An Electro-Hydraulically Controlled Cylinder on a Loader Crane

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Abstract

With tighter emission regulations for road vehicles pushing the technology forward, fuel savings are indirectly affecting the designs and technical solutions of loader cranes. By decentralizing the hydraulic power through driving each actuator separately, the goal of a more efficient crane drive is strived for. This thesis analyzes if the simple concept of a pump-controlled cylinder directly driven by an induction motor is achievable for a loader crane. Further, the crucial role of the induction motor is studied both mathematically and physically. A special research is also performed on energy efficiency and the capability of electric energy regeneration. By forming the transfer function of the system and performing measurements on a physical setup, the conclusion is drawn that the proposed pump-controlled cylinder concept is fully functional for its purpose which implies that the technology is promising. The report identifies a number of complications with this configuration, such as the induction motor demonstrating reduced performance at high loads and low speeds. Suggestions of improvements are presented with regards to these issues. The thesis also demonstrates high efficiency during a lifting motion and that the possibility of efficient electric energy regeneration is achievable if an optimum lowering speed is considered.
Acknowledgments

This project would not have been possible without all the help and support throughout the whole process, and we would therefore like to express our special gratitude to those involved. First of all, we would like to thank HIAB AB for giving us the confidence of investigating the future of loader cranes, but also for providing us with the majority of the required parts for this project. Also thank you Tube Control AB who assisted us in selecting the appropriate on/off-valve and provided us with the necessary information and related components. The work has been carried out at the division of fluid and mechatronic systems, and we would like to thank them for giving us free access to the crane, all equipment and the freedom to carry out our work in the laboratory.

We would like to thank Jörgen Rickan at Inmotion Technologies AB assisting us while configuring the inverter to meet our specific needs. Also, thank you Mikael Axin for helping us getting started with the lab crane, its related components and control systems. This project would not been the same without the inspiring Tuesdays-discussions with Alessandro Dell’amico. Thank you for encouraging us to focus on the goal. Finally we would like to address our supervisors Samuel Kärnell and Amy Ranikka for your appreciated support and patience, always willing to help investigate, discuss ideas and solve problems throughout this spring. A special thanks to you two.

Linköping, June 2018
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# Contents

1 Introduction .................................................. 1  
   1.1 Background ........................................... 1  
   1.2 Proposed Concept ..................................... 2  
   1.3 Problem Formulation ................................... 3  
      1.3.1 Research Questions ............................... 3  
      1.3.2 Limitations & Delimitations ..................... 3  
   1.4 Method .................................................. 4  
   1.5 Thesis Outline ......................................... 5  

2 Related Research ........................................... 7  
   2.1 Pump Controlled Actuators ............................. 7  
      2.1.1 The Four Quadrants ............................... 7  
      2.1.2 Concept Principles ............................... 9  
      2.1.3 Valve and Pump Control ......................... 11  
   2.2 Electric Regeneration of Potential Energy ........... 11  

3 Theory ...................................................... 13  
   3.1 Inverter and Induction Motor ......................... 13  
      3.1.1 Inverter ......................................... 14  
      3.1.2 Induction Motor ................................... 14  
   3.2 Rotating Displacement Machine ....................... 18  
   3.3 Rotating Shaft ......................................... 19  
   3.4 On/Off Valve ........................................... 19  
   3.5 Cylinder ............................................... 20  
      3.5.1 Cylinder Friction ................................ 21  
   3.6 Load Sensing Hydraulic System ....................... 22  

4 System Components and Test Rigs ......................... 25  
   4.1 Components ............................................ 25  
   4.2 Test Rig Setups and Descriptions .................... 27  
      4.2.1 Variable Load Setup .............................. 27  
      4.2.2 Valve Measurement Setup ......................... 28  
      4.2.3 Load Sensing Setup .............................. 29  
      4.2.4 Pump Controlled Cylinder Setup ................. 30  
   4.3 Component Validation .................................. 31
## Contents

4.3.1 On/Off Valve Characteristics ........................................ 32  
4.3.2 Induction Motor ....................................................... 32  
4.3.3 Cylinder Friction ..................................................... 36  

5 Linear analysis .............................................................. 39  
  5.1 System 1 ........................................................................ 40  
  5.2 System 2 ........................................................................ 42  
  5.3 Combined System .......................................................... 44  
  5.4 Valve-Free System .......................................................... 45  
  5.5 Comparison of Transfer Functions ....................................... 47  
  5.6 Stiffness Analysis ........................................................... 50  
  5.7 Load Dynamics ............................................................... 53  

6 System Performance on the Loader Crane ................................ 57  
  6.1 Crane Response ............................................................... 57  
     6.1.1 Load Sensing System ................................................. 57  
     6.1.2 Pump Controlled System .......................................... 58  
  6.2 Pressure Equalizing ......................................................... 61  
  6.3 High Load, Low Speed ..................................................... 62  

7 Energy and Efficiency Analysis ............................................ 65  
  7.1 System Boundaries ......................................................... 65  
  7.2 Energy Study ................................................................. 67  
     7.2.1 Lifting Cycle Energy Regeneration ............................... 67  
     7.2.2 Energy Regeneration Capability .................................. 69  
     7.2.3 Idling Losses .......................................................... 70  
  7.3 Efficiency Analysis ......................................................... 71  

8 Discussion ......................................................................... 75  
  8.1 Test Rig and System Components ....................................... 75  
  8.2 Transfer Functions .......................................................... 76  
  8.3 System Performance on the Loader Crane ......................... 77  
  8.4 Energy and Efficiency Analysis ......................................... 78  
  8.5 Possible System Improvements ......................................... 79  
     8.5.1 Alternative Setup ...................................................... 79  
     8.5.2 Component Sizing ..................................................... 79  
     8.5.3 Variable Displacement Machine ................................. 80  

9 Conclusions ..................................................................... 81  
  9.1 Answers to the Research Questions .................................... 81  
  9.2 Future Work ................................................................. 82  

Bibliography ...................................................................... 83
Nomenclature

\( a \)  Time constant  \([s]\)
\( \beta_e \)  Effective bulk modulus  \([Pa]\)
\( \Delta p \)  Pressure difference  \([Pa]\)
\( \delta_h \)  Hydraulic damping  
\( \eta \)  General efficiency  
\( \eta_{down} \)  Efficiency of lowering motion  
\( \eta_{eff} \)  Average effective work ratio  
\( \eta_{hm} \)  Hydromechanical efficiency  
\( \eta_{reg} \)  Regeneration efficiency  
\( \eta_{tot} \)  Total efficiency  
\( \eta_{up} \)  Efficiency of lifting motion  
\( \eta_{vol} \)  Volumetric efficiency  
\( \hat{V}_a \)  Stator line-to-neutral terminal voltage  \([V]\)
\( \lambda_a \)  Armature flux  \([Wb]\)
\( \lambda_D \)  Direct flux  \([Wb]\)
\( \lambda_Q \)  Quadrature flux  \([Wb]\)
\( \lambda_{DR_{ref}} \)  Reference direct rotor flux  \([Wb]\)
\( \lambda_{DR} \)  Direct rotor flux  \([Wb]\)
\( \mathcal{L} \)  Transition between time domain and Laplace domain  
\( \omega \)  General angular velocity  \([rad/s]\)
\( \omega_b \)  Break angular velocity  \([rad/s]\)
\( \omega_e \)  Electrical angular velocity  \([rad/s]\)
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$L_2$  Rotor leakage inductance  
$L_m$  Magnetizing inductance  
$L_R$  Rotor inductance  
$L_S$  Stator inductance  
$M$  General mass  
$m_p$  Piston mass  
$P$  Number of poles  
$p$  General pressure  
$p_1$  Valve upstream pressure  
$p_2$  Valve downstream pressure  
$p_p$  Piston side pressure  
$p_r$  Rod side pressure  
$P_{max}$  Maximum power  
$q$  General flow  
$q_m$  Hydraulic motor oil flow  
$q_p$  Hydraulic pump oil flow  
$q_{in}$  Entering flow  
$q_{out}$  Exiting flow  
$R$  Rotational friction  
$R_1$  Stator effective resistance  
$R_2$  Referred rotor resistance  
$R_a$  Stator armature resistance  
$R_{aR}$  Rotor armature resistance  
$S$  Stiffness transfer function  
$s$  Laplace variable  
$s$  Slip  
$T_m$  Motor torque  
$T_p$  Pump torque
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Chapter 1

Introduction

This chapter presents the context of this thesis and why it is of interest to research the presented subject. It defines which aspects are considered specifically within this area, and what answers this thesis is intended to find in this matter.

1.1 Background

The today’s actuating systems for loader cranes are commonly load sensing (LS) systems, with the pump connected to the vehicle’s internal combustion engine. Environmental regulations drives the technology development towards fuel savings and electrification of vehicles, and the loader cranes must adapt to the new market demands. In general, hydraulic systems are used in high energy consuming and power applications. Even though load sensing systems only provides power when required, it is still a high power consumer. To push the energy savings and the market technology forward, another type of concept is introduced, the electro-hydraulically controlled cylinder (EHCC). It consists of a cylinder directly driven by a hydraulic pump that in turn is driven by an electric motor which is electrically powered and controlled by an inverter. The benefits that this configuration possesses are fewer high energy loss components, such as main control valve (MCV), load holding valve and more which are not required. Since each actuator have an individual power supply the load interference issues are eliminated. The simultaneous driving losses can only be eliminated if every cylinder on the loader crane is exchanged to an EHCC, which exposes the drawback of space, weight and high component costs. As the vehicles electrifies, the EHCC has an advantage since it is already based on an electrified platform. However, the energy density in an electrical battery is lower than diesel, which demands even more efficiency both regarding propulsion as well as for the mobile hydraulic systems on electrified vehicles.

An electrical power supply enables the ability to regenerate energy back into the battery. Since many of the high loss hydraulic components are removed in this project, more of the input energy is assumed allocated by the lifting mechanism. The benefit is that some of this energy can possibly be recovered, making the
system even more efficient.

## 1.2 Proposed Concept

Before defining the main objectives of this thesis it is necessary to describe the proposed concept. The inner cylinder of a loader crane is powered with an *Electro-Hydraulically Controlled Cylinder*. The EHCC is a branch of the EHA (*Electro-Hydraulic Actuator*) commonly found on aircrafts, but can be distinguished with the difference that it is larger and does not have requirements with regards to compactness. The components of the EHCC are displayed in Figure 1.1. The EHCC consists of a pump connected between a hydraulic tank and a cylinder. Between the pump and cylinder an on/off valve is located to close the line to the pump. It is intended to act as a load holding valve in case of a blackout, and relieve the pump from pressure when the crane is not in motion. For safety reasons, a pressure relief valve is connecting the pressure side directly to the tank. The pump is powered by electric AC induction motor which in turn is controlled by an inverter. The inverter converts the DC current from a large electric battery into AC current.

![Figure 1.1: The provided EHCC and its components.](image)

There are single acting- and double acting systems that are directly powered by a hydraulic pump and electric motor. The single acting system specifically studied in this thesis is an open controlled system of the simplest possible form to minimize the amount of components and thereby losses, but also in order to maximize the useful energy of the hydraulic power. It further enables the possibility of implementing electric energy regeneration, as the hydraulic power always travels through the pump. The limitation is that it can only act in one direction, which is why the inner boom cylinder is suitable to apply the EHCC to. The loader crane is a HIAB loader crane provided by Linköping University. The power source (i.e. the EHCC) and its components are provided by HIAB.
1.3 Problem Formulation

The objective of this thesis is to physically replace the power source for the inner cylinder on a loader crane transforming it into an electro-hydraulically controlled cylinder (EHCC), and further evaluate the controllability of the system. In order to understand its behaviour and limitations it is also of interest to obtain a deeper insight into the mathematical representation of the induction motor, as well as developing a transfer function representation of the proposed system. Comparisons are made to a simple LS-system primarily with regards to performance.

1.3.1 Research Questions

From a technology development perspective there are many aspects to consider both regarding performance and energy consumption. Within the framework of this thesis the following questions are answered:

1. Is the system controllable given the proposed concept? How well does the system perform, and what type of issues are introduced?

2. How well does the mathematical representation of the induction motor correspond to physical measurements?

3. How could energy regeneration be introduced to the system, and what improvements in an energy consumption perspective can be attained?

1.3.2 Limitations & Delimitations

There are many aspects to take into consideration when a task such as this one is addressed. Unfortunately, it is not possible to investigate all topics at once, and some must be excluded for various reasons. Below is a list of what is not looked into. It is encouraged to view this list as an example of what can be done next.

- The components are provided as an existing setup which is not optimized in any way to this specific purpose. Dimensioning of the components lie outside the scope of the thesis.
- The thesis briefly mentions EHCC that manages operation in both extending and retracing direction. Only the single acting EHCC is studied in the report.
- Only the inner boom cylinder can be powered with a single acting EHCC, and the feasibility of the concept is evaluated on that cylinder only.
- The test cycles are only involving the inner boom cylinder while the rest of the crane is fixed. No other actuators are manipulated during the course of the project.
- Only the parameters of the provided EHCC are used for the transfer function. No alternative sizes are investigated for any of the components, except the valve.
• The system is only studied from the inverter to the cylinder. Even though
the transfer function is using a very simple approximation of the crane, little
effort is given into the crane itself.

• Only some of the components are validated. The non validated components
are either not of relevance such as the battery, or sufficient information is
already known for example efficiency maps of the pump.

• The EHCC is physically compared to a load sensing system with regards to
performance measurements only. The energy analysis comparison is excluded
which is further motivated in the report.

• Two load cases are studied. The first load case is an unloaded crane. The
second load case is an arbitrary and relatively light load. The system is
further not studied close to the maximum lifting capacity of the crane.

• This report often mentions energy regeneration. This refers to electric energy
regeneration only.

• When regenerating, the capability to accumulate this energy in the battery
is not considered. As a natural consequence, the efficiency of the battery
during charging and discharging is not studied at all.

1.4 Method

The project covers many different areas of work. The types of work and how they
are being performed are presented below.

• **Theory review**
  A lot of attention is given to the literature. Issues related to pump control is
an important key to understand the test rig. The components of the EHCC
specifically must be expressed in equations in order to form the mathematical
model of the system. It is crucial to broaden the knowledge of the induction
motor and inverter, how they are functioning together and how the induction
motor is mathematically represented to draw conclusions of its behaviour.

