Backlash in slew bearings

The advantages of an all-electric drive system

Tiril Bjønnes

Nina Moe

Supervisors

Stein Bergsmark

Jørn Salthammer

Stig Svenøy

Svein Kåre Sørgaard

This Master's Thesis is carried out as a part of the education at the University of Agder and is therefore approved as a part of this education. However, this does not imply that the University answers for the methods that are used or the conclusions that are drawn.

University of Agder, 2012

Faculty of Engineering and Science

Department of Mechatronics

CONFIDENTIAL

Exempted from publication until 1st of June 2017

Abstract

The presence of backlash is a widespread problem in all rotating mechanical constructions. A significant backlash in for example the slew bearing of a crane will propagate and could result in large deviations from the requested position. The shipping and offshore industries constantly strive to improve their systems and find solutions to reduce the backlash effect. Current solutions for the motor drives of slew bearings are usually hydraulic. The only control on the amount of force exerted on the slew bearings is valves that are either open or closed. This leads to large inaccuracies and it is hard to exercise good control of such systems.

The intention of this work was to design a new and advanced electric drive system for the slew bearing. Functional control systems are easier to implement with electrical drives and will make the rotation of the slew bearing more precise and without oscillations. The result of the solution shows that if the backlash is reduced the crane has a more accurate and smoother rotation. Such an improvement will bring a significant competitive advantage to our client, Huse Engineering.

Through close collaboration with employees at Huse Engineering, the project group was allowed to gain valuable knowledge of and insights into the drawbacks of the existing hydraulic systems. The challenge of achieving a well-controlled and accurate system, the risk of pollution and need of maintenance are some of the drawbacks of the current system. To mitigate the disadvantages, a thorough study was performed and the final solution was defined through qualitative research, interviews, generic work and different methods for development and design.

A master / slave configuration is proposed as the solution for this all-electric drive system. The operational idea of the master / slave drive is that torque of a certain magnitude is applied in the opposite direction of the main driving direction. This provides a braking effect and improves the positioning and stiffness of the mechanical system. The implementation of motor pairs counteracting each other in given situations was shown to result in a reduced backlash effect on the slew bearing. This would provide a more accurate rotation and a steady load with minor deviations from the required position.

The conclusion to this project is that by altering the electric motor drive of the slew bearing, the backlash is reduced. Replacing the hydraulic motor drive with an all-electric drive allows advanced control of the system and accommodates the possibility of a master / slave operation of the slew bearing drive.

There is not found any previous research available for comparison and verification of the new design, which creates the need for further development and testing of the system. The limited previous research presented new challenges, such as finding a functional design, and the

project group had to study multiple new concepts before a solution could be defined. This report thus documents a concept at the idea and design stage. Significant work remains before implementing the selected solution on an actual crane. Testing, simulations and tuning of parameters should be performed to complete the proposed, innovative solution design. Nevertheless, it is believed that the recommended solution will operate as intended.

Preface

This report is the result of the course MAS500, Master’s Thesis, of the master’s program in Mechatronics at the University of Agder. The report is prepared during the spring of 2012.

Wayne Gretzky, often referred to as the greatest ice hockey player ever, once said that *“a good hockey player plays where the puck is. A great hockey player plays where the puck is going to be”.*

This quote really grasps the purpose of this Master’s thesis. The project was performed on behalf of Huse Engineering, a global supplier of marine lifting and handling equipment. In terms of this thesis, the company wanted to play where the puck is going to be. By examining new ideas and techniques the company constantly aims for an improved market share, innovative solutions and competitive advantages. This was the driver for Huse Engineering when they teamed up with us for the final project of our master degrees.

The crane that is the subject of this project is currently in the design phase, but will be delivered to the shipping- and offshore industries this autumn. All technical specifications and input to the project are therefore confidential. Confidentiality also applies to the outcome. Confidentiality may place restrictions on the handling of problems, both present and future, objectives and results, as well as the degree of details.

During the preparation of this report and as our technical knowledge grew, we realized that the scope of the chosen topic would not result in a definitive solution. It is a wide concept under constant development, and this made the resulting conclusion slightly different than first anticipated. No physical prototype has been developed in the course of this Master’s thesis, however we have reached a conceptual design that we believe is an excellent and innovative solution to this thesis’ problem.

We would like to thank our contacts at Huse Engineering and our supervisor, Stein Bergsmark at the University of Agder, for feedback, inspiration and valuable knowledge. We will also sincerely thank everyone else we have encountered during this project. Without your guidance and support we would not have been able to complete our journey.

Thank you.

Tiril Bjønnes and Nina Moe

Grimstad

May 2012

Contents

List of Equations

List of Figures

List of Tables

Chapter 1

Introduction

This chapter focuses on the scope of this project and outlines the background and motivation for this assignment. A clearly defined problem definition consisting of one main goal and several objectives are presented and will help to facilitate the work. A thorough literature review is also an essential part of this project and this chapter therefore mentions how a qualitative research is approached. As this project case is based on an actual crane intended for an offshore location, there are special safety and standard specifications to consider. These are briefly described, along with the planned layout of this report.

1.1 The shipping and Offshore Industry

Ever since the historic discovery of the Ekofisk field in 1969, Norway has been a major oil- and gas nation with many contributors in the shipping and offshore industry. IP Huse (IP) is one of the companies involved in the growing field and as early as 1970 the first winch for anchor handling was delivered. Today, IP produces solid winches for deepwater operations.

The company is world leading in design and manufacturing of winches for special vessels and floating offshore installations. Hence, the founder of IP is also globally known as “The Winch Builder”. IP is currently a part of the British Rolls-Royce group and the success regarding the winches eventually led Rolls-Royce (R-R) wanting to expand IP’s market area. Huse Engineering AS (HE) was thus formed as a subsidiary company of IP and was to develop and produce heavy lift cranes for marine use.

HE produces lifting and handling equipment for the global shipping and offshore industry. All products designed and manufactured by HE is owned and trademarked by R-R Marine. Consequently, all marketing, sales, service and follow-ups takes place in close collaboration with the British group.

The shipping and offshore industry is in continuous development and for a company to stay competitive it must focus on research and development. HE keeps a close dialogue with customers and the feedback is used to continually develop the products. The issues regarding backlash in the slew bearing is obtained through such dialogues. The aim for this project is to improve the precision of the slewing relative to the current system. It is decided to replace the hydraulic motor drive of the slewing with an all-electric drive system. The hypothesis is that this makes it possible to reduce the backlash in the slewing and improves the accuracy of the system.

Currently HE is developing a 50T active heave compensated (AHC) crane and it is decided to follow the development process of this crane. The possibilities and advantages by changing the drive system of the slewing will be explored. The AHC crane is designed to handle subsea modules. By constant adapting to the wave motion, the AHC enables the crane to maintain precision and load stability in unstable water.

1.2 Background and Motivation

In mechanical engineering, backlash is defined by the Oxford English Dictionary, 2nd ed. as the striking back of connected wheels in a piece of mechanism when applying pressure. During crane operations the effect of backlash is an undesirable characteristic which should be controlled. A backlash gap of a few millimeters in the slew bearing can increase throughout the system and cause considerable effect on the operation.

As preparation to reach the main goal, it is carried out a thorough investigation of the existing system. The current slewing system is operated by the hydraulic motor F12-90 from Parker. These specific motors, the F12-series from Parker, have few moving parts and are therefore very reliable. In general, hydraulic motor has a very compact design, is lightweight and has great efficiency.

The motor is part of the hydraulic system and pressurizes the hydraulic fluid to control the crane operations. A power transmission is generated from the motor, through the hydraulic pipes and to the swing gears. The swing gears will then rotate the slew ring. By controlling the pressure it is possible to regulate the torque provided by the swing gears. The 50T AHC is equipped with four swing gears. The reasons to implement four equal gears are to obtain necessary force and ensure the precision of the slewing and a through that reduction of backlash. This principal will reduce the effect of backlash and is further explained in Section 4.2.

For a company to remain competitive it is necessary to constantly develop and improve its products. The main goal for this Master's thesis is to improve the slewing motion and develop a good system design for backlash reduction. This is performed by replacing the hydraulic steering of the slew bearing with an electric motor drive.

The group's motivation to work within this field of technology is the innovating culture, the international environment and great expertise within a broad field of knowledge.

The study of the existing system forms the basis for the main goals and objectives for the project. Goals and objectives are presented in Section 1.3.

1.3 Problem definition

Regardless of the motor drive used for the crane operation, backlash in one or more parts of the mechanical drive system is a common problem. The resulting error of the backlash effect is transmitted through each joint of the crane construction and the deviation between actual and desired load position can get significant.

For the crane used as case-study a lot of time and resources are spent on the development of a top-notch AHC system. The system compensates for the vessels relative motions operating at sea with loads in water or near the seabed. This ensures a steady load and minimizes the loads vertical displacement. With such effort used on minimized vertical deviations, it is in R-R' and HE's best interest to also reduce the horizontal displacement. By reducing the backlash effect in the slew bearing, dislocations from the position of the tip point of the crane is minimal and the load is more stable.

For the crane handled in this project, the slew bearing has a hydraulic motor drive. Until recently hydraulic drives were the preferred motor drive for most crane operations, including the slewing. However, development in the area over the past few years has resulted in cheaper and more efficient electrical solutions. This has increased the focus on using electric motor drives. It is believed that a better precision of the slewing is gained by using electric motor and motor drive. An electric motor drive gives the possibility of easier control and management of the drive system and the industry assumes that by installing an all-electric system a smoother rotation of the crane is provided.

As mentioned in Section 1.1 companies within the shipping and offshore industry have always aimed for further improvement of products in close consultation with customers. The continuous feedback indicates that clients want marine solutions with minimized life cycle costs and higher precision. It is through such feedback the request for reduced backlash is expressed. Considering that backlash is assumed to be the main problem of mechanical rotations, this request is justified. Consequently, several participants in the industry have made obligations to achieve this.

This project attempts to develop an improved solution for backlash reduction in the slew bearing. In addition, it is well known that hydraulic drive systems of slew bearings have higher maintenance costs than electric drives. Consequently, reduced costs are an additional motivation for swapping the drives. The mentioned issues form the foundation for searching for a solution that includes an electric motor drive system.

Based on how backlash affects precision, and given the recognition from the industry regarding use of electric motor drives for offshore cranes, the following hypothesis is defined;

The backlash will be reduced by a transition from hydraulic to electric motor drive of the slew bearing of a crane.

This is the research hypothesis that is verified or falsified throughout this Master's thesis. During the project the stated hypothesis is addressed through a conceptual system design to see if reduced backlash in the slew bearing and improved accuracy of the slew is achieved when replacing the hydraulic drive with an all-electric system.

The majority of conventional systems today are based on hydraulic operations of the slewing. It will be quite challenging to implement an electric drive system and provide a backlash-reduced slew since most available knowledge and technology are based on hydraulic drives.

Such problems need to be addressed in order to solve the hypothesis and bring this project to a conclusion.

1.3.1 Main Goal

In order to investigate the hypothesis, the main goal of the Master's project is defined as follows;

Design and evaluate an electric drive system to verify or falsify the hypothesis.

To achieve the main goal it is essential to develop a satisfactory solution that is applicable and transferable to other crane projects. This means continuous generic work to ensure scalability and portability. The client of this project is HE, which delivers the crane to customer R-R in late 2012. HE supports this project with all necessary resources. This ensures a thorough study and the development of an innovative solution that will, if the hypothesis is verified, provide a competitive advantage. The solution will be a result of a complete background study, field studies, simulations and calculations. In all, an iterative development process by solving the objectives and investigating the hypothesis will lead to completion of the main goal.

1.3.2 Objectives

From the main goal, several objectives are derived. The objectives are defined as complementary parts of the main goal and the execution and accomplishment of these objectives are essential for a successful final result.

OBJ.1 **Study the existing system**
Present the system mechanically and functionally. Before developing a new and improved system, this study should identify the problems of the current system, as well as substantiate the hypothesis.

OBJ.2 **Select, implement and optimize the use of a suitable electric motor**
Select a suitable electric motor and electric drive. The motor drive must be integrated in the system and be subject to a number of simulations to ensure

improved and accurate operations. By means of simple control, the backlash in the slewing should be attempted reduced to a minimum.

OBJ.3 **Present key performance figures**
Present the key figures for the new solution. This includes the power and torque requirements at specified loading and operational mode in a concise manner. In addition, separately consider the topic of overspeed. The handling of overspeed is different for the various motor drives, and overspeed should furthermore be part of the assessment of safety and reliability.

OBJ.4 **Introduce functional requirements**
Develop functional requirements for the new system. This project should also result in a variety of mechanical descriptions, technical specifications and functional instructions of the new system design. A simple sketch of the control system should be proposed.

OBJ.5 **Develop technical specifications**
Develop system specifications for the new design. The new system must be able to withstand specified forces, safely lifting given loads and function in different conditions.

OBJ.6 **Comparison of the new and improved system with the conventional**
Compare the new system to the conventional in order to determine if the main goal is reached. The discussion of why an all-electric system is better and how it differs from the ordinary hydraulic system can only be literary. As the actual crane is being developed parallel to this thesis, comparable simulation results therefore do not exist.

OBJ.7 **Consider safety and reliability**
Perform a basic analysis and do succinct considerations of the safety and reliability of the new system design. In lack of possibilities of testing of the new product solution, this objective will only be handled briefly and concisely.

