is a need for several pumps/motors instead of one. Since the load functions have their own dedicated pump/motor, each has to be sized for maximum flow. A typical example of a dimensioning motion is the speed of the lowering bucket motion on a wheel loader. In the wheel loader example, the lowering flow from the boom cylinder can be several times the maximum pump flow in a similar valve controlled system. The difference is that for the pump controlled system, all flow has to be handled by the pump/motor. Proportional valves can be a part of a solution by bypassing the pump/motor a share of the lowering flow. Note that it has then become a hybrid of a displacement and a valve controlled system. Further, a displacement controlled system, often referred to as valveless, often needs additional valves to meet safety requirements and leakage issues. This also adds complexity and cost to the system. The conclusion is that when displacement controlled systems’ efficiency is compared with valve controlled systems’ efficiency it is important to consider the cost. In a given comparison, the equivalent amount of effort and resources have to be put into both systems. Likewise, compare
different concepts fulfilling the same working cycles in terms of speed and force. A benefit of valve controlled systems is that the number of pumps can be kept low considering the number of controlled actuators. However, the energy losses can be reduced further if the number of pumps is increased, but with the same total flow capacity.

A study based on measurements from a wheel loader shows the potential for reducing energy losses in that particular application. It shows for example that individual metering systems can reduce the energy consumption significantly, about 25%. To introduce individual metering systems in a system is a major challenge. There are a number of difficulties to overcome. For example, this work studies the challenges of control, mode switching, system layout and choice of valves in these systems. Most of the challenges are related to the flexibility of these systems. Flow controlled systems also have properties that give them advantages over LS systems in terms of energy efficiency and dynamic characteristics. This kind of system has also been studied in this thesis as an alternative improvement to LS systems. The step from LS systems to flow controlled systems is in some senses more natural then introducing individual metering systems. Therefore, those systems have also been studied to some extent in this work.

The discussion and conclusions are divided naturally into three sections.

7.1 Flow Controlled Systems

A flow controlled system is more energy efficient compared to a traditional LS system. There are no superfluous pressure throttle losses over the compensators due to the pump pressure margin. Since the pump is not in a closed control loop, there are potential energy savings tied to the absence of active control of the pump.

The system design process can be made simpler with a flow controlled pump because no considerations are needed for pump stability due to the LS feedback and system dynamics. LS systems often show oscillative behaviour due to the LS feedback to the pump controller. Flow controlled pump systems have no stability issues attached to the LS feedback since there is none.

Flow matching of the pump to the loads is critical in systems with flow controlled pumps and traditional pre-compensators. Flow sharing compensators are essential in systems with flow controlled pumps since no flow matching is needed for the system to work properly. Flow sharing can be realized, for example, by using post-compensated valves. There are also solutions available using pre-compensation.

Pre-compensators have the advantages that they are simpler to install and that most existing systems already use pre-compensators. In some systems, pre-compensators can be more or less replaced with flow sharing pre-compensators without replacing the valve housing.
Some fundamental design rules regarding the flow sharing pre-compensators are presented. Solutions for handling both auxiliary functions and critical functions are shown. One advantage of these solutions is that there is no need for sensors to achieve desired functionality. Tests have been performed to show the capability of the flow control approach. For example, reduction of the superfluous throttle losses at the compensator with the highest load has been demonstrated.

### 7.2 Independent Metering Systems

Issues in the field of independent metering systems are reviewed and an overview of current and recent research is also given. Research in this area has been going on for a long time. Just now, the time is right for industry to utilise the technology. This can be seen from contemporary research projects around the world.

Different hardware layouts and control approaches are feasible for independent metering systems. Pros and cons with the different possible configurations are given. Important advantages offered are reduction of losses, mode switching capability and improved control ability. The price for these advantages is more demanding control functions, often requiring more advanced software and a need for more sensors in the system. Conventional systems can be designed hydro-mechanically without the need for any sensors. The most appropriate system design depends on the application and its specific demands.