• **Modelling**
  Developing the transfer function of the system gives an insight to how it tends
to behave dynamically and how different parameters affects its behaviour.
The transfer function is obtained by using the mathematical representation
of each component, and creating a block diagram which have different ap-
pearance depending on what physical phenomenon is being considered.

• **Construction**
  The EHCC is physically connected to the crane with hydraulic hoses. Sen-
sors measuring pressure, length and other physical quantities are mounted
onto the rig and calibrated. Everything is connected together by a multi-
purpose real-time interface that reads all sensor values. This machine is
also the link between the computer and the inverter that communicates via the CAN protocol. A graphical user interface is designed in ControlDesk to control the motor via the inverter. The software itself is constructed using Matlab-Simulink and then compiled and exported to the real-time interface. The configuration of the inverter is made from the computer directly to the inverter via a Kvaser Interface also using the CAN protocol. The hardware must be in place in order to perform the measurements by controlling the EHCC and log the data from the sensors.

- **Performing measurements**
  The measurements capture the performance of the EHCC in connection with cylinder on the crane. This event uncovers technical qualities, such as response, strength and pressure equalizing across the on/off-valve. This is done to evaluate the drive-ability of the crane in practice when powered by the EHCC.

- **Interpreting the results**
  A major part of the thesis. The gathered data from the measurements are analysed and related to the theory. Different phenomenons that emerges when performing the measurements are analysed, and an energy study is performed. With the results to support the theory in this area, conclusions about this configuration are drawn.

1.5 Thesis Outline

The presentation of the thesis and the motivation why it is relevant to investigate this topic is presented in Section 1.1. Here the research questions are defined which the thesis is meant to find answers to. The related research further reviews the notions of an EHCC to understand more thoroughly what to expect from this configuration. Much of the research done in this project are gathered in the theory chapter, where all mathematical equations required to understand the research on the induction motor are presented and the development of the transfer functions is described. This information is given prior to the hardware chapter, as some of the components chosen to investigate further, such as the induction motor, are validated in this chapter and requires the knowledge of the mathematics behind it. As previously mentioned, the next chapter is presenting the hardware in this project. All components are explained with physical properties and the different test rig setups are presented, as the rest of the report refers to different setups when performing different tests. The motor, valve and cylinder are then validated, as these results are required for the transfer function, the energy analysis and more. The next chapter is the linear analysis that identifies the appropriate transfer function to describe this system, and examines the system characteristics. At this point all the required information prior to the measurement results have been reviewed and the physical tests are ready for presentation in the next chapter. The system performance is closely examined from several aspects. One of the largest performance aspects is the system efficiency. This topic deserves a separate chapter
as it is quite substantial. Approaching the end of the thesis, the discussion focuses both on the overall results, as well as the contents of specific chapters. Finally, the report is summarized by answering the research questions with support from its contents, and a quick mention of what further can be developed beyond this thesis.
Chapter 2

Related Research

This chapter reviews the literature with regards to system configurations similar to the EHCC. Previous studies of similar concepts introduces notions with regards to the definition, variants, possible challenges and energy regeneration potential for a system such like the EHCC.

2.1 Pump Controlled Actuators

Pump controlled hydraulic systems, where the actuator motion is directly controlled by the speed of the pump, are often divided into open and closed circuit systems [24]. The principle of each system can be illustrated as in Figure 2.1. The main difference is that in an open system one side of the cylinder is connected to the pump and the other one to tank. In the closed circuit both connections of the cylinder are connected to both ports of the pump.

The closed circuit system, also called throttle-less hydraulic actuator [10], visualized with a single rod cylinder, faces the issue of asymmetric flow which can cause controllability and efficiency problems [24]. Even though any pioneering solution have not yet been presented [10], it tend to be the main focus in current research [24]. However, both the closed and the open circuit system provides the possibility of energy recuperation without any throttle losses which according to studies can reach up to 35% [24, 10]. Furthermore, significant heat losses are the main consequences related to hydraulic throttling, which further reduces the need of cooling systems and large oil volumes.

2.1.1 The Four Quadrants

The interaction between the pump and cylinder can be divided into two scenarios. In the first scenario the flow is in the opposite direction from the force acting on the cylinder. In this case the rotational displacement machine (RDM) is acting as a pump opposing the force. In the second scenario the flow is in the same direction as the force acting on the cylinder piston rod. In this case the cylinder force drives the RDM acting as a motor. The lowering speed is controlled by counteracting
the torque from the RDM (braking) with the electric motor. This opens up the possibility for energy regeneration [7].

The actuators in the load path are the inner- and outer cylinder and the extension cylinders. The force acting on the innermost cylinder is always directed in the retracting direction, hence the pressure drop over the RDM is always positive. For the extension cylinder, and some special angles for the outer cylinder, the acting force on the cylinder can act in the extending direction, and the pressure drop over the RDM will act in the opposite direction. Four different cases can be derived from this reasoning:

- Extending as a pump (Actuating)
- Extending as a motor (Braking)
- Retracting as a pump (Actuating)
- Retracting as a motor (Braking)

This defines the working range for the RDM, and is further displayed in Figure 2.2. This is referred to as the four-quadrant actuation chart [8]. Note that the operating range of the inner cylinder is limited to positive pressure drop $\Delta p$, as the dead-weight of the crane is always directed in the retracting direction. This reasoning leads to two types of systems, one system capable of operation in all four quadrants, see Figure 2.3, and one simpler version for positive $\Delta p$ only, in Figure 2.4.

Figure 2.1: Principal illustration of an open and closed pump controlled system.
2.1 Pump Controlled Actuators

![Diagram of Pump Controlled Actuators](image)

Figure 2.2: The operating range for the rotating displacement machine expressed in four quadrants.

2.1.2 Concept Principles

The designs in Figure 2.3 and 2.4 are inspired by typical electro-hydraulic actuators, demonstrated in Figure 1-3 and 5-9 in [25] and similar versions in Figure 3.1 and 3.2 in [7] where also solutions to the asymmetric cylinder are presented. The way the more complex system is managing the positive and negative pressure drop is that it uses the shuttle valve in the middle to open and close the lines before and after the cylinder. In this way if the force acting on the cylinder is directed in the extending direction the rod side will become pressurized. The shuttle valve closes the connection to tank on rod side, and at the same time opens on piston side.

The valves $W_1$ and $W_2$ are present to replace the load-holding valves, but can also be used as meter in/out valves [11] to control the piston speed. This is safer as it creates lowering control redundancy. If the motor is unable to control a heavy load the valves can support the load partially or completely when lowering by establishing a pressure drop over the valve, but it induces more losses as well [11]. These are opened during operation but are configured to normally be closed. In case of a blackout, the loss of power to the valves should have them closing the connections automatically, eliminating the risk of an accidentally dropped load [7].

It is important to notice that the use of replenishing pressure might be necessary, more probably but not exclusively for the double acting system. If not, the risk for cavitation increase as the number of components increases. This can
Figure 2.3: The more complex EHCC. It can operate at positive and negative flow and pressure.

Figure 2.4: The simpler EHCC, operating in the positive $\Delta p$ quadrants only.
be achieved with a pressure relief valve between the low pressure side and tank, forcing the pressure to increase on the low pressure side before returning to tank. This solves the issue at the expense of increased losses [11]. [7] uses an extra feeding pump and accumulator on top of the PRV, which increases the complexity. Ultimately, these losses are unavoidable if the EHCC is to be double-acting.

2.1.3 Valve and Pump Control
Heybroek mentions in his thesis [7] a control strategy for when the operating point moves from one quadrant to another. Heybroek claims that the transition from positive to negative force is non-critical since the pressure $\Delta p$ is zero. It is also advised that all valves close at this point to prepare for pressure matching in the new quadrant. In his thesis the concept of pressure matching is discussed. To level the pressure before and after the valve, it can be opened slowly before the pump starts supplying flow. Another method is having the pump to equalize and control the pressure across the valve before it may open.

2.2 Electric Regeneration of Potential Energy
Studies made on a forklift where a bent-axis pump driven by a synchronous motor is used to provide the ability of potential energy regeneration, shows that up to 50% energy can be saved in a lifting and lowering cycle [19]. However, the system studied is driven by the electricity grid and not by a mobile energy accumulator such as a battery or a super-capacitor. This leaves the question whether a regular lead-acid battery is able to accumulate all the potential energy in a suitable way unanswered. Regeneration involved in mobile hydraulic systems where high potential energies are present, may result in high power recharging which is not always suitable for batteries [17].

Another forklift simulation study where a pouch-type lithium-titanate (LiTi) battery is used for energy storage, concludes that regeneration energy within the system is decreasing proportionally with increased lowering speed due to increasing internal losses [20]. It also concludes that higher load increase the regenerated amount of energy almost proportionally. However, the origin of the increasing losses that comes with faster lowering speeds are not determined whether they are mechanical or electrical losses. Furthermore, the study specifically motivates the choice of chemical compounds within the battery with respect to maximizing the recharging capability, and also discusses the ability of downsizing batteries thanks to the electrical regeneration feature [20].
Chapter 3

Theory

This chapter explains the theoretical background needed to understand the different elements of this thesis. The mathematical representation of the induction motor and the components related to the transfer function are presented here. Lastly, a short description of what is considered a load sensing system is defined.

3.1 Inverter and Induction Motor

AC machines are historically most common in stationary working conditions, such as fans and pumps. They also have limited means of adjusting speed during operation. Today however, variable speed and load applications are possible environments for AC machines thanks to modern variable speed drive and inverter technologies [28]. This chapter will introduce field-oriented control of an induction motor, which is a common approach of controlling AC machines in order to obtain properties similar to a DC motor.

There are mainly two types of AC machines, synchronous and asynchronous motors. Both are powered with a 3 phase AC source to create a revolving magnetic field in the stator. The difference lies in the rotor which is essentially a large cylindrical magnet. Synchronous motors use an additional DC source to magnetize its rotor, while the magnetic field is induced in the rotor of the asynchronous motor. The asynchronous rotor is named squirrel cage due to its appearance, and the motor is often called induction motor due to the fact that the magnetic field in the rotor is induced by the stator field. Synchronous and asynchronous motors are often mentioned side by side due to their similarities in mathematical representations [28].

In order to induce current in the rotor of the asynchronous motor the electric current in the stator must be alternating, and its frequency must be varying as well in order to operate at different speeds. The torque that is produced is generated by a relative difference in stator and rotor magnetic field frequencies, also called slip. Since no torque is produced if the stator and rotor speed are the same, the rotor will always differ from the synchronous speed in order to generate torque in
any direction. The synchronous motor on the other hand runs synchronously with the input frequency.

There are multiple advantages and disadvantages with the two configurations. Synchronous motors have more accurate speed control since the rotor always rotates with the synchronous frequency, which the asynchronous motor is not [28]. The efficiency is somewhat lower for the asynchronous motor, but is less expensive in turn since it does not require an additional DC source. The decisive criterion is pricing as well as speed controllability. In this system an asynchronous motor is used, since the flow is intended to be controlled with the motor velocity and not the displacement of the pump, which increases the demands on the motor.

3.1.1 Inverter

Both motor types requires a variable frequency drive (VFD) commonly named inverter in order to efficiently operate at different speeds. It converts DC power from the battery into three phase AC current with varying frequency, voltage and current. It achieves this by using large transistors and clever switching tables together with large capacitors. There are a couple of different methods of controlling the motor with the inverter, and it requires some knowledge of the state at which the motor is in. The method used in this project is called field oriented control (FOC) and can be divided into two groups: direct and indirect FOC. During direct FOC, hall sensors are used to measure the state of the magnetic flux. This is accurate but the method is complicated and sensors are expensive [13]. Instead, a widely spread method is to measure the rotational velocity and the stator AC frequency to estimate the rotor flux, which also refers to indirect FOC. A detailed explanation of the methods and their differences can be found in [13]. The indirect field oriented control (IFOC) is used by the inverter in this project. There are no dynamics in the inverter to consider for this project. The requested AC current is instant, and the switch losses and heat dissipation can be approximated to a constant loss [19].

3.1.2 Induction Motor

The electrical scheme of an induction motor is often simplified to what is called the equivalent circuit, see Figure 3.1. It is a single-phase approximation of a three-phase motor and identifies the physical quantities according to Equation (3.1) used in the mathematical equations. The core resistance is often neglected in these representations [28]. Furthermore, it is convenient to express the currents flowing through this circuit with a different coordinate system called direct- and quadrature-axis variables. These are real and imaginary components of a coordinate system aligned with the field-winding axis. This results in that the direct and quadrature quantities (indexed d and q respectively) experience constant magnetic paths. The transformation between three phase and dq coordinates is a comprehensive process and requires the knowledge of the rotor position, which is done automatically by the inverter. A complete derivation is described in appendix 3 in [28].
3.1 Inverter and Induction Motor

Figure 3.1: An equivalent single phase circuit of a multiphase induction motor.

\[ L_S = L_m + L_1 \]
\[ L_R = L_m + L_2 \]
\[ R_a = R_1 \]
\[ R_{aR} = R_2 \]  \hspace{1cm} (3.1)

The electromechanical torque (omitting friction- and other losses) can be related to the currents in the motor using Equation (3.2). What is of interest is that the torque is directly proportional to the quadrature component of the current, \( i_Q \) and the direct rotor flux, \( \lambda_{DR} \).

\[ T_{mech} = \frac{3}{2} \left( \frac{P}{2} \right) \left( \frac{L_m}{L_R} \right) \lambda_{DR} i_Q \]  \hspace{1cm} (3.2)

This equation is used by the controller to calculate the reference currents in dq coordinates. Given a specific torque request \( T_{ref} \) from the operator through a separate speed controller, the torque controller calculates the reference quadrature current \( i_Q \) in Equation (3.3). The direct rotor flux reference \( \lambda_{DR_{ref}} \) is determined by the flux weakening condition described later.

\[ i_{Q_{ref}} = \frac{4}{3P} \frac{L_R}{L_m} \frac{T_{ref}}{\lambda_{DR_{ref}}} \]  \hspace{1cm} (3.3)

The reference direct current component is also calculated from the reference rotor flux, see Equation (3.4). The relation between the two is not always linear, as the flux weakening principle will affect the rotor flux and consequently the current. This is explained later.

\[ i_{D_{ref}} = \frac{\lambda_{DR_{ref}}}{L_m} \]  \hspace{1cm} (3.4)

The reference currents are then fed through two independent controllers that ensures that the inverter outputs the requested torque. Complete overviews of the information flow are illustrated in [3] (Figure 2.1), [13] (Figure 14.7-1) and [28] (Figure 10.25). A great overview and detailed description is also presented in
The block diagrams differ somewhat between the figures, but the principle is throughout the same.