If time allows

If time allows, additional objectives will be solved. The success of the project is not dependent on execution of these objectives, but a full investigation can add greater reliability and scientific depth to the result. The objectives are prioritized according to the given order.

OBJ. 8 **Further improvements**
Initiate evaluations of further improvement of the system by selecting a different electric motor. Consider what alternatives that might be suitable.

OBJ.9 **Assess the existing mechanical dimensioning and selection of components**
HE has already selected appropriate gears for the current system. Perform necessary assessments and state whether or not this is the optimum gear with respect to a system that utilizes an electric motor.

1.3.3 Design Criteria

The objectives in 1.3.2 specify the line of work the assignment complies with. The aim is to address the hypothesis, as well as an improved design for the slew bearing. As this Master's project only involves the slew bearing of the crane, it will thus be necessary to establish design criteria to moderate the proposed solution. The design criteria are essential to indicate how the solution can be created, as well as the consideration of influences that may affect the solution method and final outcome.

- **Scalability:** Scalability is important to this project and therefore imposes requirements on the solution approach. Generic transferability must be maintained to ensure that the motor size and motor drive can be scaled up or down. The system must be designed for easy replacement of the electric drive without having to make major alterations. The best way to ensure scalability is to approach the solution by utilizing modular design.
- **Portability:** Portability is the ability to make use of the proposed solution, if the hypothesis is verified, in other cases than the specified case. The system solution should be transferable so it can easily be adapted into other crane projects. This limitation must also be complied when motor and drive suppliers are considered.
- **Suppliers:** The client prefers previously acquaintance and well known suppliers. Hence, it will only be focused on the supplier proposals given by HE when selecting crane components. The requirements for scalability and portability must also be fulfilled when selecting supplier. A manufacturer will supply this crane with a given

motor and drive according to specified conditions. In order to maintain scalability and portability for the solution, the supplier should also be able to provide other versions of the system design if any of the outer conditions or applications change.

1.3.4 Key Assumptions

With the objectives of the project established, assumptions and limitations are stated in order to keep the work at desired path and restrict the scope of the suggested solution. The given time frame yields that there is not enough time to evaluate all problems or results that may occur, thus the project will at all times meet the following assumptions;

- **Components:** It is assumed that all original chosen components are the best choice, except if doing considerations according to OBJ.8 or 9. In this case, the limitation regarding suppliers yields equally for gears as for motors.
- **Crane base and environment:** The floating vessel the crane is mounted on is assumed to be located at steady sea and can be treated as a stationary platform. Due to AHC the vertical positional deviation is considered minimal and other climate effects on the system in horizontal direction is also neglected. This means that there are assumed to be no external influences on the system. As a note, an actual system has to face challenging variations of the environment. For instance, temperature variations, high salinity, variations in voltage and frequency, motions of the vessel and vibrations in transit (crane is parked).
- **Safety and usability:** Assumptions are made that the existing system is already designed and developed the best way to assure safety and usability. Hence, aspects related to this will not be taken further into account.

1.3.5 Key Limitations

The examination of the hypothesis of this project can lead to a large amount of work and it is therefore just as important to define a set of key limitations. Continued work with the project may lead to new ideas and other results than predicted. This can cause an expansion of the project and/or that the execution of the task heads in a different direction than anticipated. The following list therefore limits the outcome;

- **Scope of work:** This project is to result in a conceptual design, not a working prototype.

- **Target:** The project is limited to only contain the study and improvement of the operation of the slew bearing. All other joints, components and parts of the crane are left out.
- **Simulations:** In consultation with HE it is decided to restrict the simulations. Simulations of certain load and operating conditions are restricted to the following;
    - Operations of crane at mentioned working conditions with load at a lifting radius of 10 m relative to the origin of the slew bearing.
- **Calculations:** Calculations of forces and different cases of loading are comprehensive and time-consuming work and are only identified in such matter found relevant and sufficient. With regards to forces, it is assumed that new calculations are not required as long as the new system design remains within the original specifications.
- **Input data:** The available 3D model, as well as technical sketches and data, is believed to be correct and reliable. In the continued work this limits the design and forms the basis of the simulations and results.
- **Control system:** It is decided to limit the part concerning the control system of the motor drive. Only a rough sketch of the control system for the electric drive is prepared and provided. A detailed sketch of this system will not be an outcome of the project. If the new solution is to be implemented in future work, the control system must be expanded and developed in depth, it will not be discussed any further in this project.

Regardless of these assumptions and limitations, it is believed that the results will be sufficiently accurate and generic transferable to other similar cases.

1.4 Literature Review

A wide range of research areas and sources of information are relevant for this project. To obtain an overall understanding of the process the Master’s thesis is a part of, several disciplines has to be considered. The main parts are the area of marine structures, the area of lifting and handling equipment for marine and offshore industry, hydraulic systems, electronic steering and PLS and simulation and calculation programming. Several research articles and literature of the latest news, work and trends, is searched for. This includes information of the technology itself and product development within the specified technology. By understanding the development process, the work is carried out in a more easy and satisfactory manner. The study of research articles is believed to give the project group insight and detailed knowledge of the subject, in addition to a deeper understanding of the system being designed. Additional literature gives creative inputs on working strategies and how problems can be addressed. It is important to be familiar with technologies and trends within the given field and obtain information of earlier research. Some of the prior research most used as reference and inspiration for this thesis are as listed:

- *Controlling mechanical systems with backlash – a survey*, by Nordin and Gutman (2002)
  The survey reveals some of the main practices of backlash compensation and formed the basis for the first ideas of backlash modeling in this thesis.
- *Dual Motor Control for Backlash Reduction*, by Schiffer (2009)
  This Master's thesis deals with different nonlinear approaches for backlash reduction and also introduces the principles of three-mass systems.
- *Motion Control of Systems with Backlash*, by Bruns, Diepstraten, Schuurbiers and Wouters (2006)
  In this report several methods to implement backlash control in the drive system is proposed.

These articles are also listed in the bibliography at the end of this report. Other results and references of the literature review are used in later chapters. If data, results and other work of external sources are used within the scope of this project, it is carefully referred to. Chapter 3 includes detailed information of literature concerning these topics.

1.5 Standards and Specifications

Standards for certification contain information, principles and acceptance criteria related to products and installations. When developing cranes, these have to be accounted for during the working process. Several Norwegian (NS), European (EN) and international (ISO) standards of certifications specifies the requirements in most industrial sectors. Within the offshore and shipping industry, there are also extraordinary criteria in terms of safety. For further information of standards and specifications, see Section 3.6.

1.6 Approach

The engineering paradigm is a working strategy often used during product development to ensure an efficient and clear working process. There are several possibilities for adapting to the paradigm. For this project the prototyping model is chosen. One of the main principles of this model is to have an open working process. This means that the project requirements are being developed through the modeling and definition process. By close contact with the client the developer gets feedback and useful information during the developing operation. It also provides the client the possibility to specify more detailed requirements in an early stage. The model has four main stages. These are:

- Requirement definition
- Testing and user review
- Simulation and calculations
- Specifications and early versions

The model provides a dynamic developing process where dialog and client feedback influences the next step. The steps are further described in Section 2.2 [1][2][3].

1.7 Report Outline

The second chapter of this report concerns the methods used for research and how to best develop and obtain the requested result. The development process is based on a modular, prototyping approach. This means that the system requirements are being developed during the modeling and definition process. In addition, the final system design consists of simple blocks that are easily tested and altered. This ensures a good, generic solution.

The third chapter deals with the examined background theory. Hydraulic and electric motor drive systems are explained. Further, a description of control theory and the importance of functional feedback control for motor drives are given. A review of system components and the basic principles of backlash are introduced. The final system must also be in accordance with a number of standards and specifications. These are examined.

The fourth chapter gives a technical description of the crane and defines the current hydraulic slewing system. A brief overview of the proposed all-electric drive system is presented.

The fifth chapter is the solution chapter. It contains a description the development process and how the improved system is obtained. The operation and function of the proposed solution are also defined. The electric motor and drive system are implemented in the existing mechanical design, and a simple control scheme is proposed. The final system solution is simulated by means of SIMULINK.

The sixth chapter presents the results of the implementation and simulations. It also indicates if the results were as expected.

The seventh chapter is a discussion of the results found in chapter six. It includes a discussion concerning choices made when implementing the new all-electric system and how these affect the result. Any problem areas of the proposed solution are highlighted. It is also discussed topics of future work that has to be performed if the proposed solution is realized. Eventually, the discussion leads to a conclusion to the project hypothesis.

The eighth chapter presents the outcome of this project and concludes if the hypothesis is verified or falsified.

Chapter 2

Methods

The purpose of this chapter is to describe the methods and procedures used in this project. The topics are intended to simplify the work to achieve the goals and a credible result. The report's methodological approach is mainly based on qualitative research methods in combination with theoretical studies. As this project has a theoretical research part and a practical development part, the procedures applies for both. The first section reviews the research methods used to ensure secure collection and analysis of data. Further on, the method for development is specified. The project is to culminate in a system design for an actual crane project. The focus has been on implementing quality assurance and generic work in the methods used.

2.1 Research Methods

The main reason to perform research is to gain new knowledge within the topic of interest. Acquaintance with specific fields is essential for the project to be successful.

Research can be divided into two main categories: basic research and applied research [4]. Basic research is performed without any particular application in mind, whereas applied research poses the possibility of using the ideas in operational form. In this project a modified version of basic research is used. Research is performed on the generalities of the technology in interest, yet is still aimed for this specific application.

Several research methods are used for the study of existing solutions in the technologies of offshore cranes, motors and motor drives. The methods of qualitative research and, to some extent, quantitative research is used when collecting and analyzing theories, technical information and data. The obtained knowledge is contributing to the development of the new and improved system design.

2.1.1 Qualitative Research Methods

The qualitative research method is a scientific and investigative. Qualitative research has multiple focus points and through these it aims to understand given issues. The research method examines problems, recognizes certain circumstances and answers questions. By analyzing collected information, qualitative research attempts to answer 'why' instead of 'how'. When using the method the subject of the research is studied and interpreted in its natural setting [5][6].

The method of qualitative research is through the following steps producing comprehensive and useful information about the desired topic.

- Data collection
- Data analysis
- Possibly propose recommendations

These steps are followed throughout the survey of the existing technologies. Since the qualitative research method is chosen for the project, the opposing quantitative research method is not subject for severe discussions. Briefly, where the qualitative research method focuses more on articles, theories, interviews, information retrieval and case studies, the quantitative research approach is to a greater extent facilitating analysis of hard facts and numerical values. As the names indicate, the quantitative method scopes a wider range of specific numerical data, while the qualitative research methodology focuses on a narrower path with respect to fewer numbers and an objective, qualitative vision of few things.

When collecting information for the qualitative approach, the project team focused on obtaining data from sources considered reliable. It is of great importance that the references supplies accurate information, as the theories and obtained data forms the basis for the development of an actual product. Each source is therefore subject to an individually assessment of reliability, and technological sites, regulations, journals, manuals and textbooks are preferred as references. Examining topics directly related to the scope of this thesis has also been in focus.

Additionally, observations of the crane construction and operation are executed as a part of the research. Observations of the system in its natural operational modes, has given the team unique and practical understanding of how the slew bearing operates. This revealed shortcomings in the existing system. The observations proved to be highly valuable, as the shortcomings are exploited for improvement of the current system design. As part of the data collection and general research, interviews and meetings with HE is performed. Through requests, contact and exchange of information with suppliers, several relevant solutions and system components are examined. Regular meetings with external and internal mentors have supported the progress of the work, as well as securing that the proper solutions are selected.

The gathered qualitative data is careful analyzed in order to model and transform it to applicable information. The analysis highlights beneficial properties of certain technologies that are further utilized in this project. Consequently following the gathering and analysis of data, there are proposed simple and early-draft recommendations of how existing technologies can be implemented into a new idea.

2.2 Development Methods

The engineering paradigm is the most used development method when engineers are working with a task. A paradigm is defined by the Oxford English Dictionary as a *'pattern or model'*, and engineering paradigms are therefore used in technology to explain distinct approaches. The chosen development method is presented in the following Section.

2.2.1 Prototyping

To ensure an effective investigation of the hypothesis stated in Section 1.3, it is of great interest to incorporate a working strategy. A method often preferred during product development is the engineering paradigm. As paradigms in the art of engineering are constantly changing, it makes it practical to use in the development process for this product. There are several different engineering paradigms and these are as listed:

- Code and fix life circle model
- Waterfall model

- The prototyping model
- The spiral model
- Fourth generation techniques

For this project the prototyping model is chosen. The remaining options are not discussed any further.

Using the prototyping model the working process is not fixed from start, meaning that the requirements are developed during the modeling and definition process, as shown in Figure 2.1.