Independent metering systems contain fairly complex dynamic behaviour, especially in systems where mode switching controllers are applied. Computer simulation is therefore an important tool and has a vital role to play in the design and evaluation process.

In the fluid power area pressure feedback has been a traditional measure to increase damping in conventional siso-systems with spool type directional valves. This work has shown that the LQ-technique is a successful method for designing similar damping measures for mimo-systems such as independent metering systems. An implementation was performed in a forwarder application.

The operational conditions in a fluid power system change considerably during operation. In spite of this, controllers are usually designed for the worst case scenario, for example with the heaviest load and smallest valve openings. Systems with this kind of controller are often sufficient. In this work it is shown that it is not necessary to model the system fully with all the features and dynamics to achieve a sufficient and satisfactory controller. The example is an LQ-controller implemented at the swing function of a forwarder. It is possible to take advantage of modern control theory for multi-variable systems without adding too much complexity. It is important
to analyse what the main problems are before beginning to model and implement too complex controllers. It is shown that by using an “over-simplified” linear model, simple but sufficient and useful controllers can be designed for fairly complex systems.

A novel control strategy for an independent metering system is proposed in the thesis. The system is designed in a flexible manner to enable energy efficient operation, for instance recuperation, regeneration and energy neutral operation. Due to the choice of control variables, only a few mode switches are needed to utilize these operations. Measurement shows that the system operates smoothly and switches automatically between modes to minimize the throttle losses in the system. In a “jib in” motion, for example, the first part runs in regeneration operation due to the relatively high pump pressure (caused by another load). Only about one third of the flow is then needed in the regenerative part of the motion for the same cylinder speed.

### 7.3 The Valvistor Concept

The Valvistor valve concept has been chosen for the proposed system. The concept has been studied and further developed in the work. The final valve concept used is an alternative bi-directional design. It has some unique properties that makes it feasible for systems with flexible flow directions. Because of the arrangement of double check valves inside the poppet and the double pilot circuits it guarantees that flow in undesired flow direction will never occur. This property is essential for human safety in systems that handle serious loads such as forwarders, wheel loaders, excavators and so on.

This type of valve can be designed for a high range of flows due to high flow gain capabilities, typically several of hundreds of litres.

A valve utilizing the Valvistor concept may be seen as a flow amplifier. As this kind of valve is a true flow amplifier it means that the main stage mimics the pilot characteristic but amplified by a gain factor. This is often used to create different functions such as pressure compensation on the pilot stage, which is an advantage compared to doing so on the main stage of the valve. The amplification gain is a design variable and is typically 30-70.

The Valvistor can be modelled as a first order system with a break frequency which is proportional to the square root of the pressure drop (typically 50 Hz at 10 MPa pressure drop and a flow gain of 40). Due to the high bandwidth of the Valvistor the pilot dynamics often dominate the valve as a whole. It is shown that, if the valve is properly designed, higher order dynamics can be ignored. However, with pressure compensated pilot valve, the underlap of the slot in the main poppet, must be considered.

The fundamental properties of the valve such as flow gain and bandwidth, allow the same pilot valves to be used in a wide range of flow capacities of the main valve. Only the flow gain needs to be changed. This is done by changing the geometry of the feedback slot in the main poppet. Actuation
forces of the pilot valve can be kept small by the use of pressure compensators in the pilot circuits. The pilot valves can also be kept small if the flow gain is chosen to be high. This is a trade-off in the design because the dynamic response is also a function of the feedback slot. Higher flow gain means lower bandwidth of the valve.

The dynamical properties for proportionally controlled poppet valves are important to consider in the design. A linear dynamic model of the Valvistor valve is presented which describes the flow characteristics. By fair simplifications the model ends up in a favorable first order system. The model can be used to size valves and perform analysis of closed loop systems.

7.4 Hopsan NG – the New Generation Simulation Tool

It is shown how TLM can be implemented in a modern way using the widely known C++ language and the Qt GUI framework.