There are mainly three quantities that are of interest when running an induction motor, and these are the armature current $I_a$, armature voltage $V_a$ and the armature flux $\lambda_a$. The armature is essentially the stator. Equations (3.5) through (3.7) describes the relation between these quantities and the direct and quadrature currents.

$$I_a = \sqrt{\frac{i_D^2 + i_Q^2}{2}}$$  \hspace{1cm} (3.5)

$$\lambda_a = \sqrt{\frac{\lambda_D^2 + \lambda_Q^2}{2}} = \sqrt{\frac{(L_S i_D)^2 + \left(L_S - \frac{i_D}{I_R}\right)^2 i_Q^2}{2}}$$  \hspace{1cm} (3.6)

$$V_a = \sqrt{\frac{(R_a i_D - \omega_e \left(L_S - \frac{i_D}{I_R}\right) i_Q)^2 + (R_a i_Q + \omega_e L_S i_D)^2}{2}}$$  \hspace{1cm} (3.7)

where $\omega_e$ is the electrical angular velocity and is calculated in Equation (3.8).

$$\omega_e = \frac{P}{2} \omega_m + \frac{R_a R_l i_Q}{L_R i_D}$$  \hspace{1cm} (3.8)

It is important that these quantities are kept within certain limits, otherwise the motor can take damage. These conditions can be formulated as in Equation (3.9) and (3.10).

$$\begin{cases} I_a < I_{a_{\text{max}}} \\ V_a < V_{a_{\text{max}}} \\ \lambda_a < \lambda_{a_{\text{max}}} \end{cases}$$  \hspace{1cm} (3.9)

$$V_{a_{\text{max}}} = \frac{V_{\text{max}}}{\sqrt{3}}$$

$$\omega_{e0} = 2\pi f_{\text{rated}}$$

$$\lambda_{a_{\text{max}}} = \frac{\sqrt{2} V_{a_{\text{max}}}}{\omega_{e0}}$$  \hspace{1cm} (3.10)

$$I_{a_{\text{max}}} = \frac{P_{\text{max}}}{\sqrt{3} V_{\text{max}}}$$

Where $V_{\text{max}}$, $f_{\text{rated}}$ and $P_{\text{max}}$ are predefined quantities from the manufacturer. The currents in the stator are directly connected to heat in the armature. The flux must be lower than its limits to avoid saturation in the motor, and too large voltage could damage the insulation [28]. These quantities are monitored continuously by the inverter during operation. If the currents are too high the reference torque is reduced, and if the flux or voltage is too high the reference flux $\lambda_{DR_{\text{ref}}}$ is weakened before the reference signals are sent to the controllers.
3.1 Inverter and Induction Motor

Flux weakening

As the rotational speed increases the armature voltage does too, and at some point it reaches the maximum allowed value. What can be done then is to reduce the direct rotor flux $\lambda_{DR}$ that in turn reduces $i_D$ according to Equation (3.7). This enables the rotor to spin faster without violating the constraints. This comes at the expense of reduced maximum torque, since the rotor flux affects the torque equation as well. Bare in mind that the quadrature current $i_Q$ has to increase in order to compensate for the reduced flux, see Equation (3.2). Eventually the increased velocity forces $i_Q$ to increase to maintain the torque until it reaches a point where maximum current (and therefore torque) is achieved for that speed. A further increase of speed will cause loss of torque and the motor will not be able to accelerate. There are multiple different methods of achieving an efficient field weakening that maximizes as much as possible for all constraints. One method presented in [21] and [22] is a graphical representation of the constraints where the limiting factors are current (circle), voltage (ellipse) and power ($\frac{1}{\omega}$). As the velocity increases, the ellipse shrinks. This method is sure to give an optimum operating point but is quite advanced. Another method is seen in [3] where the weakening is a function of the voltages only, but quite sophisticated. The simplest and very commonly used method [21] is to simply reduce the field inversely proportional to the rotational speed, which is demonstrated in [28], [21] and [27]. Repeating the field weakening equation from [27] in Equation (3.11) where $\omega_b$ is the break frequency and comparing to Equation 10.10 in [28], it is apparent that the equations are the same in the flux weakening region, given that $V_a = V_{a_{max}}$.

$$\lambda_{DR_{ref}} = L_m i_Q \quad \omega_m < \omega_b$$
$$\lambda_{DR_{ref}} = L_m i_Q \frac{\omega_b}{\omega_m} \quad \omega_m > \omega_b$$

(3.11)

This flux weakening method is well presented at page 162 in [27]. An illustration to give a graphical understanding of what the flux weakening region is, is displayed in Figure 3.2. With increasing frequency the voltage also increases, and at a certain point the flux must be weakened in order to further increase the speed. The break frequency is selected as high as possible without any of the constraints being violated.
3.2 Rotating Displacement Machine

The rotating displacement machine (RDM) can act both as a pump and motor, and the current state of utility depends on how the RDM is used. The RDM is of fixed displacement type where the pressure and flow are directly dependent on the input rotational power, in contrast to variable displacement machines where the flow can be varied with the displacement $D$ even during constant speed. The domain conversion Equations (3.13) to (3.15) are found in [1].

\[
q_p = \frac{D}{2\pi} \omega \eta_{vol} \tag{3.12}
\]
\[
T_p = \frac{D}{2\pi} \Delta p \frac{1}{\eta_{hm}} \tag{3.13}
\]
\[
q_m = \frac{D}{2\pi} \omega \frac{1}{\eta_{vol}} \tag{3.14}
\]
\[
T_m = \frac{D}{2\pi} \Delta p \eta_{hm} \tag{3.15}
\]

Studying Equations (3.12) to (3.15) it is evident that the conversion between torque and pressure, and rotational speed and flow, is not only linear but almost identical for both types of machines. The difference lies in the conversion direction between the different quantities. For the pump the efficiencies $\eta$ are multiplied at the input, i.e. the torque and rotational speed respectively, and analogously at the hydraulic power for the motor equations. With reference to the four quadrants in Figure 2.2, the difference in the Equations (3.12) to (3.15) is that the volumetric and hydro-mechanical efficiency maps are multiplied or divided depending on the actual functionality of the RDM which is clarified in Figure 3.3.

Figure 3.2: An illustration between how the flux and voltage behaves as the rotational speed increases.
3.3 Rotating Shaft

The pump is mounted directly onto the motor, and there is no extra shaft between them. It is however necessary to cover the differential equation that describes the dynamics of a shaft and its angular velocity, as well as the rotational moment of inertia for the pump and motor. The differential equation for the shaft can be expressed as

\[ T_m - T_p = J \dot{\omega} + R \omega \]  \hspace{1cm} (3.16)

Rearrange and transform into Laplace domain

\[ \omega = \frac{T_m - T_p}{Js + R} \]  \hspace{1cm} (3.17)

Where \( J \) is the total rotational moment of inertia, and \( R \) is the total friction.

3.4 On/Off Valve

A valve is generally approximated as an orifice, where the flow is determined by the pressure drop over the valve and the valve opening. This project uses an on/off valve and the flow is not controlled, only enabled or disabled. The general function of an orifice has the structure displayed in Equation (3.18), and can also be found in [1].

\[ q = C_q x \omega \sqrt{\frac{2}{\rho} \Delta p} \]  \hspace{1cm} (3.18)

For the transfer function it is required to mathematically express the valve linearly. This can be done by applying Taylor series expansion to the equation at a typical
operating point. This will create a linear expression of the flow, with respect to pressure and valve displacement, see Equation (3.19).

\[
\Delta q = K_q \Delta x_v - K_c \Delta p
\]

\[
K_q = \frac{\partial q}{\partial x_v}
\]

\[
-K_c = \frac{\partial q}{\partial p}
\]

(3.19)

Since this is an on/off valve, and the displacement \(x_v\) is not controlled, \(K_q = 0\). The flow through the valve can be expressed as a flow-pressure gradient, Equation (3.20) and can be obtained from measurements.

\[
K_c(p_1 - p_2)
\]

(3.20)

The coefficient is simply the slope of a flow-pressure graph. Higher \(K_c\) means a steeper slope and a larger flow for a specific pressure drop.

### 3.5 Cylinder

The cylinder is governed by two equations, one for each domain. The hydraulic domain is derived from the continuity equation and accounts for the pressure build-up inside the cylinder chamber. It is presented in its general form in [1] and is repeated in Equation (3.21) where \(q_{in}\) is flow into the volume and \(q_{out}\) is out.

\[
\sum q = q_{in} - q_{out} = \frac{dV}{dt} + \frac{V}{\beta_e} \frac{dp}{dt}
\]

(3.21)

Rewriting the volume expressed in the cylinder stroke gives an expression as Equation (3.22). Note the added term \(V_0\) that represents the volume in direct connection with the cylinder chamber.

\[
q_{in} - q_{out} = A \dot{x}_p + \frac{Ax_p + V_0}{\beta_e} \frac{dp}{dt} \Leftrightarrow
\]

\[
\frac{dp}{dt} = (q_{in} - q_{out} - A \dot{x}_p) \frac{\beta_e}{Ax_p + V_0} \Leftrightarrow / \mathcal{L} / \Leftrightarrow
\]

\[
p = (q_{in} - q_{out} - Ax_p s) \frac{\beta_e}{s(Ax_p + V_0)}
\]

(3.22)

This equation applies only to the piston side, as the volume increases with increased stroke. For the rod side, the volume is decreasing, and Equation (3.22) must be modified accordingly, see Equation (3.23).

\[
p = (q_{in} - q_{out} + Ax_p s) \frac{\beta_e}{s(Ax_{p_{max}} - x_p) + V_0}
\]

(3.23)
The flow terms \( q_{\text{in}} \) and \( q_{\text{out}} \) are specific for each chamber, as well as \( V_0 \). For volumes that are not directly connected to a changing volume, the terms depending on cylinder position \( x_p \) can simply be omitted which is stated in Equation (3.24).

\[
p = (q_{\text{in}} - q_{\text{out}}) \frac{\beta}{sV_0}
\]  

(3.24)

The pressure inside the cylinder chamber is converted to a linear force by the piston area. This force will counteract the load force on the cylinder piston rod, as well as dynamical and resistive forces to produce a movement. The mechanical force and motion can be derived by using the free body diagram in Figure 3.4 and applying Newton’s second law. Further solving for the resulting force acting on the cylinder piston rod, and transforming it into Laplace domain gives the results in Equation (3.25).

\[
A_p p_p - p_r A_r - m_p \ddot{x}_p - B_p \dot{x}_p - F_L = 0 \Leftrightarrow \mathcal{L} / \Leftrightarrow \\
F_L = p_p A_p - p_r A_r - m_p s^2 x_p - B_p s x_p
\]  

(3.25)

The pressure \( p_r \) is assumed to be 0 for the inner cylinder, as it is directly connected to tank for the simpler system used in this project. The cylinder leakage can be neglected [4].

### 3.5.1 Cylinder Friction

The cylinder is subject to friction that behaves in accordance with the **Strebeck Friction Curve**. A theory initially proposed by Richard Strebeck in the early 20th century while studying bearings. In [23] the phenomenon is studied with respect to hydraulic cylinders and the Strebeck curve is apparent in this field as well. What is significant with the Strebeck curve is the characteristic shape of the curve, which is a sum of separate friction forces generated by different physical causes.

The Strebeck curve is composed by three different types of frictions. An overview of them can be observed in Figure 3.5. The Coulomb friction, \( F_c \) in
Figure 3.5 a), is a constant force opposing the direction of motion, regardless of the relative velocity. This friction model alone is a common but rough approximation of the total friction force. Further, the viscous friction $F_v$ in Figure 3.5 b), is proportional to the velocity. As the relative velocity between the surfaces increases, so does the friction. The so called Strubeck effect occurs at low velocities, and is the transition region from static to dynamic. In this region there is a small sliding relative movement, but a lubricating oil film is not fully developed. The effect is illustrated in Figure 3.5 c), where $F_{stiction}$ is the static friction force. Paper [18] mentions the same friction components, and also discusses other dynamic behaviors such as friction lag and hysteresis. These phenomenons are not considered in this study.

Figure 3.5: The different friction force contributions, with the sum of forces a), b) and c) in the lower right corner d), which is the characteristic Strubeck friction curve.

3.6 Load Sensing Hydraulic System

This section defines what is considered a load sensing system with respect to this thesis. A load sensing system is a hydraulic system that senses the magnitude of the load, and increases the system pressure with a predefined pressure difference $\Delta p$ on top of the load pressure. By doing so the system does not output unnecessary large power. This can be achieved with a variable displacement pump or, as used in this project, a fixed displacement pump and a shunt valve. It is located between pressure side and tank and its pressure setting is connected to the system load. As the load pressure increases, the shunt valve adjusts the pressure side to the load pressure plus $\Delta p$, and the desired behaviour is obtained. The shunt-LS is less expensive since it uses a fixed pump, but its lowest pressure is minimum $\Delta p$.
which means that it has idling losses. The variable pump LS-system does not have this issue. Common for both systems are the drawbacks of losses related to driving multiple functions at the same time, named simultaneous driving losses. The pressure side is set by the largest load, and if two functions are driven at the same time with different load size the difference in pressure generates losses. Another loss contributor is the main control valve which controls the speed of the functions, which induces major losses as it throttles each function. Add the pressure compensating valves which are common on the MCV, and even more energy is dissipated as heat. Pressure compensating valves ensures a constant pressure drop over the MCV which results in load speed independence. The final mentionable and quite obvious component is the load holding valve which is a loss contributor. It opens when the cylinder is manipulated, but it also dissipates energy since pressure is needed to keep the valve opened. This component can be replaced with a solution used in this project, an electrically controlled on/off valve which however is rarely used.

![Diagram](image)

Figure 3.6: The illustration demonstrates graphically how losses occur when multiple actuators are used simultaneously.