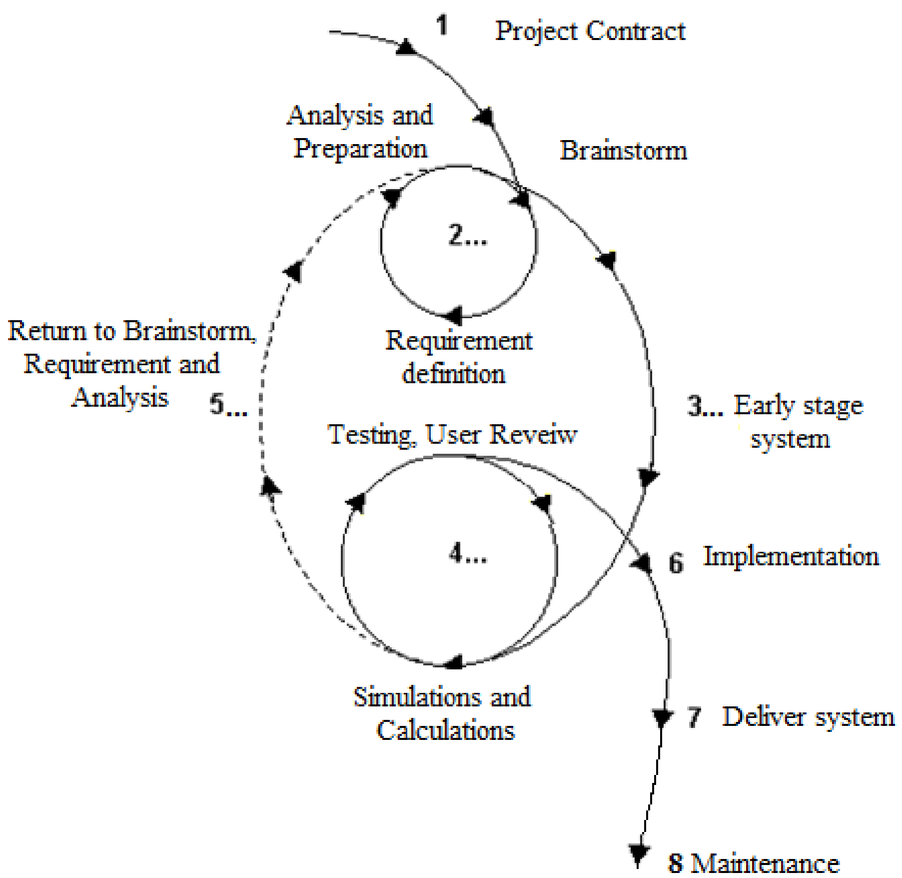

Figure 2-1 System Development using the Prototyping Model

The prototyping model consists primarily of four main stages;

Requirement definition

Specify the requirements as accurate as possible in an early stage of the working process. The stages 'Analysis and Preparation' and 'Brainstorming' are preparatory work in order to state the requirements. The requirements are divided into functional and non–functional. To get started with the development process and obtain a functional model, there are defined project modules. These modules are catalogued under functional requirements. The non-functional requirements contain topics like extensibility, runtime, etc.

Specifications and early versions

Define the necessary specifications to make the early versions function as intended.

Simulation and calculations

After identifying specifications and requirements, the calculation and simulation procedure can start. Creating an overview of the intended working progress, an effective developing process is obtained. If this is ensured, every developer working with the assignment gets an overall understanding of the process.

Testing and user review

The user review is a useful tool and gives valuable feedback. A provisional requirement definition or limited product version is delivered to the client early in the development process. The client is given the possibility to understand the characteristics of the final product. The approach gives the client the possibility to determine and specify detailed user requirements in an early state and is especially useful with unclear objectives.

The finalized product will now be implemented and delivered to the client, as given by step 6 and 7 in Figure 2.1. For this specific project the "Testing and user review" is the last stage. After final testing the hypothesis is either verified or falsified and the project comes to a conclusion.

In industries like the shipping and offshore sector where the user requirements and other factors of production are constantly changing, a working strategy like the prototyping model is important. The approach allows developers to quickly adjust and adapt to changes while maintaining the focus on the overall objectives of the project [1][2].

2.3 Quality Assurance

There is a variety of definitions of the term quality. Different dictionaries define quality in different matters. However, the main essence can be expressed with the phrase *fitness for purpose*. This means that quality can be interpreted to mean *meeting customer's expectation or exceeding the customer expectations and conform to all specifications*. It is the customer's definition of quality that matters and in this case it is client HE and customer R-R that set the quality requirements for the system design and development approach [7].

Conversations with HE revealed that the project team works according to all applicable standards when considering the technical aspects of a project, and it is therefore those standards that set the requirements for the quality within these areas. In terms of the development and design process, HE is working to sustain ISO 9001, as well as R-R's own quality assurance program. The program, named Derby Gated Reviews, follows 21 steps that deal with the entire process from idea to finished product. Through quality assurance of the work process, the 21-steps program ensures a comprehensive, quality product. Simply put, in addition to ISO 9001 and other standards, HE is using a simplified model of the Derby Gated Reviews at daily basis.

In short, the standard ISO 9001:2008 defines requirements for a quality management system that is applicable to all kinds of organizations. The international standard is a particular useful tool in business assurance and is used to ensure quality, prevent mistakes from recurring and identify errors in advance. Quality assurance systems, management responsibility, resource management, product sales and measurement, analysis and improvement are key elements of the standard of certification [8].

It is important that the quality of the product satisfies the requirements, as the final system design might be commercialized for the next crane development. Accordingly, the actions of quality control and quality assurance needs to be implemented in this project. Quality control and quality assurance ensure that the product and project work maintains the wanted quality and arrives at a satisfactory completion of the main goal and objectives.

Quality control (QC) is a system of activities routinely performed to measure, compare and control the quality of the object as it is being advanced. The routine of the QC system should provide consistent control to guarantee data integrity, correctness and completeness. Errors and anomalies relative to the plan must be identified and all activities related to the QC must be documented. Thus, QC focuses on the output of the process [9][10].

Quality assurance (QA) refers to planned activities of a quality system that is necessary to perform in order to meet the quality requirements. QA systems monitors, measures and compares the quality in the project management life cycle by evaluation of the process and procedures. QA prevents errors from occurring and therefore possesses a highly critical role in the success of the project [11][12].

Prior to the following method to assure the quality of the product and quality assurance throughout the development process, it is determined to make use of QA, and not QC. As the final product of this project might be reused by HE it is therefore essential that the deliverables contain no anomalies or errors. This is best prevented by QA.

Quality assurance is implemented in the project work from the beginning. This brings good communication between the team members, as well as an early understanding of problems and concerns related to the product.

Still, the most important element regarding quality assurance is to develop a good project quality plan and to continuously relate to this plan throughout the project phase. Various parts within such plans are systems and control for the following; responsibilities, management, design, testing, corrective actions, nonconformance, records, audits and training [13].

For quality assurance to be effective and functional within this project, the key quality elements to carry out are mainly analysis and corrective actions. The procedure followed to assure the quality of both the development process and the final product is first and foremost the execution of *good practice*. Through common sense QA is undertaken by the use of objective reviews of the product design. The latest version of the system design is under continuous evaluation for a constant strive for excellence. Constant review is carried out to identify and solve problems throughout the development process. In this way, improvements are made in advance and errors of the final product prevented. However, due to time constraints and limited access to resources for this Master's thesis, the quality assurance may not continue infinitely. The project work must arrive at a final product and conclusion, and the procedure of assuring the quality requirements must at some point come to an end.

2.4 Generic Work

Generic work within this project is important to sustain the ambition of the solution, if successful, to be adaptable to other crane projects. This is a request from HE, as it provides for a great competitive edge in the industry. The product development process and the product design should thus be consistent with the idea that parts or the entire product is usable in several different types of applications. For the solution to be generic transferable it is important that certain methods of design are followed. In this project it is chosen to maintain modularity within the design.

2.4.1 Modular Design

The approach of modular design, also termed modularity in design, exploits the idea of subdividing complex systems into smaller components. The rotational mechanical device can be organized into a set of several modules, thereby the term modular design. Each of the created modules is developed independently and then reassembled to drive the system functionalities, often in different combinations to each other [14].

In this project, the ideas of modular design are kept in mind when designing the all-electric drive system. For example, the frequency converter, the electro motor, the controller, the gears, and so on, is represented by separate modules. The design is in accordance with the structure indicated in Figure 2.2.

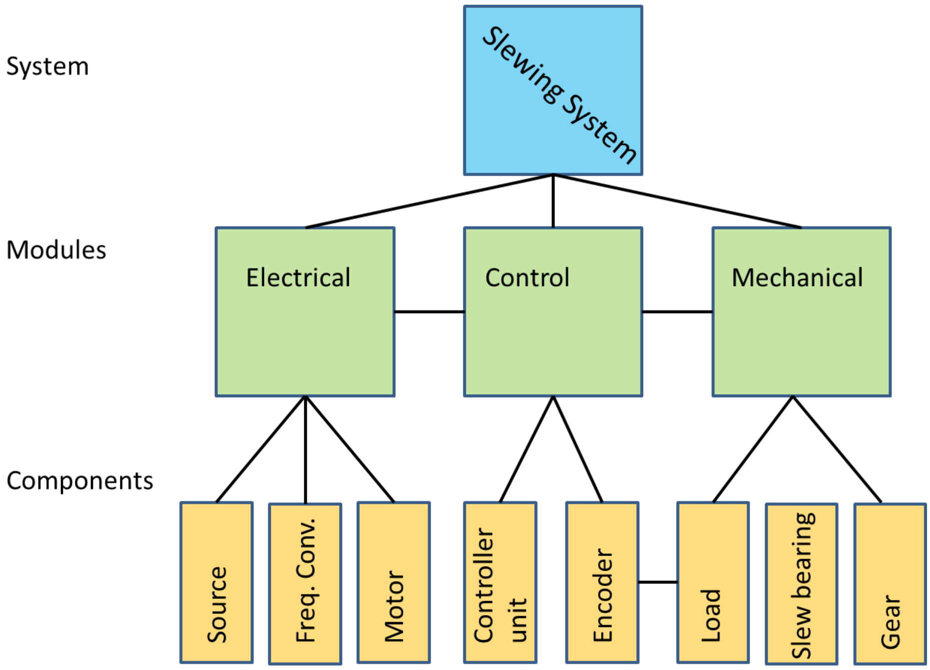

Figure 2-2 Modular Design Structure

Each element in the modular design has a simple interface ensuring the modules to be reusable, isolated and self-contained. Such interface helps verifying if the system works as intended, and it also provides an easier reuse of the components. Reuse reduces time and cost in the design phase of each project, as already designed modules can individually be replaced in new system designs. Each module should encapsulate information that is not necessary and/or available to the rest of the system. This means that each module and its input/output should be independent of each other, which eases the replacement of certain parts of the system. The benefit of modularity is thus how the decomposition yields enormous differences in how easily systems is implemented and modified [13][14].

To summarize, each module is designed subsequently to the following list:

- Clearly defined module structure
- Clearly defined module purpose
- Sufficiently abstract interface
- Sufficiently subdivided with respect to usability
- No replication of other modules functionality
- Isolation of aspects of the system that most likely will change

Through modular design the aim is to supply a solution that is both scalable and portable.

2.4.2 Scalability & Portability

As stated in Section 1.3.3, scalability and portability has to be within the scope of this project.

Scalability in the system is the ability of adjusting the dimension of the electric motor drive up or down as needed. Portability is maintained if the solution can be used in other cases than the case it is developed for.

Scalability in this specific project is secured by ensuring that the selected suppliers can deliver the chosen component with several different dimensions and properties. It is also taken into account to protect the possibility of scalability when choosing the motor and motor drive. This is particularly assessed when considering weight and size for the placement of the motor. The selection of motor drive and its components is also carefully executed. The choice and design of the motor drive should possess modularity. In that way individual parts can be replaced independently of each other.

The aspect of portability within this project work and solution design is partly achieved when selecting suppliers that contribute to scalability. If the suppliers are able to produce components in different sizes, it is implied that the supplier also can provide components with different qualities. This is important, as the component might have to accommodate other characteristics than the original component due to changing appliances or external influences. Changes in the area of application might be the scenario if it is requested to transfer the system design for this construction to another project. Thus, portability is equally important to achieve as scalability.

Hence, the topics of scalability and portability restrict the solution design. Nevertheless, it is assumed that the criteria for scalability and portability are complied with through modularity within the design. It is thus ensured that the final product can be changed, transferred and/or implemented in other projects, if desirable, by focusing on generic work and modular design.

2.5 Software & Modeling Tools

To arrive at a functional solution, a lot of work is required. In all, simulation, regulation and trial by errors are performed to achieve a good design. Completion of the main goal and objectives requires the use of good simulation tools. For this project it is determined to primarily use the computing programs Matlab / Simulink and/or SimulationX (SimX). The main reason for this is that HE uses these specific simulation tools for the design of the crane. By choosing Matlab / Simulink and SimX it is easier for HE to apply the results and solution from the study in later projects. The motivation and objective for HE is to use and install the solution on their next projects, if applicable. Then there is highly advantageous to base the simulations and calculations on equal programming tools.

The complete electrical and mechanical part is built in SIMULINK, by using different blocks available in the SIMULINK Library Browser. SimPower Systems, a subset in the library, is used when modeling the electric drive, whilst another subset, Simscape, is the basis for the mechanical modeling.

In terms of the moment of inertia applied in the mechanical part of the system, it is extracted from the 3D model of the crane. This information is retrieved by means of Autodesk Inventor, which is 3D CAD software for mechanical design. For the investigation of power, loads and impacts on the mechanical part of the system, SimX is used.

Chapter 3

Background Theory

This chapter introduces the underlying technology for heavy lift cranes, especially focusing on offshore cranes. To design efficient motor drives for such cranes and understand how to further improve the construction, relevant topics are examined. The aim is to investigate existing solutions, basic technology and standards for certification in order to decide how to implement the new idea with the well functional conventional one.

3.1 Essential System Components

The crane is made up of a number of key components. This section provides a brief review of the main rotary mechanical elements involved in the operation of the slew bearing.