Large and/or fixed time steps can be taken without jeopardizing the numerical stability due to TLM. Hopsan NG is therefore suitable for high-speed simulations which are needed in real-time applications and simulation in optimization loops.

The object-oriented implementation of Hopsan NG maps the physical world. This makes it intuitive for users to implement their own custom components.

In Hopsan NG, TLM makes it possible to use distributed solvers and pre-compiled models. This makes the simulation procedure faster as no compilation is needed before it begins. It also enables models to be shared in a binary format. This is an advantage for industry, which often deals with propriety simulation models.

The distributed way of modelling in Hopsan NG enables verification and testing of sub-models independently of each other.

Errors are traceable in TLM models since they are not flattened as they are in for instance the Modelica language. This simplifies debugging for example.
Outlook

The research area of which this work is a part has expanded in recent decades. Both universities and companies put effort into the area of energy efficient fluid power. This will most likely continue.

In the near future the mobile hydraulics market will probably see more of flow controlled systems where the load sensing pump is omitted. Challenges in these systems are how, and to what extent, sensors should be used. Sensors are needed, for example to detect end stops at cylinders. A pump controller without feedback from the consumer needs to be aware of the situation and reduce the flow in such cases.

In a longer term, individual metering systems are upcoming. They are already on the market, but there are only a few providers and the systems are still at the development stage. Individual metering systems have superior possibilities for control compared to conventional systems. This is because of the increased number of input signals\(^1\) to the system. But how these signals should be controlled is not a trivial matter. There are openings for smart control strategies for these systems. More advanced control theory can be applied. A challenge is to keep safety at a sufficient level at the same time as the valve signals are subjected to sensor signals and complex control algorithms.

Also, hybrids of individual metering and displacement controlled systems are of interest for the future. In these, a system with for example four loads can share two pumps. The loads are individually valve controlled but split up in groups so that the part load pressure losses are minimized.

Energy storages, for instance, can be added to all the above mentioned concepts for energy use reduction as well. The control problem can then be formulated in a delicate manner.

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\(^1\)With input signals the valve openings are intended. Those are typically one for a conventional system and two or more for an individual metering system for each load.
Simulation tools will have a central role to play in all this. Tools like Hor-san NG which is able to execute models in real-time and beyond, will see a bright future when the need for optimization using simulation models becomes even more important than today.
Review of Papers

Paper I

Individual Metering Fluid Power Systems: Challenges and Opportunities

This article gives an overview of recent and current research on individual metering fluid power systems. Different systems and their pros and cons are studied. General challenges related to independent metering fluid power systems are discussed. The major choices in the design of these systems are the hardware layout and the control strategy. The evolution of existing independent metering fluid power systems from the 1970s until the present day is also presented.

Paper II


This is one of the earliest publications of the proposed energy efficient individual metering system shown in chapter 5, section 5.3.3. This paper introduces the novel system design, which utilizes independent meter-in and meter-out valves. This system design has the potential to increase energy efficiency considerably in a system that consists of a pump connected to more than one fluid power actuator. The system proposed here is not dependent on pressure transducers for either flow control or mode selection. The main difference between the work presented in this paper and earlier work is the control strategy. The output signal choices in the closed loops are new. Some
functionality is kept in hardware to avoid critical sensor dependency. The presented system uses pressure compensators to achieve desired flows.

Paper III

Energy Saving System Utilizing LQ-Technique Design

This paper shows how the linear quadratic (LQ) framework can be used in the control of individual metering systems. Since an individual metering system is more flexible than a conventional system, there are more control signals and thereby more outputs to control. This opens up for use of more sophisticated control laws, for example the LQ-technique. Energy saving aspects are among the main reasons for the research on this kind of system, but there is also an opportunity for improvements in the dynamics compared to a conventional system. In this paper an approach using the LQ-technique is presented for improvement of system dynamics. Since all states in the system can not be measured a state observer is also considered in the control design. The paper presents simulations, implementations in a real world forwarder application and results from verifying experiments.