In Figure 3.6 both the inner and outer cylinder are used at the same time. In this example, the inner cylinder requires high pressure and low flow, while the outer cylinder requires the opposite. The required hydraulic power must accommodate both the sum of flow and pressure, and the operating point moves to the upper right corner of the graph. This causes much of the power turning into waste (in orange) over the MCV. Even though the LS system is considered to be relatively energy efficient, more can be done in this matter. The information in this chapter are based on reference [4].
Chapter 4

System Components and Test Rigs

This chapter explains the different setups for the project and is referred to from other chapters. The rig setups are configured for multiple purposes. Some of the rig setups are used for validation of the components that are examined more closely in this project. Other rig setups are used to examine the actual performance of the EHCC. Lastly, the induction motor, valve and cylinder are validated using different rigs.

4.1 Components

To further understand the test rig and its different setups, the most central components used in tests and evaluations are presented in the list below.

Battery

The battery used as energy storage in the physical setup is a lead-acid battery with a rated capacity of 40 kWh and a voltage of approximately 94 V.

Inverter

An inverter with maximum rated voltage and current of 80 V and 550 A is used to control and supply the electrical motor with alternative three phase currents. The inverter is fed with DC voltage and current directly from the battery, and is controlled and configured through Controller Area Network (CAN-bus).

Electrical Motor

The electric motor is a three phase induction motor with rated data shown in Table 4.1, and can also be seen to the right in Figure 4.1. The motor is supplied with a speed sensor which signal can be obtained from the inverter.
Table 4.1: Induction motor rated data.

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Power</td>
<td>47</td>
<td>[kW]</td>
</tr>
<tr>
<td>Voltage</td>
<td>80</td>
<td>[V]</td>
</tr>
<tr>
<td>Current</td>
<td>418</td>
<td>[A]</td>
</tr>
<tr>
<td>Speed</td>
<td>2339</td>
<td>[rpm]</td>
</tr>
<tr>
<td>Frequency</td>
<td>80</td>
<td>[Hz]</td>
</tr>
</tbody>
</table>

**Hydraulic Pump**

Mounted directly on the electrical motor illustrated in Figure 4.1 is a fixed displacement pump with a displacement of $34.2 \text{ cm}^3/\text{rev}$.

![Fixed displacement pump and electrical motor](image)

Figure 4.1: The fixed displacement pump and the electrical motor connected together. Down in the left corner is a voltage converter not used in this project.

**Control Unit**

The unit used to control the test rig is a dSpace Autobox equipped with boards capable of sending and receiving analogue signals, but also CAN-bus information. This allows the software control system to be created in a *Matlab-Simulink* environment together with a specific dSpace-library. The control system is further compiled within *Matlab* to a file readable by the *Autobox*.

**On/Off Valve**

The valve is a low restriction on/off-valve with a check valve in one direction. When closed, the flow can only go in one direction. When open, flow is possible in both directions. It is therefore arranged in the extension direction on the test rig. The valve must be opened when retracting the piston, similarly to a load holding valve. The load holding valve is supposed to prevent the load from accidentally falling by cutting the flow out of the cylinder during a power blackout. The maximum
flow of the valve is three times higher than the capacity of the pump, to further promote a low pressure drop.

**Loader Crane**

All tests are performed on a HIAB crane model 070AW with a rated supply flow of 40 l/min and a supply pressure of maximum 225 Bar. Since only the inner boom is manipulated during this project the other actuators are used to fixate the rest of the crane. To represent an easy and typical load case the outer boom is aligned with the inner boom, and the extension is extended half way. This gives a working radius of approximately 5 m. The maximum hook load at that predefined distance is 1800 kg.

### 4.2 Test Rig Setups and Descriptions

Depending on measurement purpose, the test rig is modified in different ways. Common between all setups are the components from the battery to the pump. Each setup and individual purpose is further described below.

#### 4.2.1 Variable Load Setup

In order to control the hydraulic load acting on the pump, a test setup with a variable orifice is introduced according to Figure 4.2. By continuously reading the actual pressure before the orifice (sensor $p_1$) an accurate load on the system can be chosen. This allows the electrical motor and the pump to be examined during loading conditions without the need or risk of moving a loaded physical crane in an early control system development stage.

![Flow chart of the test rig setup with a variable orifice](image)

*Figure 4.2: Flow chart of the test rig setup with a variable orifice. The figure is meant to illustrate the way the system is connected physically, and the flow through each component. The placement of flow and pressure sensors $q$ and $p$ is also illustrated.*
4.2.2 Valve Measurement Setup

To exactly determine the actual losses through the on/off valve meant to act as load holding valve in the pump controlled cylinder system, a measuring setup is introduced similar to the variable load setup. The valve measurement setup is illustrated in Figure 4.4, where the valve is placed in the check valve opening direction which allows the valve losses to be measured both during closed and opened.

Figure 4.4: Flow chart of the valve measurement setup.
4.2 Test Rig Setups and Descriptions

4.2.3 Load Sensing Setup

The load sensing setup acts as a comparison reference to the pump controlled concept which is to be evaluated from different point of views. Based on the same battery-powered motor and fixed displacement pump, the load sensing (LS) control valve shown in Figure 4.6 is supplied with a constant flow of oil from the pump which rotates with a fixed speed. The LS system is equipped with a shunt valve which maintains a constant pressure drop over the valve independent of the actual load size. It also enables the flow back to tank when all valves are closed. The pressure drop across the shunt valve of approximately 30 bar is maintained even when no functions are manipulated. The load sensing setup is not equipped with a conventional load holding valve, which function is to prevent unexpected lowering in case of a hose rupture. Usually this kind of valve adds orifice losses to the system which is not the case in this setup. Finally, the cylinder is mounted on a loader crane.

Figure 4.5: The on/off-valve mounted in its valve block with incoming flow in the bottom right, and outgoing flow to the left. The pressures $p_1$ and $p_2$ are measured with the sensors directly before and after the valve.
The LS setup consists of two separate control systems. The first system is illustrated in Figure 4.6. The second part is the valve control system, which is controlled through the hardware communication system IQAN developed by Parker. Each function on the LS valve is electronically controlled through a joystick connected to a separate control unit. However, only the boom function is connected and used in the tests.

Figure 4.7: The load sensing valve of model Parker L90LS used to control the reference system.

4.2.4 Pump Controlled Cylinder Setup

The final setup is the introduced pump controlled concept to be tested and evaluated as main purpose of this project. The physical setup is illustrated in Figure 4.8 where the cylinder is mounted on a loader crane as in the load sensing setup. The main difference is the absence of a control valve, but instead the on/off-valve is introduced in the cylinder extending direction as a load holding valve for safety reasons. The piston rod side of the cylinder is directly connected to tank which
makes this system a single acting system unable to handle extending forces according to Figure 4.8.

Figure 4.8: Illustration of the pump controlled cylinder setup.

Since the pump is directly connected to the electrical motor, a hydraulic pressure will be generated directly as the motor starts to move. The inverter which powers the induction motor is controlled from the Autobox which gives a speed reference value to the inverter. In turn, the inverter has its own speed controller which calculates and delivers a suitable current and frequency to the induction motor to maintain the requested motor speed.

4.3 Component Validation

Some of the components have been given extra attention during the course of this project. It is of interest to further investigate the characteristics of these components, which is explained in this section. For example the electric motor is mathematically complex, and its behavior is non-trivial. The valve has theoretically a squared relation between pressure and flow, but in practice it turns out to be very linear. Another rather unknown component is the cylinder which is quite old.
4.3.1 On/Off Valve Characteristics

The characteristics of the valve is captured by ramping the flow over the systems range. The physical setup is explained in Section 4.2.2. The pressure is measured before and after the valve in order to read the pressure drop. Special attention is being paid to lower velocities of the valve capacity, as the maximum pump flow in both directions is 100 l/min which is one third of the rated valve maximum flow.

![On/Off-Valve Characteristics](image)

Figure 4.9: Flow-pressure coefficient for the on/off valve in the check valve direction, open and closed.

By studying Figure 4.9 it is apparent the flow-pressure characteristics are linear. The slope represents the flow gain with respect to the pressure drop over the valve, $K_c$ can be directly extracted from the graph and used in Equation (3.20), rather than derived from (3.19). The gradient is $5.88 \cdot 10^{-9} m^3/Ns$ when the valve is open, and $5.08 \cdot 10^{-9} m^3/Ns$ when the flow is running through the check valve. The value of $K_c$ is significant for the transfer function.

4.3.2 Induction Motor

With the possibility to measure and log most of the parameters from the inverter, including $i_D$ and $i_Q$, the equations describing the induction motor can be verified with relatively high accuracy. First of all, the break frequency $\omega_b$ that defines the initiation of the field weakening region has to be found. This is done by ramping the speed from 0 to 2400 rpm and studying the currents, using the rig setup in Section 4.2.1. The adjustable orifice is set to generate a pump pressure of 100 bar at full speed.
Figure 4.10: A ramping of the electric motor while the angular speed and currents are logged. The currents differ from each others at higher speed, as the motor enters the field weakening region.

Studying Figure 4.10 the direct and quadrature currents follow each others until a certain speed is reached. At that point the direct current is relatively constant, while the quadrature current increase. What happens is that the direct current is limited in order not to violate the voltage and flux constraints, see Section 3.1. Since the increased speed means increased flow through the orifice the pressure, and therefore the torque on the shaft, increases. $i_Q$ is compensating the reduced flux $\lambda_{DR}$ by increasing even further. It can be noted that the relation between armature current $I_a$ and pump speed is unchanged, even though the $i_D$ and $i_Q$ components change drastically.

In practice the break frequency is dependent on a number of factors, and the results will later give a hint of the inverter using a more sophisticated flux weakening algorithm than suggested in this project. Since the angular velocity is a major factor, a fixed frequency will turn out to be a good enough approximation. This breakpoint between the direct and quadrature current components in Figure 4.10 is a strong suggestion that the break frequency is at approximately 1900–2000 rpm.

In steady state, i.e. the shaft is rotating at a constant rate, the pump and motor torque are equal in magnitude. Using the pressure to calculate the pump torque, and the electric currents to calculate the motor torque, the motor equations can be verified. Four measurements are made at operating points where the efficiency maps and parameters of both the pump and motor are known, to eliminate this factor of uncertainty. These are provided from the manufacturer of each component respectively. The pressure is logged in order to calculate the pump torque according to Equation (3.13), the direct and quadrature currents for the motor torque according to Equation (3.3) and (3.4), and the angular velocity for the field weakening algorithm according to Equation (3.11). The operating points are presented in Table 4.2 together with the results from the relevant equations.
Table 4.2: Operating points for validation purposes. \( \omega_b = 1900 \) \( \text{rpm} \).

<table>
<thead>
<tr>
<th>Pressure [bar]</th>
<th>Speed [rpm]</th>
<th>( T_p ) [Nm]</th>
<th>( T_m ) [Nm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>1000</td>
<td>31.3</td>
<td>28.7</td>
</tr>
<tr>
<td>100</td>
<td>1000</td>
<td>58.5</td>
<td>65.1</td>
</tr>
<tr>
<td>50</td>
<td>2000</td>
<td>33.2</td>
<td>30.8</td>
</tr>
<tr>
<td>100</td>
<td>2000</td>
<td>60.5</td>
<td>63.6</td>
</tr>
</tbody>
</table>

The results are that the equations coincide moderately well. A larger deviance can be noticed at higher pressure and lower rpm. The remaining points differs approximately 3 \( Nm \) which is acceptable.

A measurement is made with manual control of the speed reference, by turning a slider on the display in the Autobox GUI, in order to demonstrate the interaction between pump and motor dynamically in a broader operating range. The break speed is set to \( \omega_b = 1900 \) \( \text{rpm} \).

In Figure 4.11 the motor is attempting to follow the edgy speed reference. At two occasions the actual speed of the motor is faster than the break speed and is therefore entering the flux weakening region. The effect can be seen in the lower graph which is derived from Equation (3.11). It is further noticeable that the flux is considerably lower when the speed increases above the break frequency at the end, but only a minor impact can be observed around 15 s. The flux is also proportional to the torque according to Equation (3.2).
4.3 Component Validation

Figure 4.12: The upper graph displays the armature voltage both with and without flux weakening implemented, and the lower graph shows the pump torque together with the motor torque, with and without flux weakening implemented.

In the upper graph of Figure 4.12 the effects of the simple flux weakening algorithm is demonstrated. Without this effect the voltage is exceeding the limit as seen in blue, but it does not despite the fact that the speed is increasing. In the lower graph the pump torque induced by the pressure is plotted against the motor torque, which is derived from logging $i_D$ and $i_Q$ currents with and without implementation of flux weakening. Naturally, the motor torque is larger than the pump torque when accelerating, and the opposite when decelerating. A negative torque is even noticeable at the end of the run, as the motor is trying to stop rotating completely. The pump torque is also lagging behind due to the fact that pressure needs to build up in the hoses between the pump and orifice.