3.1.1 Slew bearing

The slew bearing used for the 50T AHC crane is a large diameter slew bearing from Rothe Erde (RE) manufactured in Germany [15]. Slew bearings from RE are fundamental machine components absorbing all axial and radial forces subjected to the crane. Directly interpreted, slew means rotation without change of place, yielding that a slew bearing is a rotational rolling-element that supports slow turning of heavy horizontal loads. Slew bearings are made up of ball or cylindrical shaped bearings and can therefore accommodate forces in any direction and may also function with or without gears [16]. This specific crane is to perform under the presumption of four gears.

3.1.2 Gear

The swing gears used for the 50T AHC crane are important components in rotational mechanical devices and its motor drives. Gears are mechanical constructions that transmit radial forces. The gear unit is intended to transfer and adapt the torque from the motor by means of sprockets in different dimensions, diameter, number of teeth, etc. Machinery equipped with rotating gears provides a change in speed and torque, and the gear maintains accordance between the power provided by the motor and the speed and position of the mechanical element [17].

3.1.3 Encoder

Encoders are applied in hoisting equipment for various applications. The most common ones are for overload warning for stable and secure stand, for restriction of work zone to protect against collisions and to ensure positioning straight to the point. In this project the last point, the straight to the point application, is the focus area and the other applications is not investigated any further.

To monitor the movement of the crane, shaft encoders are installed in each joint. The encoder is an electromechanical device converting reliable angular position or movement into an analog or digital code. The output provides data that is further processed into information concerning speed, distance, RPM and position. Encoders are widely used in applications requiring precise shaft rotations. The information gained from the encoder is compared with the requested position to find the error in the slewing motion. In the current system there is no feedback control to compensate for this error. In the all-electric system, however, feedback from the encoders is continuously provided to the control unit for comparison with the input command. Figure 3.1 shows a simplified block diagram of the control loop. The sensor represents the encoder which converts the state of the controlled quantity to an electrical signal. In the selector the desired output is stated. The comparison of the input with the actual values, data from the sensor, is executed in the control circuit. The controlled quantity is sent to the output transducer. For detailed information of the control system, see Section 3.5, as well as Chapter 5.

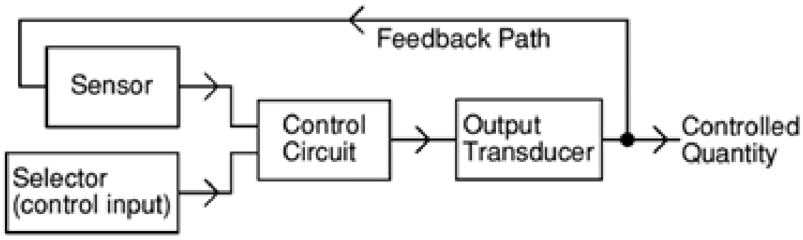

Figure 3-1 Control Circuit [18]

3.2 Backlash

The hypothesis this study aims to verify is whether the backlash in the slew bearing of a crane is reduced by replacing the hydraulic motor drive. It is believed that the control of the slewing is optimized by introducing an electric motor, motor drive and a well functional control system. This is discussed in Chapter 1.3, where it also is stated that the accuracy of the slewing is improved as a result of the advanced control system and the backlash adjustment.

Adjusting the backlash effect in the slew bearing is intended performed by changing the motor drive. This means that the rotating gears of the slew bearing receive torque from electric motors instead of hydraulic. To arrive at a good solution and verify or falsify the hypothesis, an additional theoretical study of backlash is carried out. How backlash occurs, and how it acts in relation to the gears discussed in Section 3.2.1, is essential for this project.

3.2.1 Backlash Definition

In every mechanical system where the motor is not directly connected with the load, backlash is represented. Backlash is the error, the deviation, occurring in gears when there is a change in direction or a disturbance on the system and is defined as the quantity of lost motion due to a slack between the teeth when the gears are reversed. See Figure 3.2 [19]. The phenomenon is a nonlinear behavior and if it occurs it changes the stability and performance of the system. The nonlinearity arises when the backlash gap opens and the load loses contact with the motor side. At this moment, no torque is transmitted through the shaft and the torque generated by the motor is only driving the motor itself. When the contact between the gears is recovered, the impact results in an enlarged torque. The change from zero to a larger magnitude of torque makes the system unstable and limit cycles generating oscillation can occur. The oscillation often acts in an irregular fashion and the peak-to-peak amplitude can exceed the total size of the backlash gap [20]. A limit cycle is an isolated closed trajectory and can only occur in nonlinear systems.

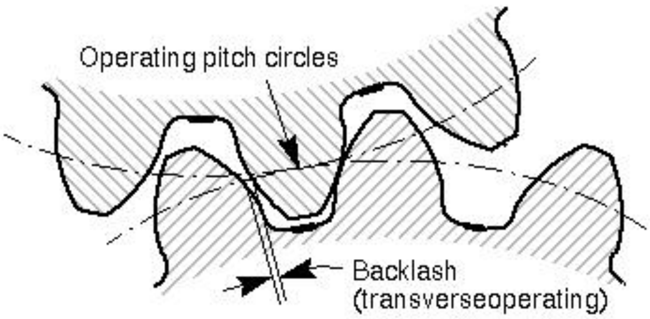

Figure 3-2 Backlash [21]

Although backlash theoretically should be zero, there is always some backlash present. A certain amount of backlash is allowed to prevent the transmission system to get completely deadlocked. If suddenly fixed, the gears may lead to critical mechanical failure. There is a delicate balance between the minimal and maximal allowed backlash, to ensure the best operation of the gears and motor drive. Other reasons for accepting some backlash is lubrication, deflection under load and extensions due to thermal strains. Material factors, with respect to design, influencing the minimal necessary backlash in gears are tooth thickness, helix angle, center distance, pitch circle and runout. Several of these features are seen in Figure 3.3.

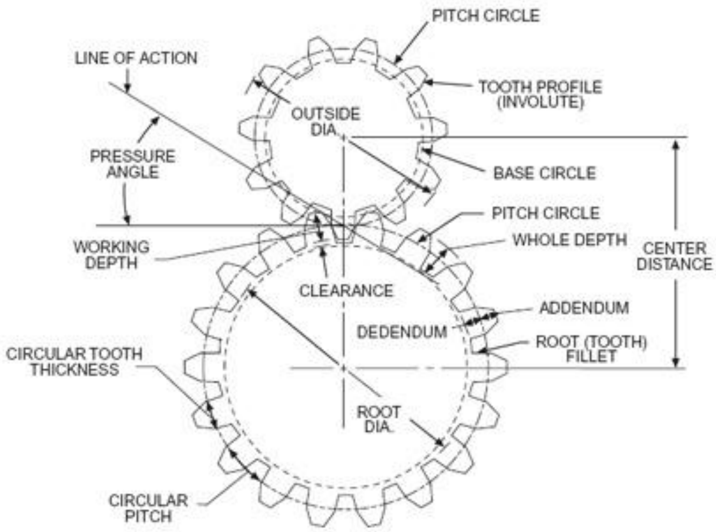

Figure 3-3 Backlash in Detail [22]

The better designed gear system, the less backlash are required. Increased backlash is introduced in the system by 1) forming deeper teeth than necessary in the gear, e.g. tooth thickness is below the value of zero backlash, or 2) the distance between the gears is increased, meaning greater center distance than the value of zero backlash. The numerical value of backlash is calculated differently based on which of the equations (3.1) that are preferred [23].

$$J = S_{std} - S_{act} = \Delta S$$
$$J = 2 \times \Delta a \times \tan \alpha \tag{3.1}$$

Where,

J	Linear backlash along the pitch circle in either case
S_{std}	Standard tooth thickness for ideal gears (no backlash)
S_{act}	Actual tooth thickness (after introducing backlash)

Δa Change in center distance

α 0.5 × backlash angle

The standard practice is to make half of the allowed backlash for tooth thickness for each gear. However, when the pair of gears consists of a significantly smaller pinion, as is the case for this specific crane, it is customary to place all allowed backlash on the mating gear.

The balance of minimal and maximal backlash is extremely subtle and the backlash can neither be too large nor too small. In order to meet the criteria the following aspects must thus be in place; backlash reduction by precision gears and an accurate control system.

The backlash is reduced by two methods, a static and a dynamic method [23]. The methods are not subject to any severe analysis in this Chapter and only briefly explained. The static method for controlling backlash in gears involves the action of introducing backlash as described. By different combinations of fixed and adjusted gear size and center distance, the static method provides the desired backlash. The dynamic method is related to the static techniques involved. An external force is introduced and this imposes adjustments on either the tooth thickness or center distance. This eliminates the backlash in the system, regardless of its position. All combinations and methods for controlling backlash need assessment in order to achieve precise gears, and furthermore, an accurate slew bearing.

As already stated, the implementation of an additional control system is highly essential and needs to be precise and well designed to achieve reduced backlash. An accurate control system provides a precise gear system, even when, and due to, minimal backlash is present.

3.2.2 Master / Slave Drive

In the attempt to reduce backlash, the most common solution is to implement a control system that regulates the input to the motor side. Various approaches are used, but all are relying on feedback from the system. In this report a combination of a control system and the influence of an opposite directed torque are used in the effort to reduce the effect of backlash. To stabilize the system and avoid limit cycles, it is decided to apply a torque, T_{m2}, in opposite direction of the main driving direction. The approach is to use two of the four motors to counteract the main torque. There exist two distinct procedures of doing so; either by applying torque from the beginning, $t = 0$, or by defining a point in time, $t > 0$, from where the constant torque is applied. Considering the system's total energy consumption, the second option is preferred. When not in backlash, all four motors are supposed to contribute in the main driving direction. By doing so, the counteracting motors act as spring and brake to improve positioning and stiffness just when the system enters backlash mode. This technique is called master / slave and from here on the motors is referred to as master or slave. The motors counteracting are the slave.

The master / slave system is modeled as a three-mass system. The three masses, represented by the two motors and the single load, are shown in Figure 3.4.

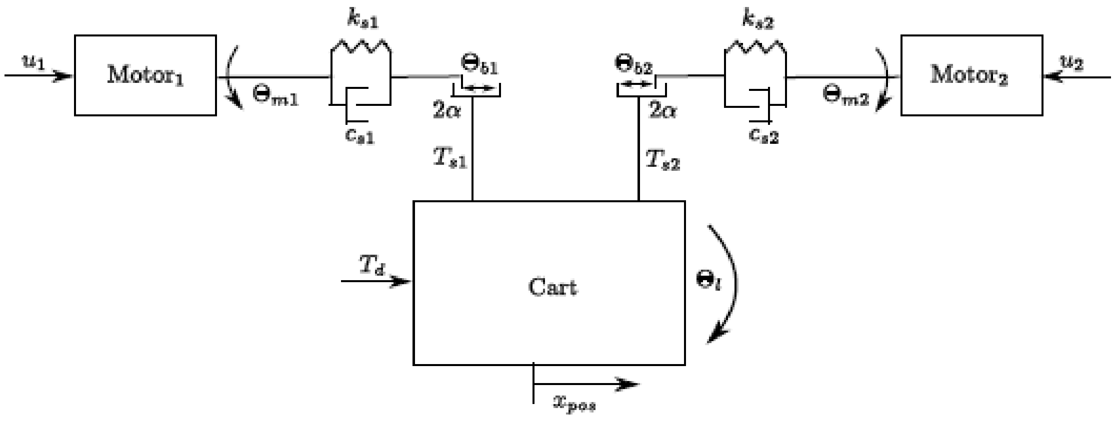

Figure 3-4 Three-mass System with Backlash Elements [24]

The three-mass system is derived from the more common two-mass system consisting of one motor and load. Motor and load are considered to be connected by a mass or inertia free shaft. This is a widely used model in motion control systems of practical interest and studied in literature since 1940 [20]. In this specific case a second motor is introduced to reduce the impact of backlash. A torque acting in opposite direction of the main driving torque, see Figure 3.5, is aimed to close the backlash gap.