Paper IV

How to Handle Auxiliary Functions in Energy Efficient, Single Pump, Flow Sharing Mobile Systems

This paper studies an interesting flow dividing system that is an alternative to conventional load sensing (LS) systems. The studied system uses pre-compensated valves with flow sharing properties. The fundamental difference between a conventional LS system and a flow controlled system is that the pump is controlled based on the operator’s total flow demand rather than maintaining a certain margin pressure over the maximum load pressure. One of the main advantages with flow controlled systems is the absence of the feedback of the highest load pressure to the pump. Flow controlled systems also present some challenges, one being how to handle auxiliary functions with unknown flow demands. Auxiliary functions are typically support legs, external power takeouts etc. This paper analyses one kind of flow controlled system, gives some design rules, and shows one way of dealing with auxiliary functions.
Paper V

A Novel Valve Concept Including the Valvistor Poppet Valve

A novel design of the pilot circuit in the Valvistor valve, suitable for independent metering systems, is presented in this paper. The design allows flows in both directions; the valve is bi-directional. It also prevents flow in undesired flow directions. The functionality remains of a check valve that prevents back flow in conventional systems. The valve is also pressure compensated in both directions. The pressure compensators are placed in the pilot circuit. Because of the relatively small pilot flow, these pressure compensators can be made small compared to the main poppet.

Paper VI

The Dynamic Properties of a Poppet Type Hydraulic Flow Amplifier

The Valvistor valve concept is a proportionally controlled poppet valve. When designing such valves it is important to be familiar with the dynamic properties. This paper studies the dynamic properties of the Valvistor valve analytically. It begins with a complete complex valve model that is narrowed down to a simplified model. The dominating frequencies are also related in a somewhat normalized fashion to, for example, material and geometrical properties of the poppet itself.

Paper VII

Hopsan NG, A C++ Implementation Using the TLM Simulation Technique

The original HOPSAN simulation package was first released in 1977. Modelling in HOPSAN is based on a method using transmission line modelling, TLM. No numerical errors are introduced at simulation time when using TLM; all errors are related to modelling errors. This yields robust and fast simulations where the size of the time step does not have to be adjusted to achieve a numerically stable simulation. This is preferable in optimization applications. The distributive nature of TLM also makes it convenient for use in multi-core approaches and high speed simulations. This paper presents the development version of HOPSAN NG and discusses some of its features and possible uses.
References


References


This PhD work has resulted in two international patent applications being filed. In this appendix the first pages of the international applications\textsuperscript{1} are shown. In both applications the inventor is signed and the applicant is the company Parker Hannifin AB, who were a cooperation partner in this work.

\textsuperscript{1}International application is the same as a PTC application.
A.1 Individual Metering System

The system presented in [23] and was considered interesting by the cooperation partner and resulted in a patent application. The patent PTC application is shown here. It has also been filed in Finland, Germany, South Korea and USA. The application numbers in those countries are: fi20096407, de112007003562, kr20100031589 and us12/665744.

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The invention relates to a method for controlling a fluid valve arrangement comprising: a fluid conduit arrangement having a first conduit and a second conduit, the first and the second conduits being connectable with a fluid consumer; a supply connection arrangement having a pressure connection and a tank connection, and a number of valve arrangement for connecting the consumer to the pressure connection or the tank and a control device, which controls the valve arrangements. The method involves detecting output signals from opening degree sensors connected across at least the first and second valve arrangements to determine the magnitude and direction of a pressure drop across at least the first and second valve arrangements, using the said output signals. The fluid consumer is then controlled on the basis of the output of said sensors.
A.2 Bi-Directional Pressure Compensator Valve

When working with energy efficient systems that utilize flexible flow paths a conventional compensator can be a restriction because it can only handle compensated flow in one flow direction. An idea that came up during the work with flexible system solutions was a bi-directional compensator component that allows compensated flow in both flow directions. The resulting patent PTC application is shown here. It has also been filed in Germany and USA. The application numbers in those countries are: de112007003560 and us12/900860.