At low speeds in the beginning the motor and pump coincide very well. At the first field weakening area around 12-17 s the motor torque is somewhat larger since it is still accelerating. A minor flux weakening can also be observed. Between 23-30 s a peak in motor torque can be seen. This is because the speed reference is taking minor steps, see the previous Figure 4.11, and the speed controller within the inverter tries to follow the reference speed. In the second field weakening area towards the end (32-27 s) there is a major difference between flux weakened and non-flux weakened motor torque. $T_{mFW}$ is closer to the pump torque than the one without flux weakening, $T_m$, which is unreasonably high. It is clear that the motor torque does not reflect the reality without flux weakening. However, small uncertainties regarding the actual break speed impacts on the resulting torque. Overall the electric motor equations reflects the measured values with some uncertainties regarding the impact of the flux weakening region, but it turns out that this simple approach fairly accurately describes the behaviour of the motor.
4.3.3 Cylinder Friction

It is of interest to examine the cylinder friction to determine the amount of system losses that can be derived from it. By looking at Equation (3.25), repeated in (4.1), and Figure 3.4, there are five force terms in the free body diagram of the cylinder, whereas the cylinder friction \( B_p \dot{x}_p \) is sought.

\[
A_r p_r - p_p A_p - m_p \ddot{x}_p - B_p \dot{x}_p = F_L
\]  

(4.1)

The pressures at each side of the cylinder, the cylinder areas as well as the cylinder position are known. The cylinder velocity can be derived by differentiating the cylinder position. What is left are further the dead-weight of the crane \( F_L \) and the acceleration term \( m_p \ddot{x}_p \). The dead-weight can be known by orienting the crane in such a way that it has minimal variations when the inner cylinder is extending. This orientation is when the outer- and extension cylinders are fully retracted. This keeps the centre of gravity close to the pivoting point, the link between the column and inner boom, and also minimizes the acceleration effects. The dead-weight is then mapped as a function of the extension length, by measuring the pressures on both sides of the inner cylinder at static equilibrium throughout several points of the cylinder extension range.

\[
(4.1) \Leftrightarrow /\ddot{x}_p = 0, \dot{x}_p = 0 / \Leftrightarrow F_L(x) = A_p p_p(x) - A_r p_r(x)
\]  

(4.2)

The term \( F_L \) is then considered as known. The acceleration is eliminated by running the supply pump at a constant speed for each measurement point. The pressures and cylinder extension for a range of pump speeds (and consequently cylinder velocity) are then measured. Equation (4.1) can now be rewritten to the following:

\[
(4.1) \Leftrightarrow /\ddot{x}_p = 0 / \Leftrightarrow B_p \dot{x}_p = A_p p_p - A_r p_r - F_L(x)
\]  

(4.3)

Measurements are done at 12 different speeds, ranging from 2 mm/s to 68 mm/s, with somewhat higher density of measurement points at slow speeds to make sure that the Stribeck characteristics are captured. Transients are removed from the data, and the captured data is stable enough to be approximated by its mean value, which simplifies the calculations drastically. Fluctuations in the values will otherwise lead to a swarm of measurement data at different points in the plot, and reduce the readability of the graph. This approach is acceptable since the measurements are stationary. The cylinder friction is then calculated and plotted as a function of the extending velocity in Figure 4.13.
4.3 Component Validation

Figure 4.13: The obtained Strubeck curve from measurements.

Note the similarities with the Strubeck curve, explained in Section 3.5.1. The lowest point in the graph at 5 \text{mm/s} is low compared to the maximum value. This point corresponds to the Coulomb friction and is approximately 2 \text{kN}. The largest part of the curve is a linear line that increases with speed. This corresponds to the viscous friction and the slope corresponds to the viscous friction constant. It is approximated to $4.25 \times 10^5 \text{ Ns/m}$ which is 20 times higher than in [6], who studied the same crane in 1990.
Chapter 5

Linear analysis

This chapter is intended to give a more thorough insight of the system dynamics and its characteristics by constructing the corresponding linear transfer function, and point out some useful aspects and tools when designing an EHCC. The system resonance frequencies and damping can be identified and the stiffness can be studied. It is however not obvious which mathematical representation that best describes the system. This chapter is about identifying different versions and discussing which transfer function to select.

Figure 5.1: The components included in the transfer function divided into two parts, with the breakpoint at the valve.

It would be convenient to divide the system into two parts and study them independently of each other as seen in Figure 5.1. Transfer functions quickly becomes complex, and it is simpler to analyse parts of the system if they can be studied isolated from each other. A suitable point of dividing the system in two different systems is the breakpoint at the on/off-valve. This point is a natural choice since the valve separates the total fluid into two different volumes, and the valve further describes the flow between them. On each side there is a rotational...
or translational mass that can oscillate together with each volume.

The transfer function is derived from the input torque on the shaft, onwards to the cylinder position. The inverter and motor together are considered as one component with regards to modelling. The provided torque can be assumed instantly delivered with the requested magnitude, given that the operating point is achievable and if flux weakening is accounted for [28]. A thorough model of a version of the electronic motor is constructed and validated in [3]. Modelling at such complex level is in excess of the purpose of this thesis.

5.1 System 1

Using Equation (3.17) for the shaft, Equation (3.12) through (3.15) for the pump, and the volume Equation (3.24), the transfer function from input torque \( T_m \) to pressure before the valve \( p_1 \) can be expressed in a block diagram. The pump-motor efficiencies \( \eta_{hm} \) and \( \eta_{vol} \) are neglected when calculating the flow and pressure, since they have a relatively small effect compared to other parameters. The hydro-mechanical efficiency of the pump is however included as shaft friction. The friction at the motor is negligible compared to the rotational friction at the pump, which can be derived from \( \eta_{hm} \), using Equation (3.16) in [4]. This is a linear approximation of an otherwise complex friction, which can be rearranged as Equation (5.1). Note the division by the shaft speed to achieve correct units of \( R \).

\[
R = \frac{D \Delta p (1 - \eta_{hm})}{2\pi \omega \eta_{hm}}
\]

\( (5.1) \)

\[ p_1 = \frac{2\pi}{D K_c} \frac{V_\text{A}}{\beta e} \left( \frac{2\pi}{D} \right)^2 s^2 + \frac{V_\text{A} R}{\beta e} \left( \frac{2\pi}{D} \right)^2 s + 1 \]  

\( (5.2) \)

Processing the block diagram in Figure 5.2 gives the transfer function according to Equation (5.2). It is of interest to include the flow-pressure gradient in the transfer function to achieve a readable graph, for reasons described later. Using the identification Equation (5.3), the static gain, resonance frequency and damping
5.1 System 1

can be identified as Equation (5.4) - (5.6).

\[ G(s) = \frac{K}{(\frac{s}{\omega})^2 + \frac{2\delta}{\omega}s + 1} \]  

(5.3)

\[ K_1 = \frac{2\pi}{D} K_c \]  

(5.4)

\[ \omega_{h1} = \frac{D}{2\pi} \sqrt{\frac{\beta}{V_J}} \]  

(5.5)

\[ \delta_{h1} = \frac{R\pi}{D} \sqrt{\frac{V_J}{J \beta_e}} \]  

(5.6)

Figure 5.3: Bode graph of the first system.

The bode diagram for system 1 is plotted in Figure 5.3. The gain is low for all frequencies due to the inclusion of \(K_c\), and there is a low damped resonance peak (0.016) at 10.9 rad/s. There is no interest in studying this graph further as it later turns out that this information is not a correct representation of the system.
5.2 System 2

The second system describes the behaviour from the valve onwards to the piston position. A good note regarding cylinder friction and stability is mentioned in [15] Equation (26), which implies that the coulomb friction might cause instability at small speeds. However, only the linear viscous friction is considered in the transfer function.

![Block diagram of the second system](image)

Figure 5.4: Block schedule of the second system.

The block diagram in Figure 5.4 produces the transfer function in Equation (5.7) after the above mentioned modifications. \( V_{eq} \) is the equivalent volume, which corresponds to the volume before the cylinder \( V_B \) and the volume in the cylinder, which in the worst case is as large as possible: close to maximum stroke. \( V_{eq} = V_B + A_p x_{p_{max}} \). As the volume increases, the resonance frequency becomes smaller.

\[
\frac{x_p}{p_1 K_c} = \frac{A_p}{s \left( \frac{V_{eq} M_1}{\beta_c (A_p^2 + K_c B_p)} s^2 + \frac{(K_c M_1 + \frac{V_{eq}}{V_{eq}})}{(A_p^2 + K_c B_p)} s + 1 \right)} \tag{5.7}
\]

Identification using Equation (5.3) gives the static gain, resonance frequency and damping according to Equation (5.8) - (5.10).

\[
K_2 = \frac{A_p}{A_p^2 + K_c B_p} \tag{5.8}
\]

\[
\omega_{h2} = \sqrt{A_p^2 + K_c B_p} \sqrt{\frac{\beta_c}{M_1 V_{eq}}} \tag{5.9}
\]

\[
\delta_{h2} = \frac{K_c \sqrt{M_1 \beta_c} + B_p \sqrt{\frac{V_{eq}}{M_1 \beta_c}}}{2 \sqrt{A_p^2 + K_c B_p}} \tag{5.10}
\]

Since the flow-pressure coefficient is moved upstream from the sum it is natural to include it in the first system. The breakpoint between the systems is set upstream of this sum. Compare Figure 5.5 and 5.6.
5.2 System 2

\[ K_c \]

Figure 5.5: The initial block diagram for the valve.

\[ \begin{array}{c}
\text{System 1} \quad \text{System 2} \\
\includegraphics[width=0.5\textwidth]{image1}
\end{array} \]

Figure 5.6: The manipulation of the block diagram, and the natural breakpoint between system 1 and 2.

\[ K_c \text{ is the linearised flow-pressure gradient from Equation (3.20). Small } K_c \text{ is equivalent to a low flow at a certain pressure drop. Normally, } K_c \text{ is in the } 10^{-11} \text{ range. The gradient is however much larger for this system } (10^{-9}), \text{ since a valve is selected specifically that has as small effect as possible on the flow. For normal values, } A_p^2 >> K_cB_p \text{ is true, but since the selected valve is intended not to introduce any losses, the second term in } (A_p^2 + K_cB_p) \text{ is larger and can not be ignored. The result is a somewhat less clear expression. A quick note can be made regarding the damping in Equation (5.10). Increased } K_c \text{ means less flow resistance through the valve which further results in more damping.}

\[ \begin{array}{c}
\begin{align*}
\text{For this system, no complex conjugated pair of poles is present and therefore there is no resonance peak, see Figure 5.7. The phase margin at } 0 \text{ dB is positive which implies that the system is stable. An interesting comparison is made with the flow-pressure coefficient of a regular valve. This system shows a resonance peak at approximately } \omega = 98 \text{ rad/s, and } \delta = 0.07. \text{ The gain is larger than 1 at the peak, and the phase is negative. This system is, at first glance, not stable. This illusion is due to the exclusion of } K_c \text{ from the second system. When system 1 and 2 are combined, both flow-pressure characteristics are stable. Solving Equation (5.7) for the resonance frequency produces the gain at the peak in Equation (5.11).}
\end{align*}
\end{array} \]

\[ \text{Amp}_{\text{peak}} = \frac{A_p}{K_c \frac{A^2_p}{M_e} + \frac{B_p}{M_t}} \]  
(5.11)
5.3 Combined System

What is more important than the partial systems separate behaviour, is when they are combined together forming the complete system. The transfer function is simply a multiplication of both systems. In Figure 5.8 it is clear that the complete system is stable since the gain for both systems are smaller than -10 dB at the peaks, with a magnitude of -18 dB for the current system and -50 dB for the regular system. The larger margin for the regular system is due to the presence of $K_c$ in the gain. For both curves the resonance peak for the first system is the critical one. For a regular system this would have to be considered when dimensioning the components. This is however not the case for the version in this report, since it does not consider the flow out of the first volume and the transfer function is a false representation of the true system. This effect is included later.

Figure 5.7: System 2 with $K_c$ of a regular valve, and the valve used in the EHCC.
5.4 Valve-Free System

The valve is present only for safety reasons, and is ideally not affecting the system. For such a system there is only one volume $V_{tot} = V_A + V_{eq}$, and the flow in and out of it is the pump flow minus the cylinder stroke. The block diagram will have the appearance as illustrated in Figure 5.9. From the block diagram the transfer function in can be derived as Equation (5.12), and the bode diagram as in Figure 5.10.

$$\frac{x_p}{T_m} = \frac{2\pi A_p}{\left( \frac{JV_{tot}}{\beta_n} \right)^2 s^2 + \frac{RV_{tot}}{\beta_n} \left( \frac{2\pi}{D} \right)^2 s + 1} \left( M_1 s^2 + B_p s \right) + (J s + R) \left( \frac{2\pi}{D} \right)^2 A_p^2 s$$

Figure 5.8: System 1 & 2 in series, with different $K_c$.

Figure 5.9: Transfer function of the valve free system.
The bode plot is quite different compared to the combined system. The lower resonance peak is not present at all. This peak represents the small volume resonating with the rotational inertia, which now has changed in size and is influenced by the cylinder as well. The system is stable since the gain is almost -100 dB at the phase of -180 degrees. One complex conjugated pair of poles can be observed at 86 rad/s with a damping of 0.06. The damping is however not analytically obtained, but instead extracted from the location of the poles from now on. The damping is defined as the cosine of the angle to the poles in polar coordinates from the real axis in the complex plane [5]. This peak is being further discussed with regards to the stiffness of the system. The conclusion can be drawn that the lower resonance peak is not an issue if the effects of the valve is negligible. This model is overall considered as stable.
5.5 Comparison of Transfer Functions

The question arises which transfer function that most accurately represents the proposed concept studied in this thesis. The previous models are quite different from one another. To further investigate this issue, the first transfer function with system 1 & 2 combined is modified to account for flow out of the first volume. This gives the most accurate representation of the EHCC. As seen in Figure 5.11, the block diagram and the corresponding transfer function in Equation (5.13) is a bit more advanced.

\[
\frac{X_p}{T_m} = \frac{\frac{A_0}{s}}{\left(\left(\frac{1}{\omega_p^2} (M_1 s^2 + B_p s) + A_0^2\right)\left(\left(\frac{1}{\omega_p^2} + \frac{1}{\omega_c^2}\right) - \left(\frac{D_2}{\omega_p^2}\right) + (J_s + R) + (J_s + R) \left(\frac{M_1 s^2 + B_p s}{\omega_c^2}\right) + (J_s + R) \left(\frac{1}{\omega_c^2} - \left(\frac{D_2}{\omega_p^2}\right)\right)\right)\right)}
\]

Figure 5.11: The block diagram of a more complex interpretation of the system.

Figure 5.12: Shows the transfer function with the inclusion of the output flow from the first volume. \(K_c\) is that of this system.
The bode plot in Figure 5.12 is stable, with a large margin of almost 100 dB at -180 degrees, and a resonance peak at 86 rad/s with a damping of 0.06. The system is very similar to the system with the negligible valve. There is no low frequency resonance peak due to the flow out of the first volume. Putting the three transfer functions side by side confirms the similarities between the valve free system and the more complex flow considered system. It also shows that the system consisting of two separate systems combined is off from the other two.

![Comparison between transfer functions, $K_c = 10^{-9}$](image)

Figure 5.13: A comparison between the three different transfer functions with $K_c = 10^{-9}$ for the valve in this system.