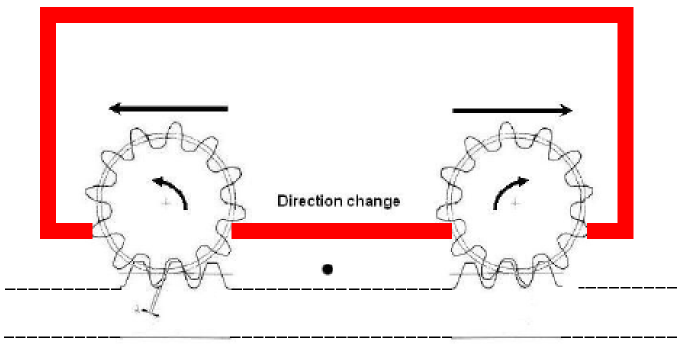

Figure 3-5 Master / Slave Technique used for Minimizing the Backlash Gap [24]

Equations (3.2) - (3.4) describe the three-mass system and assume the gear ratio between motor and load to be 1:1.

$$
\begin{aligned}
&J_{m1}\dot{\omega}_{m1} = -C_{m1}\,\omega_{m1} - T_{s1} + T_{m1} \\
&J_{m2}\dot{\omega}_{m2} = -C_{m2}\omega_{m2} - T_{s2} + T_{m2} \\
&J_l\dot{\omega}_l = -C_l\omega_l - T_{s1} - T_d \\
&\omega_{d1} = \omega_{m1} - \omega_l \\
&\omega_{d2} = \omega_{m2} - \omega_l \\
&X_{pos} = r_l\theta_l
\end{aligned}
\tag{3.2}
$$

With

$$\dot{\theta}_{mi} = \omega_{mi}, \dot{\theta}_l = \omega_l\,, \dot{\theta}_{di} = \omega_{di} \tag{3.3}$$

$$T_s = \begin{cases} k_s(\theta_{di} - \theta_{bi}) + C_s\omega_{di}, & \textit{in contact} \\ 0, & \textit{in backlash} \end{cases}$$

and

$$\theta_{bi} = \begin{cases} \max\left(0, \Delta\dot{\theta} + \frac{k_s}{c_s}(\theta_{di} - \theta_{bi})\right), & \theta_{bi} = -\alpha \\ \dot{\theta_{di}} + \frac{k_s}{c_s}\ (\theta_{di} - \theta_{bi}), & |\theta_{bi}| < \alpha \\ \min(0, \Delta\dot{\theta} + \frac{k_s}{c_s}(\theta_{di} - \theta_{bi})), & \theta_{bi} = \alpha \end{cases} \tag{3.4}$$

i = {1,2}

Where;

J_{m1}, J_{m2}	Motor moment of inertia
c_{m1}, c_{m2}	Viscous motor friction
T_{s1}, T_{s2}	Transmitted shaft torque
T_{m1}, T_{m2}	Motor torque
J_l	Load moment of inertia
c_l	Viscous load friction
T_d	Load torque disturbance
k_s	Shaft elasticity
c_s	Inner damping coefficient of shaft

The angles θ_{mi}, θ_l and θ_{di} are the motor angle, load angle and difference angle given in radians, respectively. $\dot{\theta}_{mi}$, $\dot{\theta}_l$ and $\dot{\theta}_{di}$ are the angles time derivatives; the motor angular velocity, the load angular velocity and the difference angular velocity. The absolute position of the load is denominated as x_{pos} [m], while the radius of the load is r_l [m] [24].

In master / slave driving mode, the distinction between operational modes has to be expressed in a more complicated manner than stated in equation (3.3). When the system is in contact, transmission of torque from at least one motor is possible. The torque transmission is zero when in backlash. The two different cases are defined in equation (3.5) and (3.6).

$$Backlash: \begin{cases} \theta_{bi} = (-\alpha) \land \omega_{bi} > 0 \\ |\theta_{bi}| < \alpha \\ \theta_{bi} = \alpha \land \omega_{bi} < 0 \end{cases} \tag{3.5}$$

$$Contact: \begin{cases} \theta_{bi} = \alpha \land \omega_{bi} \geq 0 \\ \theta_{bi} = (-\alpha) \land \omega_{bi} \leq 0 \end{cases} \tag{3.6}$$

Where i = {1, 2} and the notation *bi* refers to the backlash in motor 1 or 2.

Concentrating on the model in Figure 3.4, the maximum backlash gap is represented with α. The mathematical model chosen for this purpose depends on the mechanical surrounding and operating conditions. The dead zone model for backlash is a common method used for analysis of nonlinear systems. This model was first found in 1954 and the theory is further investigated in later years.

The model assumes that a constant torque output best estimates the operation point for a shaft with backlash. The dead zone model is a simple and widespread backlash model and is in fact the model most used in practice. It is a static, scalar nonlinear function [25]. There is important to point out that shaft torque, T_s, and shaft twist, θ_s, is proportional in this case. See equation (3.7).

$$T_{si} = k_s \theta_{si} = k_s D_{\alpha i}(\theta_{di}) \tag{3.7}$$

Where the dead zone, $D_{\alpha i}$, is defined in equation (3.8).

$$D_{\alpha i} = \begin{cases} \theta_{di} - \alpha_i & if\ \theta_{di} > \alpha_i \\ 0 & if\ |\theta_{d_i}| \leq \alpha_i \\ \theta_{di} + \alpha_i & if\ \theta_{di} < -\alpha_i \end{cases} \tag{3.8}$$

When modeling a nonlinear system, the system can be divided into two parts; a linear part and a nonlinear part as shown in Figure 3.6.

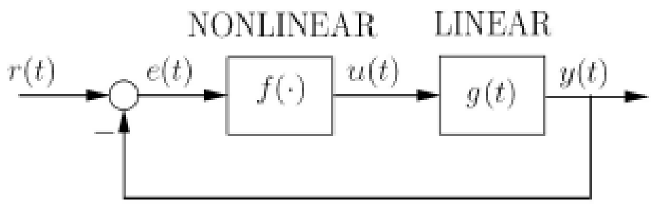

Figure 3-6 Linear and Nonlinear Part of System [25]

The linear part, g(t), represents the system without backlash and the nonlinear part, f (•) represents a modulation of the backlash using the dead zone model. The nonlinearity can be described with a describing function, which is an approximation of the actual situation. The steps defined in equations (3.9) - (3.10) are used to derive the description function. The input of the nonlinear element is given as a sine wave with constant offset B;

$$\theta_d\ (t) = B + A\ sin(\omega t + \phi) \tag{3.9}$$

An approximation of the output of the nonlinear elements is defined by a constant offset $N_B B$ at the first harmonic $N_A A$:

$$\theta_s = N_B\, B + N_A\, A\, sin(\omega t + \phi)$$
$$N_A\, (A, B, \omega) = N_p\, (A, B, \omega) + j\, N_q\, (A, B, \omega) \tag{3.10}$$
$$N_B = N_B\, (A, B, \omega)$$

These parameters are called the duel input describing function, DIDF, and can be represented by the reproduction in equation (3.11):

$$T_s(t) = T_s(\theta_d, \dot{\theta}_d\,) \approx N_B\, B + A\, N_p\;\; sin(\omega t) + AN_q\, cos(\omega t) \tag{3.11}$$

Operational conditions are described by equation (3.12):

$$T_0 = BN_B\, (A, B, \omega) \tag{3.12}$$

Equation (3.12) has the unique solution B*(A, T_0, ω), and therefore the system around the working point T_0 can be described by B*(A, T_0, θ) and NA(A, B*(A, T_0, ω), ω) in DIDF setting.

In the case where $T_0 = 0$, the describing functions are reduced to a sinusoidal input describing function, SIDF. It is most common to describe the backlash with a SIDF. This gives the describing function given in equation (3.13).

$$N(x, \omega) = \frac{Y_1}{X} e^{(j\phi_1)} \tag{3.13}$$

With

X	Amplitude of input sine
Y_1	Amplitude of the first harmonic component

In many cases the describing function is frequency dependent, even though this is not a necessary characteristic. In a nonlinear control system, limit cycles occur when the sinusoid at the nonlinearity input regenerates itself in the loop as defined in equation (3.14).

$$G = -\frac{1}{N(X,\omega)} \tag{3.14}$$

For the model to be valid, the shaft damping has to be negligible or very small [20][25][26].

Furthermore, the direction of the torque for the slave motor has to be decided. To assign the switching variable, υ, of the applied torque for the slave motor, equation (3.15) is applied.

$$v = sign(x_{ref,old} - x_{ref,new}) \tag{3.15}$$

This is the simplest approach and the torque is constantly counteracting from t = 0.

$x_{ref,old}$ and $x_{ref,new}$ refer to the actual position, while x_{meas} refers to the desired final position. The difference between $x_{ref,new}$ and $x_{ref,old}$ allows determination of the direction of the motion.

From earlier research [24] it is known that limit cycles mainly occur when the crane slows down and the load reaches its final position. This knowledge justifies the decision of applying opposite torque only close to end positions. The strategy intended for this report is based on the relative error, e_{abs}, and utilizes the knowledge presented above. e_{abs} is defined in equation (3.16). This method reduces the total energy consumption of the system.

$$e_{abs} = \frac{|x_{ref,new} - x_{meas}|}{|x_{ref,new} - x_{ref,old}|} \tag{3.16}$$

The switching variable, υ, is defined in equation (3.17).

$$v = \begin{cases} 0, & e_{abs} > e_{max} \\ sign(x_{ref,old} - x_{ref,new})\frac{e_{abs} - e_{max}}{e_{min} - e_{max}} & e_{min} < e_{abs} < e_{max} \\ sign(x_{ref,old} - x_{ref,new}) & e_{abs} < e_{min} \end{cases} \tag{3.17}$$

e_{max} and e_{min} are free parameters. Sign is a function in MatLab generating and returning an array containing a υ-value for each x. The value of υ is either -1, 0 or 1 depending on the size of x.

The switching function υ = f(e_{abs}) is shown in Figure 3.7. |υ| takes the value 1, when $e_{abs} < e_{min}$. If $e_{abs} > e_{max}$, υ takes the value 0 and for all other cases υ ∈ [0, 1] [24].

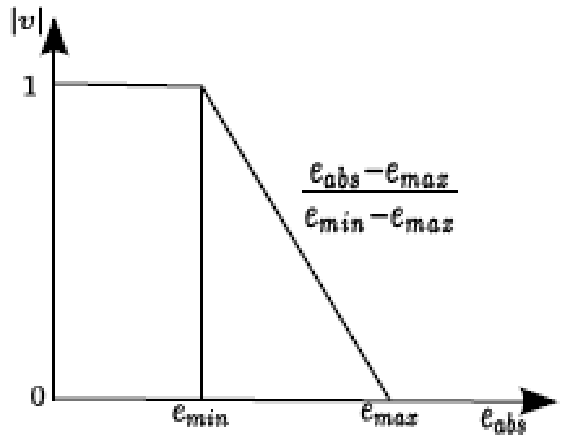

Figure 3-7 Switching Function for Slave Motor

Determining T_m is by the defined function u_2, given in equation (3.18).

$$u_2 = K_{u2} \upsilon \tag{3.18}$$

Where,

υ Switching variable

K_{u2} Constant

3.3 Hydraulic Motor Drives

The existing drive system exploits the technology of hydraulics. Hydraulics is a topic within fluid mechanics and deals with the properties and behavior of fluids, i.e. fluids either at rest (hydrostatics) or in motion (fluid dynamics) [27]. Hydraulics is commonly used for several different disciplines, including transmission of power from motors to rotating shafts. In this particular case it is for rotation of the slew bearing of a crane. For mechanical rotations of devices, hydraulic motors are generally the actuator [28]. In order to rotate the mechanical installation, a drive or transmission system is needed. The drive system consists of three main parts; an electric motor to control the system, a generator (hydraulic pump) to push the fluid through the hydraulic system of pipes and valves and a hydraulic cylinder (hydraulic motor) to drive the machinery. In hydraulic drive systems, see Figure 3.8, the task of actuators is to transform the fluid pressure into torque.

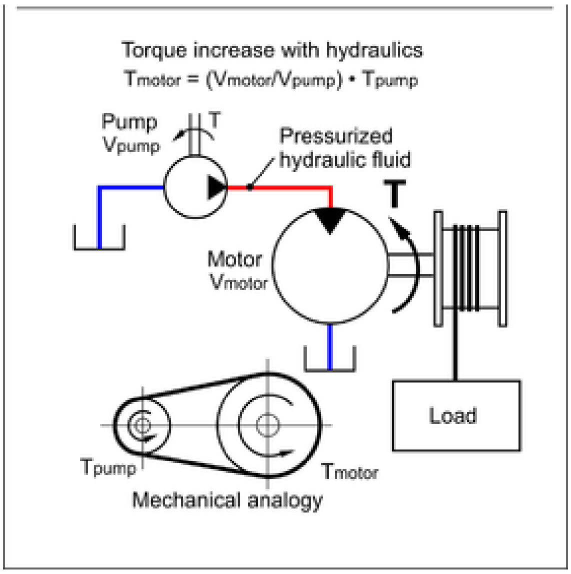

Figure 3-8 Hydraulic Motor Drive [29]

The hydraulic motor must be carefully selected. If choosing wrongly, the desired power or control signal might not be achieved and critical failure can occur. The following are important aspects to consider when selecting the motor; operational speed, pressure, volume, what hydraulic fluid to choose, how to operate it, for what application, price, size, etc.

Several characteristics of hydraulic drives state that there exist more advantages than disadvantages with the technique. The main benefit of the hydraulic drive is provision of an excessive force compared to its modest size. In addition, the power is significantly greater than the power exerted from other motors of similar size. Other advantages are high reliability, continuously variable control of speed, self-lubrication and relative accuracy.

The main disadvantage is the issue of control. A hydraulic system is more challenging to control than an all-electric drive system. Because of the dependency of hydraulic oil there is always a risk of pollution of the environment in case of a leakage. A third down side is the need for maintenance. Hydraulic motor drives often require more thoroughly and frequent maintenance than equivalent electric systems.

3.4 Electric Motor and Drives

Although hydraulic motor drives are most common for rotation of heavy mechanical devices, it is also possible to exploit properties of electrical machinery for this purpose.

There are at least three different principles of how to create electric motors. These are the principles of magnetism, electrostatics and piezoelectric. The theories of magnetisms are by far most common and are therefore the continued focal point in this study.

The primary components in electric motors are displayed in Figure 3.9. In general, all motors have a stationary part, the stator, and a rotating/moving part, the rotor. Stator and rotor are separated by an air-gap allowing free rotation of the rotor. Furthermore, rotor is connected to the motor shaft, whilst stator is fixed to the foundation of the motor case. Windings and magnets are other complementary parts and additional commutators can be installed at the motor assembly. Commutators are used to switch the current to the windings depending on the angle of the rotor, and are also used to internally convert the source from DC to AC for parts of the motor (if DC motor) [30][31][32].

The stator windings of Figure 3.9, often called coils, are conductor elements wounded around cores to create an electromagnet. When an induced electric current passes through the coils, the coils start to move in relation to the magnets of the motor assembly. This gives a formation of magnetic poles and further supplies a magnetic field. In electric motors magnetic fields are produced both in rotor and stator. The product of these fields forms the force providing the motor shaft with the necessary torque. This torque is used for linear or rotational movement. As additional information, one or both of the magnetic fields must obtain the ability to change if the motor reverses direction. This is achieved by switching the poles on or off at a certain switching frequency.

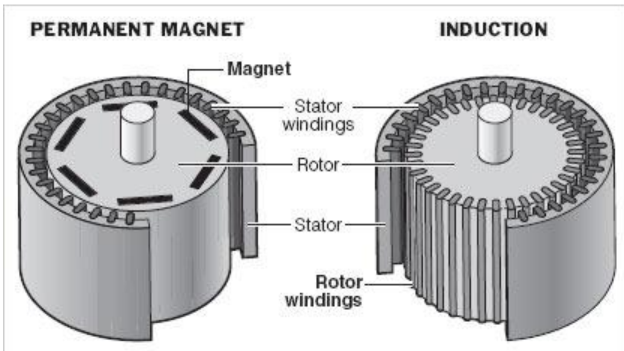

Figure 3-9 Electric Motors [30]

Furthermore, motors are characterized by the shape of the supplied electric current. DC motors is powered by direct current (DC), whereas AC motors are subjected to an alternating current (AC). The latter scenario implies that there is no need of commutators within the motor assembly, as yields for PM DC motors. That is due to specifically improved control systems evening out the distinction between the DC and AC motor, as the control loops has cleared the commutator from the motor.

Motors of an AC source are further classified into synchronous types and asynchronous types. The clear difference in the motor types becomes visible when examining the average torque generation of linear/rotational motors. It is a necessity to have synchronicity between the moving magnetic field and the current. This gives the synchronous motor. The synchronous motor does not request the presence of slip to produce torque, thus the main difference between the two motor types is that the asynchronous do.

Slip is the speed difference between the synchronous and asynchronous (rotor) rated speed of the motor. The speed difference occurs as a result of the interaction of the current flowing in the rotor bars and the stators rotating magnetic field. As the rotor speed actually lags the speed of the magnetic stator field, the rotor bars carve magnetic flux lines in the field, induce a current in the windings and generate torque. These currents further produce their own magnetic fields, which in interaction with stator produces rotational torque to the rotor. It is essential for the current induction that the speed of the rotor is less than synchronous speed, otherwise the stator field will not move in relation the rotor. Slip will increase with increasing load, and decrease with decreasing frequency (speed).

In synchronous motors, induction (slip) is not necessary for the production of magnetic field/current. In other words, synchronous electric motors contain rotors that rotate at the same rate as the rotation of the magnetic stator field. Torque is thereby produced at synchronous speed. The rated synchronous speed is decided by the number of poles in the AC machine.

The asynchronous motor is a kind of AC motor and the general example of such motors are the induction motor. AC induction motors have most often a squirrel-cage shaped rotor. In contrast to the synchronous motor, where rotor rotates at the same speed as the magnetic field of the stator, the rotor in asynchronous motors rotates at a lower speed than the given stator field. Asynchronous machines therefore require slipping.

Traditionally, motors were operated uncontrolled and with constant speed. The process industry today, however, utilizes systems with adjustable-speed drives. By use of electric drives, the speed and/or position of the mechanical load are controlled in more efficient manner and such systems offers high reliability and efficiency.

Historically, the most popular drives for speed and position control have been the DC-motor drive. Relatively low cost and ease of control are the main reasons for its popularity. Recent development in the area of AC-drives has however resulted in an increased and extensive use of such drives. Improved properties as well as no need of commutation, has made it beneficial to choose AC-drives over more conventional DC-drives. Figure 3.10 presents a basic electric drive receiving its power from the utility source, regardless the shape of the source.

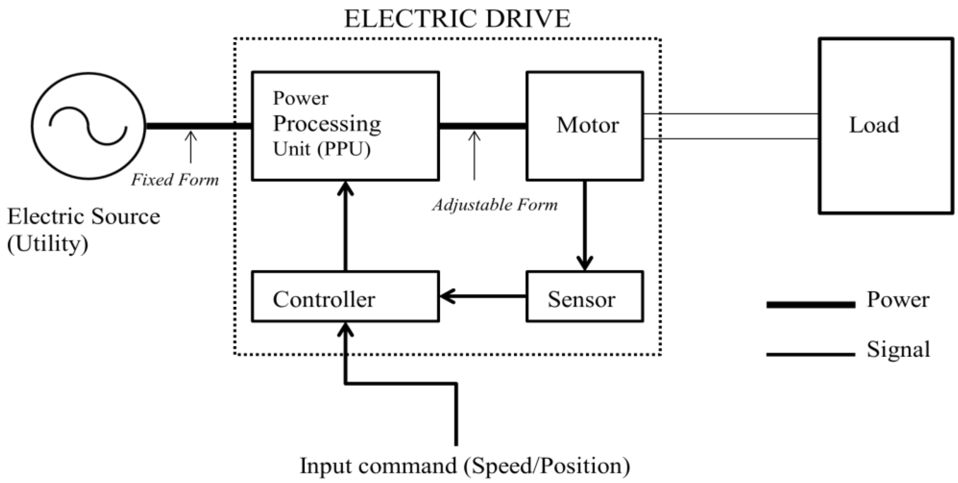

Figure 3-10 Electric Motor Drive [31]

The utility source provides power to the system. It is a single-phase or three-phase sinusoidal voltage of fixed frequency and constant amplitude. The input to the electric drive may be a command from a process computer considering the overall process and issues commands to control the load. The controller then compares the input command for speed and/or position with actual values measured through sensors. This provides controlled and proper input signals to the power processing unit (PPU). The PPU converts fixed-formed input voltages into outputs of the appropriate kind. This means frequency, amplitude and/or number of phases suitable for the selected motor. Eventually, the motor supplies the load with the necessary torque. From the output of the motor, the sensor accurately measures the actual value for speed and position [31].

3.5 Control Theory and Feedback Controllers for Motor Drives

All successful and efficient motor systems rely on some sort of control, where the degree of control varies. The engineering of control systems applies basic control theory in order to design systems with desired performance. Control theory implements several disciplines to deal with the behavior of dynamical systems and obtain the requested effect. For this Section the 2003 edition of Electric Drives, An integrated approach, by Ned Mohan, is used as theoretical ground [31].

The operation of a heavy lift crane is example of application requiring precise control. Feedback control systems are widely used in such applications. The block diagram in Figure 3.10 presents an electric drive containing feedback control.

Feedback control is exercised over several different variables. The variables are termed controlled outputs and for this specific drive it is convenient to choose speed, position and/or torque as the output variables. Sensors register the controlled output values, which is fed back for comparison with the input command.

3.5.1 Feedback Control Systems

The reason for the frequent use of feedback control structures is that it is simple to design and implement. However, the drawback is that the set of stacked, inner loops has most likely a slower response to changes, than for example a control system where all variables are processed and acted upon concurrently.

In feedback systems, the variables are processed separately and consecutively, and it therefore takes a fair amount of time before the system parameters change according to the external influences and desired reference values.

In feed-forward systems, on the other hand, *approximated reference values* of the inner-loop variables are available at all time and are used as input to the process. The position variable is the only one not approximated, as it is measured directly at the system output by encoders.

By feeding approximated variables simultaneously into the system, the disadvantage of slow responses of the cascade control structure is minimized. It is therefore beneficial to implement both feed-forward and feedback in the control of industrial systems like this application. It is precisely the structure shown in Figure 3.11 that is proposed for the control circuit of this motor drive. Reference values for torque and position are applied to the control circuit, in addition to actual values from the encoder.

The cycle of regular feedbacks or feed-forwards are called a controlled closed loop, or a feedback/feed-forward control system. Any error in measured and requested value is sent as amplified control signals to a controller. The controller manipulates the input to reduce the error and achieve accordance between controlled output and command input. Examples of errors are load disturbances, parameter variations and/or imperfect modeling.

The most used structure for control in electric drives is the cascade control. The structure may contain a current loop, a speed loop and/or a position loop. In cascade control the innermost torque loop is required to be the fastest, whilst the outermost position loop is the slowest. Figure 3.11 is an example of how to design simple feedback control loop.

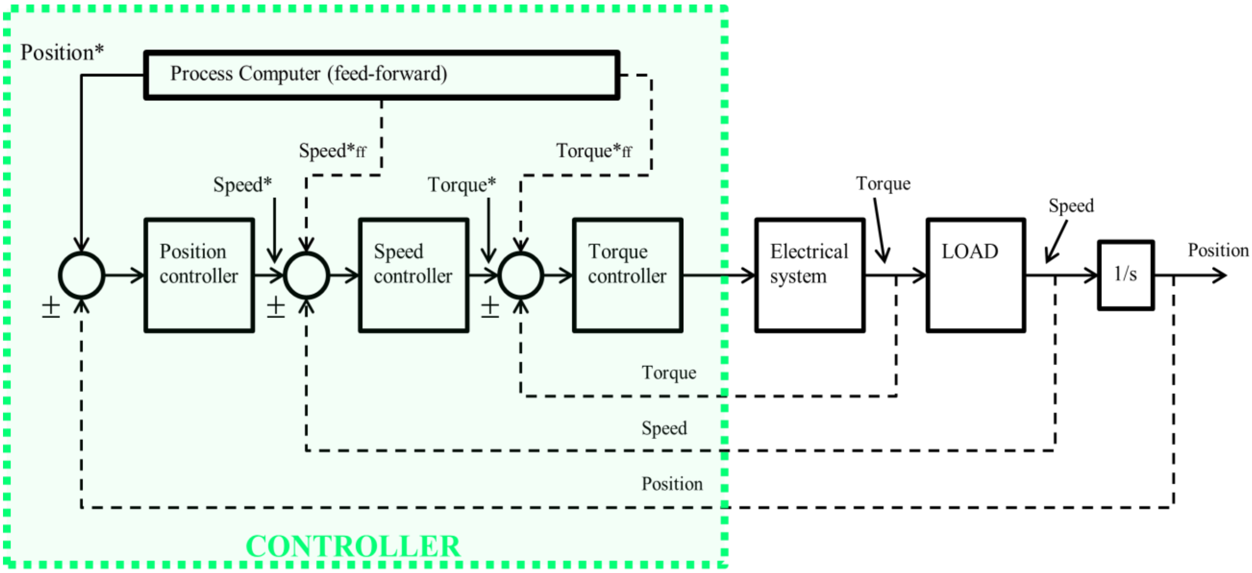

Figure 3-11 Cascade Control Loop

In cascade control loops the speed of response (bandwidth) is increasing towards the inner loops. When designing the controller it is therefore convenient to start with the torque (current) control loop.

As the current is proportional to the torque in permanent magnet DC motors, it is useful to choose the current as control variable. The induced back-emf in the system is dictated as the feedback for the current loop. The midst loop of the controller, the speed loop, has a bandwidth of one order of magnitude smaller than the current loop, as it is not required to respond as quickly. Outermost is the position loop. It has a bandwidth of one order of magnitude less than the speed loop. Consequently, it is the slowest of all the loops.

3.5.2 Controllers

If the objective of the control design is motion control (speed, torque and position) of mechanical applications, this is most often performed by a proportional-integral (PI) controller. In general, it is adequate with P-controllers in the position loop and PI-controllers in the speed and torque loops.

The basic properties of proportional and integral functions, is that P-controllers produces outputs proportional to the present error input, whilst I-controllers gives an output proportional to the integral of the error. In other words, the integral action gives the accumulation of past errors, which should have been corrected in previous stages. If proportional actions are used single-handedly in the speed or torque loop, the result is a steady-state error as response to the step-shaped change in reference input. This is the reason why proportional and integral actions often are combined in control systems.

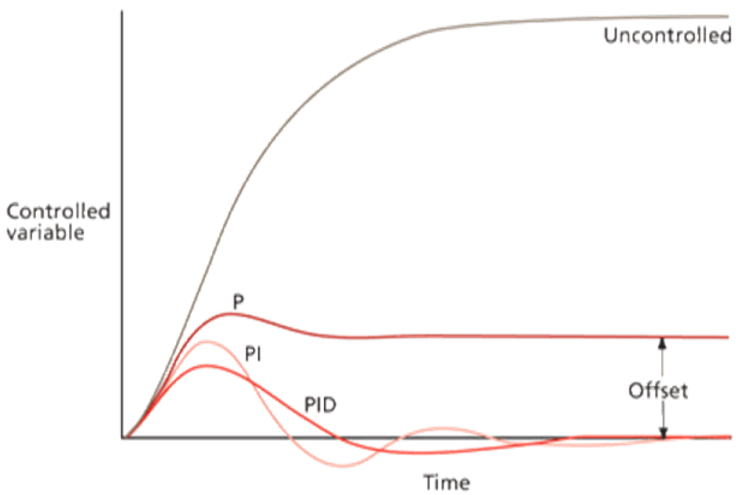

Figure 3-12 P, PI and PID Controllers [33]

As seen in Figure 3.12, the various control actions, treat the variables quit differently. Response time varies from very fast to slow. In addition, the controllers have diverse approaches and behaviors to reach the requested value and minimize the offset/error.