The conclusion can be drawn from Figure 5.13 that for $K_c = 10^{-9}$ which is the range of the flow-pressure coefficient of the valve in this system, the transfer function with a negligible valve is a good representation of the system while the combined system is not. Continuing the study and decreasing $K_c$ to the range of a regular valve, the combined system and the flow considered system still does not coincide as seen in Figure 5.14. This implies that not even with a regular valve the volumes can be considered as two separate, and the combined system still does not represent the true system. Decreasing $K_c$ to extreme values in Figure 5.15 finally gives another result.
5.5 Comparison of Transfer Functions

Figure 5.14: A comparison between the transfer functions with separate volumes, with and without flow between them, and a regular flow-pressure gradient.

Figure 5.15: A comparison between the transfer functions with separate volumes, with and without flow between them, and an extremely low flow-pressure gradient for demonstration.

At $K_c = 10^{-13}$ the flow considered system finally demonstrates similar behaviour to the combined one. In physical terms this corresponds to having a very small valve or very large flow resistance through it. So large that the volumes can be viewed as two separate ones. Finally, one important question to answer with these results is: for what $K_c$ is the valve negligible? In Figure 5.16 the valve free, and the flow considered transfer functions are plotted with variations of $K_c$. It is
evident that the valve does not have to be considered for flow-pressure gradients in the $10^{-9}$-range. The borderline case is approached in the range of $10^{-10}$, and for lower values the dynamics are lost. The conclusion can also be made that when considering the dynamics of this system, the valve free transfer function can be used.

![Valve free system and flow-considered system](image)

Figure 5.16: The valve free transfer function, and the advanced, flow considered transfer function, with varied flow-pressure gradient.

From Figure 5.16 the conclusion is that the effects of the valve can be neglected if its flow-pressure gradient is in the range of $10^{-9}$ and all the way down to $10^{-10}$. For smaller flow gradients the valve cannot be neglected any more as the valve free transfer function no longer coincide with the flow considered system.

### 5.6 Stiffness Analysis

The stiffness of a transfer function is a measure of how susceptible the system is to disturbance. The higher stiffness, the lower sensitivity to disturbance. Since the transfer function with the valve neglected is a good representation of the system, this transfer function is a reasonable basis for the stiffness analysis. Using the block diagram (5.12), the full transfer function with both input torque and disturbance can be derived.

$$x_p = \frac{2\pi}{D} A_p \left( \frac{JV_{tot}}{\beta_c} \left( \frac{2\pi}{D} \right)^2 s^2 + \frac{RV_{tot}}{\beta_c} \left( \frac{2\pi}{D} \right)^2 s + 1 \right) F_L \left( \frac{JV_{tot}}{\beta_c} \left( \frac{2\pi}{D} \right)^2 s^2 + \frac{RV_{tot}}{\beta_c} \left( \frac{2\pi}{D} \right)^2 s + 1 \right) (M_1 s^2 + B_p s) + (J s + R) \left( \frac{2\pi}{D} \right)^2 A_p^2 s$$

(5.14)
5.6 Stiffness Analysis

Setting $T_m = 0$ and rearranging the results in the stiffness Equation $S$ further gives the expression in Equation (5.15).

\[
S = -\frac{F_L}{x_p} = \frac{\left(\frac{JV_{tot}}{\beta_e} \left(\frac{2\pi}{D}\right)^2 s^2 + \frac{RV_{tot}}{\beta_e} \left(\frac{2\pi}{D}\right)^2 s + 1\right) \left(M_1 s^2 + B_p s\right) + (J s + R) \left(\frac{2\pi}{D}\right)^2 A_p s}{\left(\frac{JV_{tot}}{\beta_e} \left(\frac{2\pi}{D}\right)^2 s^2 + \frac{RV_{tot}}{\beta_e} \left(\frac{2\pi}{D}\right)^2 s + 1\right) \frac{2\pi}{D} A_p} \tag{5.15}
\]

Figure 5.17: The stiffness Equation of the system without on/off valve.

A bode plot of the expression in Equation (5.15) can be seen in Figure 5.17. There are three points in this graph which are worth discussing. These are the static gain, the resonance peak and the anti-resonance peak. Following the end of the graph, the static gain is 133 dB or $4.5 \cdot 10^6$ N/m. Studying the appearance of the graph, the system is the most susceptible to disturbance at static conditions given that no feedback of $x_p$ is implemented. The system will never be less stiff than this. The positive peak is due to the complex conjugated pair of poles in the denominator in Equation (5.15). The poles can be derived analytically by comparing the denominator to the identification Equation (5.3) which is seen in Equation (5.16).

\[
\omega_{h1} = \frac{D}{2\pi} \sqrt{\frac{\beta_e}{V_{tot} J}} = 5.14 rad/s \tag{5.16}
\]

The anti-resonance peak is the result of the complex pair of zeros in $S$ at 86 rad/s. This is the inverse of the peak appearing in the transfer function of the valve free system, and can be seen in Figure 5.10 for comparison. The inverse peak has a
damping of 0.06, which is low. The system components should be dimensioned in such a way that the damping is not further reduced, since this will result in the inverse peak growing even deeper and possibly becoming the new lowest point in the graph. Another important aspect is the stiffness curve which is changing if the system is implemented with feedback of \( x_p \). The amplitude margin in Figure 5.10 is approximately \(-100 \text{ dB}\), and with the rule of thumb considered in [1] the feedback gain can be expected to be in the range of \( 10^5 \). Setting the input value to \( T_m = 0 \) results in the block diagram according to Figure 5.18, where \((5.14)_{num}\) and \((5.14)_{den}\) is the nominator and denominator of Equation (5.14) respectively. \( K \) is the proportional feedback gain, and the resulting modified stiffness function is seen in Equation (5.17).

\[
\frac{F_L}{x_p} = \frac{4.16_{den} + K}{4.16_{num}} = \frac{S_{num} + K}{S_{den}}
\]  

(5.17)

Figure 5.18: The block diagram of the system with feedback, disturbance and zero reference.

Figure 5.19: The stiffness of the system with feedback.
5.7 Load Dynamics

The dynamics of the crane that enters the system have previously been modelled as a disturbance $F_L$. There is however another representation of the crane which is more accurate, which is illustrated in Figure 5.20.

The second spring-mass system represents the outer links of the crane, where the spring and damper is a consequence of the hydraulic capacitance and friction of the outer cylinder respectively. The external load seen as a disturbance, is entering the system at the crane tip and travels through the crane’s structural components. The mass $M_2$ is the equivalent mass of the crane outer boom and extension. The equivalent mass is generally the effect of a mass with a lever if they are to be replaced with a mass directly connected to the cylinder. It is further a simplified representation of the system. $M_1$ is the equivalent mass of the inner boom, and can be expressed mathematically as in Equation (5.18).

\[
\begin{align*}
J_1 &= m_i \cdot l_i^2 \\
J_2 &= J_2 \\
m_{eq} &= \frac{l_2}{l_1} \cdot m_1
\end{align*}
\]

(5.18)

A quick check of the added dynamics is done by deriving the transfer functions
from the tip of the crane onto piston position using the free body diagram in Equation (5.19).

\[
\begin{bmatrix}
M_1 & x_p \\
0 & M_2
\end{bmatrix} s^2 + \begin{bmatrix}
B & -B \\
-B & B
\end{bmatrix} \begin{bmatrix}
x_p \\
x_L
\end{bmatrix} s + \begin{bmatrix}
C & -C \\
-C & C
\end{bmatrix} \begin{bmatrix}
x_p \\
x_L
\end{bmatrix} = \begin{bmatrix}
p_2 A_p \\
F_L
\end{bmatrix}
\]

\[\Leftrightarrow X = (Ms^2 + Bs + C)^{-1} F \Leftrightarrow \begin{bmatrix}
x_p \\
x_L
\end{bmatrix} = \begin{bmatrix}
f_{11} & f_{12} \\
f_{21} & f_{22}
\end{bmatrix} \begin{bmatrix}
p_2 A_p \\
F_L
\end{bmatrix}
\] (5.19)

The data is fetched from paper [6] which is based on a hook load of 130kg. It is evident that \(f_{12}\) describes how the disturbance affects the piston position. Plotting this transfer function results in the graph in Figure 5.21. The dashed lines indicates a decrease of \(M_2\) by 20\% above, and an increase of 200\% below the solid line. A large variation is required, since a small increase in load at the tip will produce a large increase in the equivalent mass, see Equation (5.18). 200\% covers the majority of the crane’s working range.

![Figure 5.21: The transfer function of the disturbance through the crane boom system, with limits of the equivalent mass indicated between -20 - +200\%.

The result is a low gain response with a resonance peak at approximately \(\omega = 32 rad/s\) and does not change considerably despite large variations of \(M_2\). The low gain will result in increased open loop stiffness for the inner piston \(x_p\). How stiff the outer piston is will not be investigated. The small peak will result in a dip in the stiffness graph between the existing peak and dip. The only concern will be if the two inverse peaks coincide. This will not be an issue since the mass \(M_2\) would have to be only 2\% of its current value which is unreasonably low. With current values, the stiffness is in the range of \(10^{16}\), which is extremely stiff and therefore of no concern which is seen in Figure 5.22.
Additional Remarks

A more thorough modelling of the crane, both inner and outer cylinder, can be found in paper [6]. The system operating in both directions will have the mathematical structure closer to classical transfer function modelling, as there will be a finite volume and an orifice on the outlet side of the cylinder as well. The worst case stroke will instead be around $\frac{x_{p_{\text{max}}}}{2}$ and the hydraulic resonance frequency will increase. More information regarding this type of analysis can be read at [14], where the damping of the crane is investigated more closely. Another great reference with regards to the complex dynamics for this type of cranes can be read in [26].
Chapter 6

System Performance on the Loader Crane

The next following pages contains measurements of the EHCC connected to the loader crane, as well as comparisons to the reference system. Lastly, a standalone performance evaluation of the induction motor is presented.

6.1 Crane Response

The crane response is examined with two of the test rig setups. Firstly, the reference system using a load sensing control valve as shown in Figure 4.6 is tested, to further be compared with the pump controlled cylinder system according to Figure 4.8.

6.1.1 Load Sensing System

The load sensing system also referred to as the reference system consists of two separate control systems where everything except the LS valve is controlled through the dSpace Autobox, which is further explained in Section 4.2.3. However, this leads to issues regarding the measurement data since the reference control signal from the IQAN-joystick is recorded on another recording device than the rest of the system signals. Pairing these signals together gives what is illustrated in Figure 6.1, where the control reference signal is manually synced with the rest of the signals. This means that the reference Valve signal in Figure 6.1 contains errors regarding exact time stamps. However, the most interesting thing is not the time delay between requested speed and actual motion, but instead the cylinder acceleration itself.

The response test with the load sensing system is carried out with an unloaded crane where the fixed pump is running at a constant speed of 1200 rpm, which generates a flow corresponding to the rated working flow of 40 l/min for the crane used in the experiment. The control joystick connected to the IQAN system is
further quickly moved to maximum position for a few seconds and then released back to its original position.

To determine the cylinder velocity the cylinder position is measured with a linear position sensor mounted directly on the boom cylinder. The measured signal is then filtered and differentiated.

![L90LS Response](image)

Figure 6.1: A step in maximum control valve signal on the right axis with corresponding cylinder velocity on the left, which is derived from filtered and differentiated cylinder position measurements.

From the step response in Figure 6.1 it can be seen that a fully opened control valve gives a cylinder velocity of 0.05 m/s. Further the speed of the system can be determined, where the time constant is defined as 63 % of the end value at 0.05 m/s [5]. The time constant of the LS controlled system is \( a = 0.11 \) s. It can also be seen that the cylinder velocity is rather oscillative at the endpoints which gives a fitful feeling to the system.

6.1.2 Pump Controlled System

The direct pump-controlled system according to Figure 4.8 is entirely controlled through the Autobox. A simple lever is used during the response test, programmed to give a maximum speed reference of 1200 rpm in order to limit maximum hydraulic flow to the rated work flow of 40 l/min. The lever is used in the response test partly to imitate the lever movement made in the load sensing response test, but also due to safety reasons since the ceiling height in the lab is limiting the movement of the crane.

The test is carried out in the same way as for the load sensing system with an unloaded crane. The lever is quickly moved to maximum position for a few seconds and is then released to its original position. The step response is presented in Figure 6.2. The reference signal limit ensures that maximum speed in the pump-controlled case correlates to the maximum valve signal in the load sensing case.
6.1 Crane Response

Figure 6.2: A step in maximum motor speed on the right axis with corresponding cylinder velocity on the left, which is derived from filtered and differentiated cylinder position measurements. The transients seen at 17 and 19.5 seconds are sensor disturbance.

From the results in Figure 6.2 it is easy to see that the maximum speed corresponds to the intended maximum velocity 0.05 $m/s$ as planned, which means that the pump controlled crane system in this case can reach same maximum speed as the load sensing alternative at least for an unloaded case. In fact, the cylinder velocity is not overshooting at the end points or showing oscillative behaviour in the same way as with the LS control valve. However, the pump controlled system is slower. The time constant of the pump controlled system is $a = 0.59$ s, which compared to the LS system is about five times slower.

Another interesting phenomenon seen in the step response is the slow deceleration of the motor. Firstly, it is about twice as slow as the acceleration of the crane even though it is unloaded. The acceleration and deceleration also follows what looks like a ramp, which is a behaviour specifically disabled within the inverter configuration. In practice when operating the crane with the pump controlled system, this deceleration behaviour keeps the crane moving upwards after releasing the control lever.

By studying the example where the motor is actively braked by creating an inverse speed reference with the control lever in the deceleration phase, it turns out that deceleration can be made faster. In Figure 6.3 this experiment is tested during the same crane conditions. It is further obvious that deceleration time can be reduced by at least 50 %. It is also proven that there is power available to generate a higher braking effect.

The pump controlled cylinder setup according to Figure 4.8 is an open system where the piston rod side of the cylinder is directly connected to tank. Unlike a valve controlled system like the load sensing setup in Figure 4.6, the pump controlled setup cannot force the cylinder down with pressure.
In order to further understand whether the pump controlled system maintains the response even in a lowering motion, a test is made which is presented in Figure 6.4. It turns out that the time constant for the lowering motion is $a = 0.54\, s$, which is slightly faster than for the lifting motion. The deceleration ramp which is noticed during the lifting motion can almost be identically seen in the lowering motion.