It is the P-controller that changes the output signal proportionally to the change in the input signal. The error is then amplified. Due to the proportionality factor, the P-controller experiences a relative fast response time, while being more or less free of fluctuations. However, it gets an unacceptable large deviation, as it simply just produces an amplified error/offset, as shown in Figure 3.12. It is due to this offset that the PI controller becomes relevant.

The integral function in the PI controller integrates the error over a certain time period, summarizes the results and thereby provides a better accuracy of the output. This is also seen in Figure 3.12. The PI controller has somewhat slower response time compared to the P controller, and the stability is even worse. However, and despite of the oscillations, the output is getting closer to the enquired value. For further approximation of precision, a differential (D) action is introduced in the controller unit. To reduce the oscillations, yet at the same time maintain the phase shifting towards the wanted output value, the PID controller is used. By introducing the derivate, rapid variations in the process value is reduced and this results in improved stability. The control sequence becomes more stable, yet the response time will increase when using the PID controller.

On the other hand, controllers including differential functions (D) is not necessary in the control system for this type of application, and hence PID is not subject for further studies. To support this decision, brief references to the qualities of the D-controller are made. First of all, the derivate function can never be used alone. If differential actions are included in the controller, so must proportional or integral actions. Second, the differential part is a prediction of future errors. The errors are based on the current rate of change, making the derivative part highly sensible to measurement noise. This is why the derivative action often is left out.

With the functions of the various controllers established, this must be related to the qualities HE aims to benefit from in the control design of the project. Discussions with HE stated a request for a best possible combination of accuracy, overshoot and response time. The graphs in Figure 3.13 are used as base for the discussion.

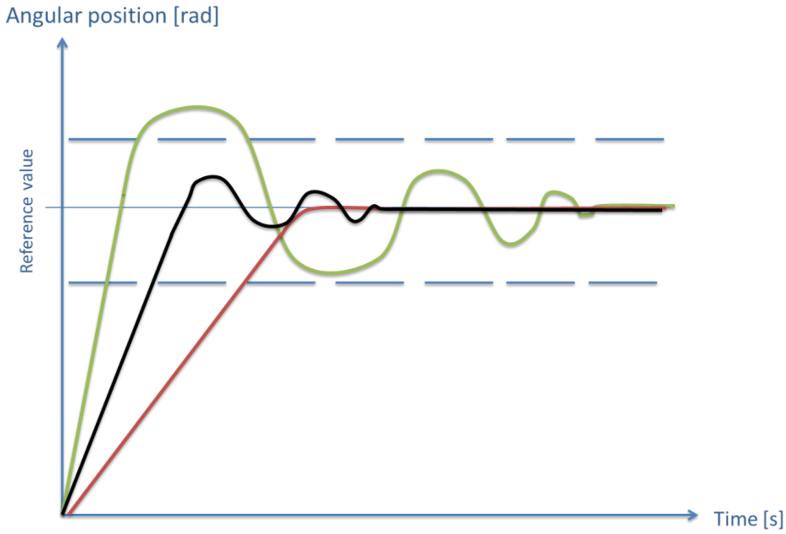

Figure 3-13 Different Response Times

The green graph has a quick response, yet it overshoots strongly and leads to an imprecise/unstable system. The red graph responds more slowly than the green graph. On the other hand, it experiences a much higher accuracy with regards to wanted output value. In between of the green and red graph, is the black graph. The black graph is merged from the green and red graph and satisfies the request from HE for this application. It has lesser fluctuations than the green graph and a quicker response time than the red graph. This yields both greater precision and a relatively fast response time. The positional response is allowed to vary between $\pm$ 1% of the reference position. This is an acceptable offset band set by the project group.

Concluding remarks yields that the PI controller unit is the best suitable choice of controller for this project. The development of the control system is further discussed in chapter 5. Related to the graphs in figure 3.13, used as base for the discussion, this means a merger of the green and red line, resulting in the black line.

3.5.3 Power Processing

When the controller unit has compared the actual output with requested output, the power processing unit (PPU) is next in line for the control of the dynamic system. It is now decided what the magnitude of the control signal needs to be to achieve the wanted position, speed and torque. The PPU, functioning as an actuator, processes the input signal from the controller and supplies the drive circuit and motor with the necessary corrections with regards to frequency, current, voltage, etc. The actuators task is in other words to provide a change in the process values. There exist several components that can act as actuators in electric motor drives; frequency converters, control valves, microprocessors and other types of motors.

The frequency converter, henceforth referred to as FC, is an electrical component used in a wide variety of industrial applications and is also known as variable speed drive (VSD) or variable frequency drive (VFD). Although the practical use of the FC is quite diverse, the basic principle is more or less the same; it continuously adjusts the speed of the motor.

FC's allow precise control of motors in relation to needed requirements. The functional principals of frequency drives are as follows: In general, the FC is split into three stages of operation. The input voltage from the utility source is rectified into DC in the first stage. At the next stage the properties of the direct current is refined before arriving at the final stage of the converter. In the final stage the electrical quantities are controlled to ensure that the desired motor action is achieved.

The contribution from VFD is improved control of the overall process (rotation/position, speed and torque), increased productivity, less energy consumption and reduced maintenance. Another benefit of the VSD is the capacity to soft-start the motor application. Set as individual points, the following is important features of the frequency drive:

- Variable frequency
- Easy change of rotational direction
- Variation of acceleration and deceleration time
- Determination of power at different frequencies
- Adjusting the rated current so motor protection can be left out

These characteristics are useful for the electric drive of the slew bearing in this project. According to the specifications, HE requests the drive to accelerate and decelerate the crane in either direction. The aim of the design is thus to obtain properties to accelerate the rotation of the slew bearing in a certain direction, then decelerate and come to a standstill. A loaded crane should be maintained completely at standstill by the motor drive. For this operation it is further preferable that the slew bearing smoothest possible changes the direction of rotation and once again obtains similar acceleration and deceleration. The highly fundamental features of the slew bearing must be maintained regardless of external influences or loads. Otherwise, if the slew bearing is not functioning optimally the crane experiences a limited area of operation.

3.5.4 Modes of Operation

As referred to, electric motor drives with FCs possesses the necessary and demanded features of motoring and generating mode. Systems capable of speed and torque control in both rotational directions are known as having four quadrants of operation [34][35][36].

Quadrant 1 and 3 in Figures 3.14 and 3.15 are defined as motoring modes of operation. This occurs when the system, powered by the source, accelerates a load in either direction. In quadrant 1 the speed and torque are presented as positive, or in the same forward direction. The speed and torque are also of equal polarity in quadrant 3, however now in the negative or reverse direction.

Quadrant 2 and 4 in Figures 3.14 and 3.15 are the generating modes, yielding a regenerative decelerating effect. In these modes the speed and torque are of mutual opposed directions, meaning that the torque opposes the rotational direction and generates electrical energy. The energy can then either be given back to the voltage source or stored in the capacitor for later use. In other applications the generated energy is simply transferred into heat in brake resistors.

As the selected drive for the slew bearing makes some of the motors counteract the others, an amount of excess energy is developed during the operation. Using this energy for other purposes is the most energy efficient. New use or disposal of the excess energy in this project is discussed in Chapter 7.

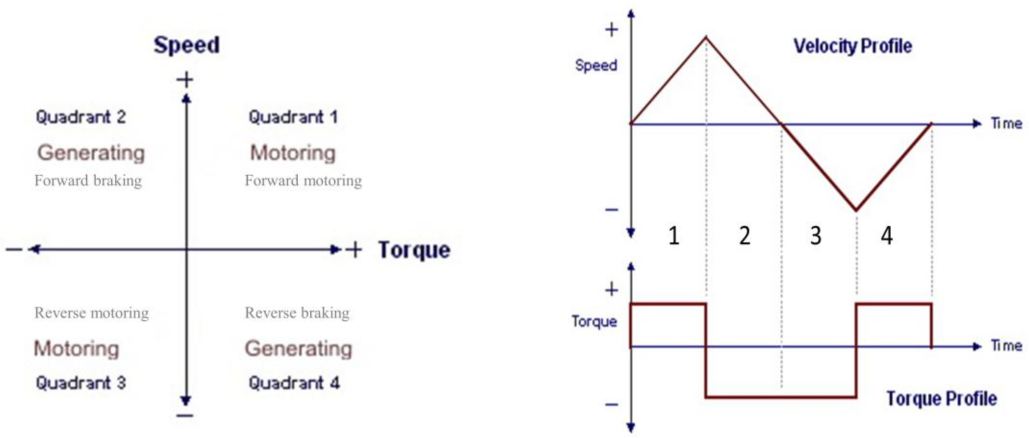

Figure 3-14 Four Quadrant Drive and Torque and Speed Characteristics [34]

The four operational modes of a VSD are possible because the torque is directly proportional to the armature current. This current depends on the difference in applied voltage and induced back-emf. Motoring and generating mode of the drive is achieved by controlling if the applied voltage is greater or less than the back-emf.

Variable frequency drives have various load combinations, in terms of speed and torque. The load of the crane is hardly constant. It rather varies widely within the load chart, and it is therefore important to consider the load characteristics to achieve a smooth and trouble-free operation of the slew bearing.

3.5.5 Benefits of Controlled Motor Drives

Compared to the existing hydraulic system, frequency control of electric motor drives provides great benefits. Today, hydraulic motor drives include traditional volume control. Control valves are installed to give signals to the actuator. The valve is either choked or freed, regardless of the motor speed. The motor is continuously operated at full speed. This is compared to driving a car at full throttle and adjusting the speed with the brakes. This results in large inaccuracies in terms of control and also greater energy consumption [37][38][39].

For this reason, adjustable speed drives are introduced instead of such uncontrolled systems. The motor speed is now controlled to required value. This provides tremendous energy savings, as well as the mentioned benefits of improved accuracy, faster response and less maintenance. An improved operation of the motor system, according to ABB, gives a total energy saving in the order of 30-60 % compared to conventional hydraulic drives with throttling control valves [40]. This is exhibited in the diagram in Figure 3.16.

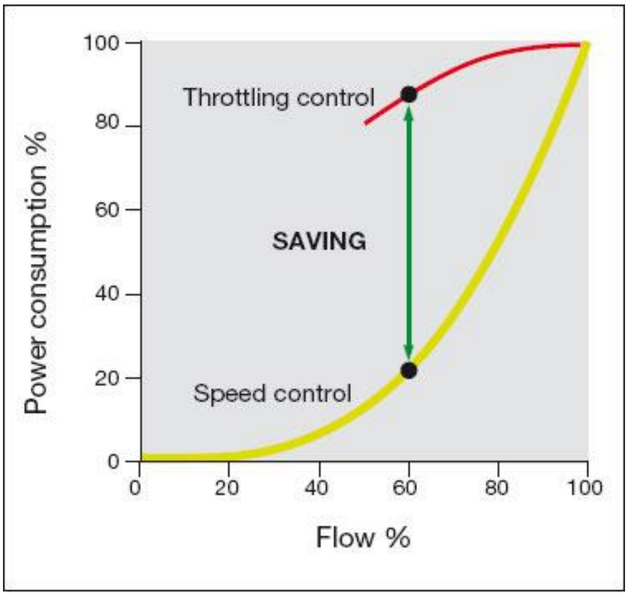

Figure 3-15 Benefits of Controlled Frequency [38]

The area between the yellow and red line represents the available amount of energy saved if the control system is improved. As well as reduced power consumption, the speed control contributes to lower mechanical strains during start and stop of the motor.

To summarize, the installation of electric motor drives provide significant savings in energy consumption. This is in addition to precision improvement and other beneficial operational features. These aspects are reasons and additional motivation for HE to exploit the technology of electric motor drives in slew bearings of offshore cranes.

3.6 Standards and Safety Considerations

The project studies an actual crane. Although the 50T AHC is the first of its kind designed from scratch and delivered by HE, both HE and customer R-R has experience with heavy lift cranes for shipboard/offshore/subsea use. The companies have therefore good knowledge to the demands and requirements related to safe use of cranes. Client HE has provided a number of standards normally valid for this type of crane, which the design should comply with. Additionally, a series of other standards related to the electrical installation of the crane is found. The standards mainly yield for heavy lifting appliances, such as the 50T AHC crane in this case. A review of the IP rating for this crane is also performed.

3.6.1 Standards of Certifications

All system design, whatever being designed and for what purpose, must follow certain specifications that apply to the given application. As guidance, there exists a number of Norwegian (NS), European (EN) and international (ISO) standards of certifications that specifies the requirements in most sectors. Within the offshore and shipping industry, there are also extraordinary criteria in terms of safety. This project tentatively results in a functional, reliable and improved electric motor drive system for the slew bearing of an offshore crane. The design should fulfill all demands the client HE, the customer R-R and/or the project group imposes on the product. In addition, all applicable fundamentals and certification specifications must be met. The listed standards of certifications are those reviewed and which the project design constantly is assessed up against.

- **No. 2.22** - Lifting appliances (DNV).
- **NS 13852-1:2004** - Cranes. Offshore Cranes. General purpose offshore cranes.
- **EN ISO 13849-1** - Safety of machinery. Safety related parts of control systems.
- **NS 5513** - Cranes and lifting appliances. Electrical equipment.
- **EN 13135** - Cranes. Equipment. Electrotechnical equipment.
- **EN 60204-32:2008** - Safety of machinery. Electrical equipment of machines.
- **EN ISO 12480** - Cranes. Safe Use.