Figure 6.4: A step in maximum motor speed in the reversed (lowering) direction. The transients seen is sensor disturbance.
6.2 Pressure Equalizing

The on/off-valve which operates as a safety function in the pump controlled concept, introduces an issue when the crane begins a lowering motion. The purpose is to prevent the crane from undesirable lowering. Due to internal leakage in the pump the pressure between pump and valve slowly descends to atmospheric pressure. When the valve suddenly opens again the pressure on both sides equalizes quickly, which leads to pressure drops and an unpredictable crane behaviour.

To illustrate the behaviour a test is made where a lightly loaded crane is lowered from a standstill position with a constant speed reference of 50 rpm. The on/off-valve is programmed to open as the motor speed reaches a threshold of 40 rpm in any direction. What happens is illustrated on the left side in Figure 6.5. It turns out that the valve controlled by an unfiltered speed signal starts to oscillate fast as the pressure fluctuates around the threshold speed at approximately $s$. However, the interesting behaviour is seen in the cylinder pressure, as it suddenly drops down and starts to oscillate with an initial amplitude of 12 bar. The motor speed also overshoots with a value of over three times the speed reference, which affects the ability of controlling the lowering crane motion negatively. The speed is not stabilized at the end value until almost after 5 s.

A simple pressure equalizing function is introduced which initially inverts and amplifies the control signal until a pressure difference $\Delta p$ of less than 5 bar across the valve is reached. The result is illustrated on the right side in Figure 6.5. The most significant difference is the smaller pressure drop, even though some oscillations still can be noted. However, the amplitude of the initial oscillations is approximately 5 bar, which is about half the amplitude than the one without the pressure equalizing function.

Another observation made is that the motor speed overshoots even if the pressure is equalized across the valve. The amplitude of the overshoot is similar to the test before, and the time until the speed is stabilized on the end value is 4 s.
Figure 6.5: A lowering motion with a constant speed of 50 rpm started from a standstill position with a lightly loaded crane. The pressure behaviours are illustrated as the on/off-valve opens at the motor speed threshold of 40 rpm with (right) and without (left) pressure equalizing function.

### 6.3 High Load, Low Speed

One special case to examine is the condition when lifting a heavy load, which preferably is moved slowly. There is an issue of driving the motor at high load and low speed, as the armature currents increases drastically. The reason why this occurs can be derived from the equivalent circuit, Figure 3.1. Principally, the impedance can be written as

$$Z = \frac{R_2}{s} + jX_R$$

(6.1)
where $X_R$ is the total rotor reactance. The slip can vary between 1 and 0, and the slip is starting at 1 at low speed and then decreases as the speed increases [28]. A consequence of this is low impedance at slow speeds, as the real term $\frac{R_2}{s}$ is low, and increases with reduced slip [12]. The results are large electrical currents during start-up, especially with high load. Induction motors are generally known for poor start-up behaviour [9, 16].

To expose the motor to similar circumstances the components are connected according to the rig described in Section 4.2.1. The reference speed is set to 100 $rpm$, corresponding to approximately 5 $mm/s$, while the variable orifice is gradually tightened by hand. This setup will prove how good the low speed performance is for the motor.

![Graphs showing speed, pressure, and current during low speed measurement with increasing load.](image)

Figure 6.6: A measurement at low speed with increasing load. The top graph displays the speed, the middle graph displays the pressure which is proportional to the torque, and the lower graph displays the armature current and its components.

Generally, the inverter experience difficulties when following the reference at such a low speed, as it is disturbed by the counteracting torque induced by the pressure through the pump. One cause of this is that the speed controller in the inverter not tuned aggressively enough. As the orifice is tightened and the pressure increases, so does the armature current. Towards the end of the measurement the motor speed drops considerably due to the smaller margins of the motor overcoming the increased counteracting torque. Note that the motor must overcome the counteracting torque to accelerate to the reference speed. At this point the armature currents has reached its limits and the measurement is stopped. The attained pressure in the system is almost 110 Bar which is low, due to the knowledge of the crane originally operating at a maximum of 225 Bar. This is an indication of poor performance of an otherwise over-dimensioned motor, and an Achilles heel for this EHCC.
Chapter 7

Energy and Efficiency Analysis

In this chapter the energy study of the EHCC is presented. The efficiency is analyzed during both the lifting and lowering motion, and the regeneration capabilities are evaluated. A brief, qualitative discussion is held that compares the results with a load sensing system.

7.1 System Boundaries

In order to evaluate and quantify the energy efficiency of the pump controlled concept, the system boundaries must be defined. The measured input quantity is DC power, which is directly calculated by the inverter and determined in both consuming and regenerating energy direction. This means that the battery is excluded from the energy and efficiency analysis which also can be seen in Figure 7.1.

Figure 7.1: Pump controlled setup where the system boundaries are illustrated with a dashed line. Input quantity is DC power to the inverter and output quantity is cylinder power acting directly on the crane.

65
The measured output quantities are cylinder pressure \( p_2 \) and cylinder position. The cylinder position is further filtered and differentiated to obtain the cylinder velocity. Since the cylinder area is known, the output quantity of the system is cylinder power. However, the output quantity also includes friction in the cylinder. By assuming an approximation of the friction derived in Chapter 4.3.3, the output system boundary can be further pushed to not include friction forces in the cylinder. In order to make a fair approximation of the experimentally derived friction, it can be assumed as a linear viscous friction with a coulomb friction of 6 kN which is illustrated with a red dashed line in Figure 7.2. The higher coulomb friction level is chosen for conservative reasons.

![Figure 7.2: A linear approximation of the measured Strubeck friction curve derived in Section 4.3.3, where a constant friction of 6 kN is chosen for lower velocities to represent the coulomb friction.](image)

Since the pump controlled system not only consumes energy but also regenerates energy, the energy input and output directions of the system switches depending on if the crane is regenerating energy or not. To determine the energy efficiency of the system, a strategy of handling the friction force depending on energy direction is needed. The strategy used in this energy study is illustrated in Figure 7.3. The figure represents a simplified lifting and lowering case where DC input and output power are illustrated by red and green colour. The effective and stored power acting on the cylinder represented by a solid blue line are derived from pressure and cylinder position measurements in both energy directions. By subtracting the friction force in the consuming case and adding the friction force in the regenerative case, the effective power curve can be displaced to represent actual cylinder power acting on the crane and the lifted load.
7.2 Energy Study

The energy study of the EHCC is divided into two parts. Firstly the energy consumption and regeneration behaviour of the system is studied during a lifting cycle. Secondly the regeneration capability is specifically studied during different lowering speeds. The idea of the lifting cycle is to cover some of the different situations such as normal lifting speed, fast lifting and fast lifting with active braking. The cycle is carried out by hand with the control lever in the previous mentioned order, which obviously includes the risk of lifting cycle deviations. The main purpose is, however, not perfect repeatability but rather understanding the overall energy behaviour of the pump controlled system.

7.2.1 Lifting Cycle Energy Regeneration

The lifting cycle mentioned earlier is first performed without any load. The resulting energy consumed and regenerated is illustrated in Figure 7.4. According to the highlighted areas, braking in the end of the lowering motion is a significant part of the regeneration. A total amount of 10.44% of the consumed energy is regenerated during these lowering motions. The amount of consumed energy which is regenerated during the lifting cycle is calculated according to Equation (7.1). However, the proportion of regenerated energy is not including the capability of accumulating the DC energy \( W_{DC_{reg}} \) within the battery according to the system.
Energy and Efficiency Analysis

boundaries explained in Section 7.1.

\[ \eta_{reg} = \frac{W_{DC_{reg}}}{W_{DC_{in}}} \]  

(7.1)

Figure 7.4: Lifting cycle performed with the unloaded pump controlled system. Consumed energy is highlighted with red and regenerated with green.

An arbitrary concrete load of approximately 250 kg is used to increase the cylinder potential energy in order to evaluate possible differences in regeneration behaviour, but without the risk of exceeding any system limitations. The loaded lifting cycle is presented in Figure 7.5.

Figure 7.5: Lifting cycle performed with the 250 kg loaded pump controlled system. Consumed energy is highlighted with red and regenerated with green.
It can be seen that the regenerated energy increases together with the load weight, which during the loaded case is 22.79%. However, it is difficult to determine the specific relationship between increasing weight and regenerated energy, since the lifting cycles deviate and the load weights are not precise.

### 7.2.2 Energy Regeneration Capability

To further understand the connection between lowering speed and the capability of regenerating potential energy, experiments with only lowering cycles are carried out with different speeds. The same concrete load weight of approximately 250 kg is used in order to maximize the possibility of a significant result. Every cycle is starting at the same height, however, the stop position differs since the control lever is used for safety reasons. The reference signal is limited to the intended lowering speed for every cycle, and the manual control lever is pushed to its limit for as long as possible and then released to its original position. The total amount of regenerated energy from each cycle is adjusted with the travelled distance, which gives a ratio between regenerated energy and lowered distance. In this way the ratio from every cycle gets comparable despite the deviations in lowering distance.

![Regeneration Capability](image)

Figure 7.6: Regeneration capability during crane lowering with different motor speeds. The black line represents the intended motor speed for each cycle, and the red line corresponds to the maximum reached motor speed during each lowering cycle. The trend between every cycle is illustrated with a dashed line.

According to the test result in Figure 7.6, there is clearly a difference between intended speed and maximum reached speed. However, there is a wide representation of lowering speeds which gives a picture of the regeneration capability. In the graph of the regeneration ratio an interesting behaviour emerges, where there seems to be an optimum lowering speed. Furthermore, the ratio decreases with increasing speed which experimentally seems to confirm Minav’s simulation
based conclusion that regenerated energy decreases proportionally with increasing lowering speed [20].

## 7.2.3 Idling Losses

By studying the lifting cycles performed with the pump controlled system in Section 7.2.1, the electric motor obviously consumes energy even if the crane does not move. A clipboard of the first lifting movement in the loaded cycle seen in Figure 7.5 is presented in Figure 7.7.

![Clipboard of Loaded Lifting Cycle](image)

Figure 7.7: A clipboard of the loaded lifting cycle illustrated in Figure 7.5, where the DC energy consumption of a loaded crane during stand still is illustrated together with the pump pressure captured between pump and on/off-valve.

Compared to the beginning of the lifting cycle at time 0 s when the pump pressure is low, the DC power is close to zero. However, when the lifting move-
ment is complete and the crane stops, the load pressure from the cylinder is partly captured between the on/off-valve and the pump. Since the speed reference of the motor at this moment is set to zero, the motor must actively provide a countering torque to prevent the pump from rotating in the reversed direction as a result of the captured pressure. The countering DC power in Figure 7.7 is approximately 400 W at a captured pressure of 30 bar. However, due to internal leakage in the pump, the captured pressure decreases over time.

7.3 Efficiency Analysis

In order to determine the energy performance of the physical pump controlled system, the efficiency is calculated for both the lifting and lowering motion separately. The efficiencies are determined by Equation (7.2) and (7.3), where the input and output quantities are measured at the system boundaries mentioned in Section 7.1.

\[ \eta_{\text{up}} = \frac{W_{\text{cylout}}}{W_{\text{DCin}}} \]  
\[ \eta_{\text{down}} = \frac{W_{\text{DCreg}}}{W_{\text{cylin}}} \]  

The effective work at the input and the output side of the system is determined by integration of the cylinder and DC power illustrated in Figure 7.8. The graph is based on the same measuring sequence as for the regeneration study in Section 7.2.1. By studying the DC power at the second and third lowering sequence, a positive power peak is visible as the crane is quickly accelerated in the lowering direction.

Figure 7.8: Adjusted effective work of the unloaded pump controlled lifting cycle, compared to the input DC power.
Since the crane in the unloaded case is not heavy enough to accelerate the pump by its weight the electrical motor needs to assist. However, the positive DC power peaks are ignored since only the amount of regenerated energy during the lowering motion is interesting in order to determine the effective work from cylinder power to regenerated DC power. The efficiencies are further presented in Table 7.1, where a total lifting cycle efficiency is introduced according to Equation (7.4) which also corresponds to the amount of regenerated energy. To verify that the efficiencies of the lifting and lowering motions are correctly determined, the total efficiency can be compared with the regeneration efficiency $\eta_{\text{reg}}$ derived in Section 7.2.1.

$$\eta_{\text{tot}} = \eta_{\text{up}} \cdot \eta_{\text{down}}$$

(7.4)

Table 7.1: Efficiency values of the unloaded lifting cycle.

<table>
<thead>
<tr>
<th>$\eta_{\text{up}}$</th>
<th>44.18 %</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_{\text{down}}$</td>
<td>22.99 %</td>
</tr>
<tr>
<td>$\eta_{\text{tot}}$</td>
<td>10.16 %</td>
</tr>
<tr>
<td>$\eta_{\text{reg}}$</td>
<td>10.44 %</td>
</tr>
</tbody>
</table>

The efficiencies are further studied in the loaded case shown in Figure 7.9. In this case the regeneration of energy begins almost immediately with the lowering motion. Furthermore, any positive DC power peaks can not be seen, but the cylinder power is intermittently affected by disturbances from the position sensor. However, these are not removed before the efficiencies are calculated. The results are presented in Table 7.2.

Figure 7.9: Adjusted effective work of the loaded pump controlled lifting cycle, compared to the input DC power. The cylinder power partly derived from cylinder velocity calculations contains transients caused by sensor disturbances.
Table 7.2: Efficiency values of the loaded lifting cycle.

<table>
<thead>
<tr>
<th>Efficiency Type</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \eta_{up} )</td>
<td>58.30%</td>
</tr>
<tr>
<td>( \eta_{down} )</td>
<td>42.59%</td>
</tr>
<tr>
<td>( \eta_{tot} )</td>
<td>24.83%</td>
</tr>
<tr>
<td>( \eta_{reg} )</td>
<td>22.79%</td>
</tr>
</tbody>
</table>

By studying the resulting efficiencies from the loaded case there is clearly a difference compared to the unloaded lifting cycle. Both of the efficiencies increases with higher load weight. Even though the lifting cycle deviates between loaded and unloaded conditions and the load weight is not exactly known, the increasing regeneration efficiency according to \( \eta_{down} \) shows that higher load weights increases the ability of regenerating potential energy. Furthermore, the results are verified with the regenerating efficiency \( \eta_{reg} \) from each lifting cycle, which increases the reliability of the efficiency determination.

The efficiency of the lowering motion corresponds to the amount of potential energy which is transformed back to useful regenerated DC power within the inverter. However, since the system boundaries not includes the lead-acid battery used as energy source, the capability of accumulating the electrical energy is not considered by this result.

A complete experimental measurement of the reference load sensing system efficiencies is performed, but the results are considered not to be valid. The shunt valve within the load sensing control valve shows an unexpected behaviour which causes the system pressure relief valve to trigger at lowering motions, which further makes the functionality of the main control valve questionable. Therefore an experimentally based comparison is not presented.

However, a hydraulic system controlled through a conventional directional valve suffers from throttling losses which according to studies reaches 35% [24]. Moreover, the pump in a valve controlled system is rotating continuously which further contributes to energy losses primary during idling. Considering a load sensing system in particular, an important loss contributor occurs when operating more than one function simultaneously. Since the heaviest of two separate loads controls the system pressure, the pressure drop across the main control valve for the less heavier function gets unnecessarily high.
Chapter 8

Discussion

This chapter discusses the main parts of the obtained results, but also briefly some possible improvements which can be made to the system according to experiences gained during the project.

8.1 Test Rig and System Components

The on/off-valve, induction motor and the cylinder friction are three components specifically studied in order to determine characteristics and behaviours of each component.

On/Off Valve

The pressure sensors are mounted directly onto the on/off-valve, which ensures an accurate measurement of the valve but will not truly represent the orifice of the whole system. The valve itself is most likely not the hydraulic bottleneck, but all the hoses and connections which are small in comparison adds to the flow restriction. The effect of these loss contributors is that the flow resistance increases and the system becomes more damped. Another remark is that the valve is not symmetric, and the flow is further not symmetric either. This fact is overlooked as the differences are assumed small, and is supported by the component’s data sheet.

Induction Motor

Without the opportunity to log the armature current components, $i_D$ & $i_Q$ in the inverter interface the validation of the induction motor would not be possible. The inverter documentation states an error margin of $\pm 10\%$, which certainly adds an uncertainty factor to the results. Achieving an exact torque estimation in the flux weakening region is extremely hard, since it is impossible to know the exact strategy that the inverter uses. The implemented method in this project gives a reasonably accurate estimation despite this fact.
Cylinder Friction

Although the crane has been around for quite some time, the friction calculated from this measurement is higher than normal. The measurements by them selves are made at an early stage of the project with little knowledge of the load sensing system that is used. For instance, the sensors are not calibrated prior to the measurement, and the valve block tend to operate with high pressure in the cylinder chambers. This may contribute to the large friction forces. The results are used in the energy analysis even though they are high, which leads to a conservative result of the overall system efficiency. However, since the friction component determined higher than normal is the viscous friction, it primary affects the result with increasing cylinder speeds. The coulomb friction on the other hand, is still present at lower speeds and stand still and is not specifically compared or determined to be valid.

8.2 Transfer Functions

The transfer functions are linear approximations of the real system and can only be taken for an indication of the actual behaviour. One of the greatest nonlinearities being the cylinder friction which have been approximated crudely. The viscous friction constant is taken from [6] for principal reasons to demonstrate the full dynamics of the system. This is a conservative approach, as friction is a contributor to system damping. If using the very high measured viscous forces, it would dominate the transfer function and other dynamics would stay hidden. Despite its questionable accuracy the transfer function is (when derived) easy to handle and gives insight to how the component properties affects the overall system behaviour. This report shows that the valve for this particular system can be neglected. One shall bare in mind that the components of this system are not optimized against each others or the ultimate purpose of this project. While one of the transfer functions do not represent the system in this project specifically, it is still relevant to derive it. If the components are to be optimized against each others the transfer function representing that system might be the one that is discarded for this project. The chapter does not only derive the transfer functions for both cases, but a transfer function that can advise if the valve is negligible or not. For a small valve, the combined system and the flow considered system have the same behaviour. For a large valve with low flow resistance, as in this case, the opposite is true. If attention is given to this subject, the frequency response of the transfer functions might appear totally different and give other results. The transfer function is definitely a tool with relevant insights to consider when performing the task of assembling an optimized EHCC, designing its regulator, evaluate signal feedback and much more. The transfer function and the physical crane are not compared to each others. This is unfortunate but not without reason. The inverter responds to a speed reference, and internal PI-regulators translates this into a torque reference. Other features built into the inverter manipulates the speed reference as well. It is therefore complicated to compare the EHCC to the transfer function in a fair way.
8.3 System Performance on the Loader Crane

By comparing the system response performance between the pump controlled and the load sensing system there is a significant difference in response time. A shunt type load sensing system used in this experiment has specifically good response since the fixed displacement pump is continuously keeping the hydraulic pressure at the input port of the valve. This gives immediate flow when the control valve opens, and might not be the most fair comparison since load sensing systems with variable displacement pumps has worse response time.

Since the pump controlled system used as proposed concept is a single acting system, the cylinder must be pushed down in order to lower the crane. Therefore, a response test with an unloaded crane is specifically interesting since the load weight is at its lowest point. In the second lowering motion during the lifting cycle performed in Figure 7.4, the reference signal is set to maximum lowering speed. The interesting thing is that the electrical motor consumes energy in order to help the pump to accelerate in the reversed direction since the crane weight is not enough to provide the requested pump speed.

A common problem between the lifting and lowering motion of the crane is the remarkable slow deceleration of almost 2 seconds. As shown in the experiment with active counter acting, the time is decreased with 50%. Further work can be done in order to decrease it even more, for example by configuring the speed controller but also by component sizing. However, a safety issue occurs when the emergency button is triggered, since the check valve within the on/off-valve is letting the crane to continue moving upwards even if the valve is closed.

In order to reach the maximum speed of the pump controlled system at 0.05 m/s, the motor is limited to 1200 rpm which corresponds to half the maximum rated motor speed. This means that the pump or the electrical motor can be replaced with a smaller one. However, in order to gain more power to the system, a smaller pump would be a better choice.

A problem that is introduced and examined when the on/off-valve is used as a load holding valve, is the need of pressure equalizing between pump and valve before lowering the crane. A simple solution is proposed and tested with a smaller cylinder pressure drop as result. However, the ideal solution would completely eliminate the pressure drop, but since the on/off-valve in this case is programmed to open at a motor speed of 40 rpm in both directions the valve opens just before the pressure is completely equalized. Even though the pressure drops are relatively big compared to the cylinder pressure, any significant movement on the cylinder position sensor is not recognized and therefore not showed in the corresponding graph. However, a very small change in boom cylinder position, quickly propagates to a larger movement in the crane tip which further can be experienced as difficult to predict and control for the operator.

Another problem seen in the pressure equalizing test, is the inability of the inverter to handle a sudden disturbance such as the cylinder pressure when the valve opens at low reference speeds. This problem can be solved with a more aggressive speed controller within the inverter.
8.4 Energy and Efficiency Analysis

Results from the manually controlled lifting cycles illustrates the problem of performing a repetitive motion with a physical crane in a safe way, since the operating height is limited and heavy loads cause risks. The deviations between the cycles further means that it is difficult to compare regenerating efficiencies from different cycles with each other in a precise manner. However, the purpose with this thesis is not to completely examine in what way different loads affect the regenerated amount of energy, but instead examine the regeneration behaviour in general.

The performed lifting cycles shows the difference between a loaded and an unloaded crane, with the result that regeneration efficiency increases with load. As mentioned in Section 7.2.1 the specific weight of the crane and the load is unknown. In order to get a more precise result the crane automatically needs to follow a specific speed reference according to a predefined lifting cycle, and the lifted loads needs to be determined. However, before relying on the crane the problem with slow deceleration within the inverter speed control must be solved.

Even though the load weight is unknown and the crane is manually operated, the regeneration capability during different lowering speeds shows that there is an optimum speed to achieve in order to maximize the amount of regenerated energy. The resolution of different lowering speeds in the test is, however, not very precise and can be further developed by defining an automated lowering cycle with smaller steps between the speeds. Furthermore, it is interesting to study how specific load weights affects the optimum lowering speed. By gaining more knowledge of the regenerating capability, the control system can limit the system behaviour in order to minimize the total energy consumption.

One of the main advantages with a pump controlled system is the possibility of saving energy while not moving the cylinder. However, the results from the lifting cycles reveals that part of the load pressure is captured between the pump and on/off-valve, which is causing the motor to consume energy even during stand still. Even if the consumed power during the stand still is fairly low, the load weight is also much lower than maximum rated load of the crane. Combined with the system in operation for a longer time, the idle losses will grow to a more significant energy loss. In order to solve this problem the pump needs to actively equalize the pressure by reversed rotation of the motor, or a valve needs to be introduced to bleed off captured pressure between the pump and the valve.

Even though any experimental data can not be used in an efficiency comparison between the pump controlled system and the reference system, there are a few obvious differences. Since the pump controlled system compared to a conventional load sensing system is virtually free from throttling losses from the main control valve and a proper load holding valve, there is an advantage regarding system efficiency. Furthermore, the interference between different functions and the loss contribution from simultaneous movements are directly avoided by the implementation of a pump controlled system on each cylinder or actuator.
8.5 Possible System Improvements

Some results but also general knowledge obtained during the project can be summarized in a few possible ways of improving the system. An alternative hydraulic setup, component sizing and variable displacement machine are the improvements discussed below.

8.5.1 Alternative Setup

By the results from this thesis a suggestion for how the pump controlled setup can be improved is proposed according to Figure 8.1. Due to the problem with slow deceleration of the pump and electrical motor, a blocked on/off-valve without any check valve in the upstream direction is suggested for safety reasons. Otherwise the cylinder continue to extend after the lever is released even if the emergency stop is triggered during a lifting motion, which may cause damage to the surroundings.

Furthermore, a check valve is introduced between tank and high pressure side, in order to allow the pump to rotate in reverse direction without the risk of cavitation. This further enables the possibility for the pump to remove the captured pressure between pump and on/off-valve during load holding conditions.

![Figure 8.1: A suggested system improvement where the on/off-valve is switched to a blocked one for safety reasons. A check valve is introduced which allows the pump to rotate in reversed direction without the risk of cavitation.](image)

8.5.2 Component Sizing

The pump and motor are dimensioned to provide power to a complete crane and not just one function. These components are in other words oversized for only one cylinder in terms of power. The motor is demonstrating lack of sufficient torque at low speeds. It can only achieve a corresponding 100 bar at 100 rpm. A heavy load is preferably manoeuvred at low speeds, and therefore is this operating area important in a drive-ability point of view. It is obvious but still worth mentioning that since high flow is not first priority, the pump displacement can be decreased
significantly. With decreased pump size the pressure capabilities increases linearly. This comes at the expense of reduced energy regeneration efficiency, as the motor would operate at a higher speed. A lot can be gained in terms of performance, energy consumption and regeneration by harmonizing the parameters of the components to each others. Decreased displacement automatically increases angular velocity, and brings the operating point to a higher efficiency for both the pump and motor. Two papers where different EHA’s are optimized for aircraft applications mentions a process that is suitable for dimensioning a system of the character in this thesis, these are paper III and V in [2].

8.5.3 Variable Displacement Machine

The system is currently equipped with a fixed displacement pump. It is easy to control and above all, less expensive than a variable one. If the pump is replaced with a variable displacement pump other possibilities opens up from a range of perspectives. With the ability of varying both the pump speed and displacement the operating point can be optimized in terms of system efficiency, regeneration efficiency, or power. Since the displacement can be varied, less torque from the motor is required to achieve the same pressure levels, if the displacement is reduced. This implies that the electro-hydraulic system becomes stronger, and both the pump and motor can be optimized in terms of performance with smaller parts. The potential issue with providing a static torque can be reduced by adjusting the torque on the motor with the pump displacement. This may also open up for an easy controllable fast response system, especially with high static loads and low speed drive for the pump. [2] mentions dynamic performance being better with a variable displacement machine rather than fixed displacement. The downside is more heat dissipation which can lead to more components and therefore more losses.
Chapter 9

Conclusions

This chapter answers the defined research questions of this thesis, and gives a suggestion to how this project can be continued into the future.

9.1 Answers to the Research Questions

− Is the system controllable given the presented concept? How well does the system perform, and what type of issues are introduced?

The system is closed loop stable and the system gain is low according to the transfer function. The physical tests confirms that the proposed configuration during the defined conditions is controllable. Issues related to system performance is the requirement of pressure equalizing for smooth operation; a safety function must be implemented to prevent the pump from cavitation under certain conditions; and the motor have limited performance under heavy load and low speeds.

− How well does the mathematical representation of the induction motor correspond to physical measurements?

Even though the mathematical representation of the induction motor is quite complicated, relatively accurate results are achievable with limited means. There are some uncertainties regarding the accuracy in the field weakening region as this method can look very differently inside the inverter, but the simple method implemented in this thesis gives decent results.

− How could energy regeneration be introduced to the system, and what improvements in an energy consumption perspective can be attained?

The single acting pump controlled system enables energy regeneration as the potential energy in the crane load can be efficiently regenerated, if optimum lowering speed is taken into account. Furthermore, the absence of major throttling losses contributes to a high lifting efficiency.
9.2 Future Work

The end of this project marks the beginning of another. Based on the gathered information in thesis there are a couple of aspects that are recommended to look into as the technology development advances to the next step.

→ The motor performance is as already stated several times poor at low speed and high load. Further studies in this matter should be aimed on investigating the possible options to solve this issue. A suggestion is to combine the induction motor with a variable displacement pump.

→ The components of the EHCC can be streamlined to its purpose. For example the pump and motor are overdimensioned for running only one cylinder. This task is preferably done with consultation of the transfer function chapter in this thesis, as it can help to identify potential solutions, and what parameter combinations to avoid.

→ Control issues are solved using basic algorithms. The control system level can be refined to solve many of the encountered issues. Including a closer look at the inverter from a control point of view, as its full potential is not revealed in this project. For example, the reference signal is currently speed, but it is possible to use current as reference instead. By using this reference the response is expected to increase, and the liberty of constructing a streamlined speed regulator.

→ Only the simpler EHCC is studied in this report. Many of the aspects can also be applied to the double acting system but also issues specifically related to that system. A functioning double-acting EHCC is required to apply this technology to all cylinders of the loader crane.

→ From an efficiency point of view, a hot topic is to study the simultaneous driving losses when having multiple EHCC’s, and compare them to a well known standard load sensing system.
Bibliography


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