3.6.2 Protection Systems

In addition to the specifications for the design and development of the system, there are also a number of other special conditions and demands to consider regarding the operation of the crane. The goal is that the given standards establishes, maintains and develops an acceptable level of safety of personnel, property and environment in the planning and execution of lifting operations.

AOPS – Automatic Overload Protection System

The European Standard EN 13852-1 and the National Regulations of Norway requires that all offshore cranes are installed with an automatic overload protection system (AOPS). AOPS's is fitted to cranes with the intention of automatically protecting the crane from major damage and high loads that might suddenly appear too quick for the operator to react [41][42].

MOPS – Manual Overload Protection System

The manual overload protection system (MOPS) works under normal operations of the crane, not only during the event of failure, and is manually activated to protect the crane for loads higher than the design load. Meaning, in situations where the load gradually increases enough for the operator to react to it [42].

Overspeed protection of electrical motor drive

According to the 3rd revised edition (1998) of F.E.M. 1.001, all electronically controlled motions of hoisting appliances must be equipped with overspeed protection. Overspeeding means that the hydraulic or electric motor reaches a condition where it exceeds its designed speed limit. If reaching the limit, greatly reduced lifetime and/or mechanical breakdown may occur. Hence, the reason for installing overspeed protection is to prevent the slew bearing from reaching its mechanical speed limit, as well as avoiding uncontrolled and unintentional motions of the crane.

The limit for overspeed protection should in accordance with F.E.M. 1.001 be triggered so that the mechanical brake can safely stop the slewing of the crane in all operational modes. The general rule is that the trigger limit should be within 1.2 times of the specified nominal speed (speed at nominal load).

3.6.3 Ingress Protection Rating

The ingress protection rating system is used to specify the environmental protection of electrical armatures. Ingress protection ratings, or IP ratings, are important as both customer and supplier should be able to trust the reliability, suitability and safety of the electrical product during extremely demanding mechanical and environmental conditions.

The first number of the IP rating states the protection from solids and foreign objects. The value of 0 yields no protection, whereas 6 implies no intrusion of dust what so ever. The second number in the classification gives the degree of protection from liquids, most often water, as the electrical equipment is to be located at sea. The scale of protection from water goes from 0 to 8, where 8 indicate totally protected against permanent submersion in water [43][44].

The IP rating of the specific crane used in this project is set to 56. The direct interpretation of this IP rating is as follows:

5 – Protected against dust. Ingress of dust is not totally prevented but dust does not enter in an amount that prevents satisfactory function or impair safety.

6 – Protected against powerful water jets from either direction or side.

Chapter 4

System Review

This chapter describes the 50T AHC crane. There is also a detailed description of the existing drive system and the components included in the slewing of the crane. The review of the existing system proved is quite substantial, as it highlights a number of shortcomings and simultaneously provides the project group with new and creative solution ideas. Thus, the improved system design is briefly presented in this chapter.

4.1 The Crane

The subject of this project is a 50T AHC crane. During the project there is not performed any modifications on the crane construction as the problem definition involves only the drive system for the slewing of the crane.

The crane is equipped with an active heave compensating (AHC) system. By constantly adapting to the wave motion, the AHC system enables the crane to maintain precision and load stability and allows accurate and efficient load handling at offshore and subsea environments. By moving the elbow derrick the active heave compensation is achieved. This technique makes the AHC operation fully linear. Using this design for AHC provides the possibility of mounting of cargo handling tools and results in easy handling of cargo on deck.

The AHC system is functional from empty hook to safe working load and allows both landing and lifting of load from seabed. Main data for the crane is specified in table 4.1.

Table 4.1 Crane Specifications

Main Data	**Values**
Max. working radius	20.0 m
Min. working radius	3.0 m
Dynamic factor	According to DNV 2.22 for subsea lift
Crane SWL	50 T at 3-8 m 20 T at 8-20 m (linear reduction) 2 T at 0-3 m (special operation only)
Wire length	1500 m
Wire type	Redaelli
Wire diameter	To be confirmed
Max. wire speed (inner layer)	70 m/min at 60 Hz
Slewing sector	Continuous
Slewing time	Max 70 sec/rev-1.8 m/s at max. boom radius
Heel + trim	5 + 2 degrees
Voltage	Variable from 575 V at 50 Hz To 690 V at 60 Hz
Frequency	50 Hz to 60 Hz
Heating and flood light	230 VAC
Starter	Y/D
Power consumption, crane operation	To be confirmed

Figure 4.1 shows a draft of the 50T AHC crane and specify the crane dimensions. Compared to a traditional knuckle boom crane the improved design on the 50T ACH has a large advantage in the “square” type load chart which provides higher SWL over larger area. SWL refers to the safe working load of the crane.

The safety systems include overload protection, operational limits and dangerous cargo and crane movements. These are described in Section 3.6.2.

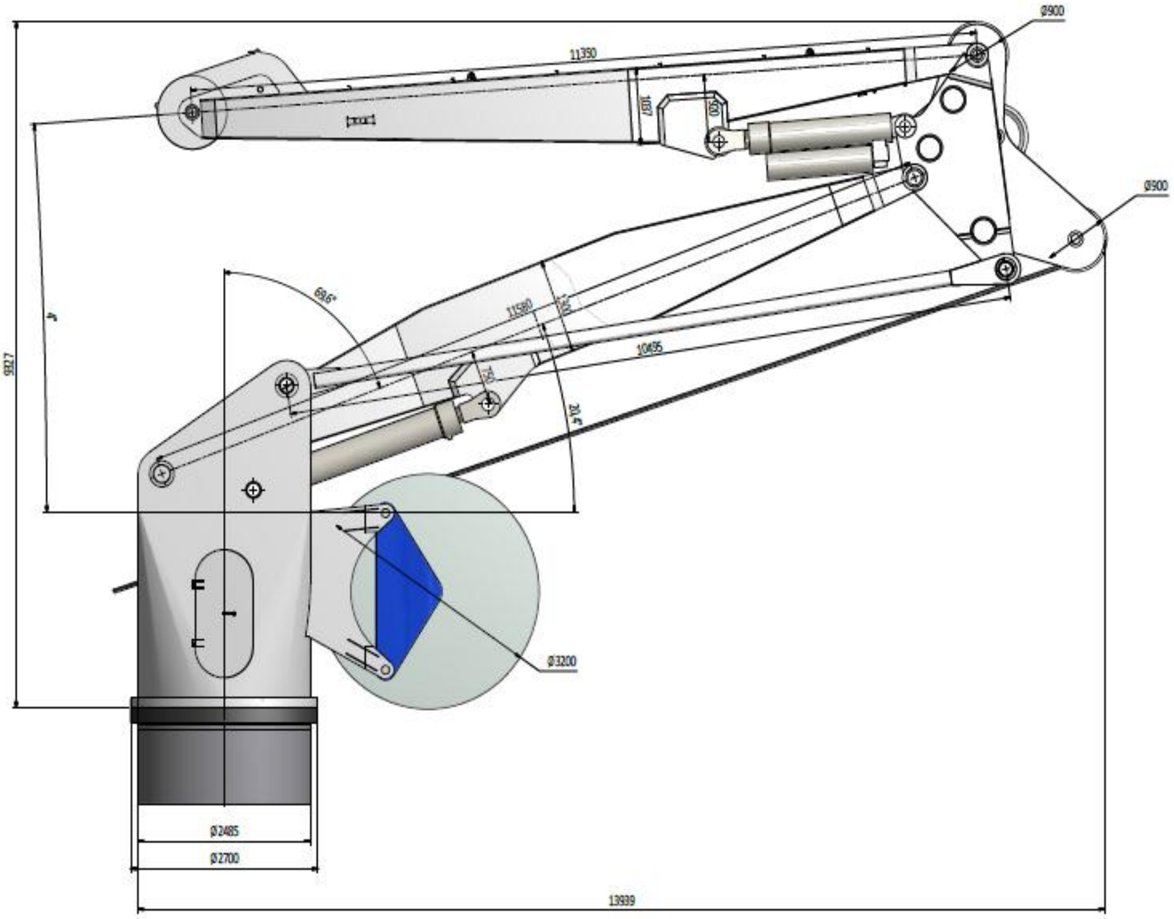

Figure 4-1 50T AHC Crane

What concerns the environmental criteria, the crane is designed to endure the criteria listed in table 4.2. As the crane is installed on vessels operating worldwide, the factors of environment effects varies over a wide range. The shifting air temperature and humidity generates a need for extra caution concerning condensation in control room etc. Machinery and equipment compartments are designed for inside ambient temperature up to maximum +55°C.

Table 4.2 Environmental Criteria

Temperature	**Value**
Design temperature	-20°C
Ambient temperature	-20°C to +45°C
Atmosphere	High in salinity
Humidity	Up to 96%
Wind exposure	**Value**
In operation	20 m/sec.
Extreme and accidental	30 m/sec.
In stowed position	according to DNV requirements

4.2 Existing System

The intention of this Section is to outline the hydraulic driven slew system of the crane. The components included in the slewing process are listed and described.

4.2.1 Slew Bearing

The slew bearing is delivered from Rothe Erde. Main specifications are listed in table 4.3.

Table 4.3 Specifications of Slew Bearing

Description	**Ch.**	**Unit**	**Value**
Outer diameter	DA	[mm]	2700
Inner diameter	DI	[mm]	2140
Height	H	[mm]	231
Weight	App	[kg]	2724
Number of teeth	Z	[1]	108

Dimensioning of the bearing is based on calculations of the static safety load specified according to the maximum loaded raceway. The loads are determined on the basis of information from HE. For more information and details see appendix B. Corresponding to the specified raceway loads and the standard *DINISO281 (L10),* calculations concerning bearing life expectations are performed. Note that for large diameter slew bearings this numbers is only to be used as benchmark. The evaluations resulted in an expected life time of 598 hours or 35 880 rotations.

Figure 4.2 and the equations (4.1) define the applied loads on the bearing ring.

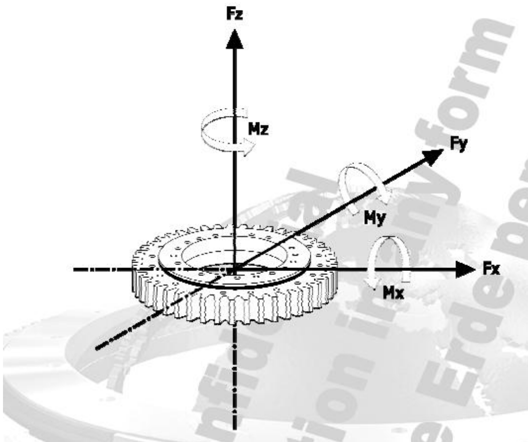

Figure 4-2 Forces Acting on Crane [45]

$$F_a = -F_z$$

$$F_r = \sqrt{(\mathrm{F}_x^2 + \mathrm{F}_y^2)} \tag{4.1}$$

$$M_k \sqrt{(\mathrm{M}_x^2 + \mathrm{M}_y^2)}$$

$$M_d = M_z$$

Where,

F_a	Axial load	M_k	Tilting moment
F_r	Radial load	M_d	Torque
F_x	Load in x-direction	M_x	Torque x-direction
F_y	Load in y-direction	M_y	Torque y-direction

F_z	Load in z-direction	M_z	Torque z-direction

The tip relief of the bearing is formed as an involute. An involute, also called evolvent, is a curve obtained from the information of another given curve. In this case this refers to the slewing wheel and the pinion. Figure 4.3 shows a sketch of the design of the tip relief. The shape of the tip is designed for the pinion and the slewing wheel to merge evenly together without any edges conflicting or damaging the construction [45].

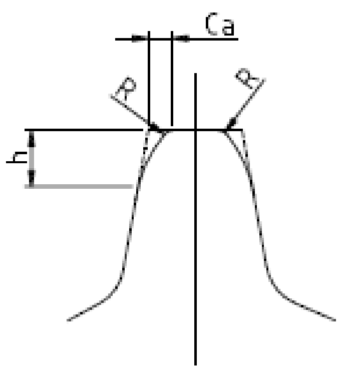

Figure 4-3 Tip Relief of Bearing Ring [45]

4.2.2 Hydraulic Motor

In the current system a Parker Series F12-90 hydraulic motor is installed. The term 90 represents the fuel displacement in the motor, given in cm^3/rev. Figure 4.4 shows an illustration of flow versus shaft rotation. In the motor application the shaft turns clockwise when port B, represented by the black arrow, is pressurized. When port A, the white arrow, is pressurized, the motor turns counter clockwise.

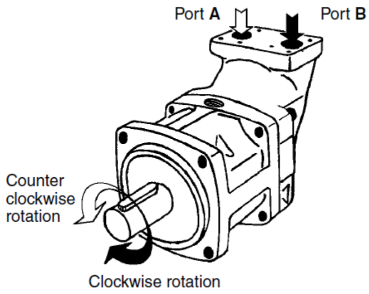

Figure 4-4 Illustration of Flow versus Shaft Rotation [48]

The F12 conforms to ISO and SAE mounting flange and shaft end configurations. The spherical piston design allows the F12 to operate with an unusually high shaft speed. Being capable to operate pressures up to 480 bars provides for a high output power capability. The timing gear is designed for high synchronization between shaft and cylinder barrel. This design makes the F12 particularly tolerant to high G-forces and torsional vibrations. Torsional vibrations are angular vibrations of an object and are often a concern in power transmission systems using rotating shafts. The motor is compact and lightweight due to a 40° angle between shaft and cylinder barrel. A cross-Section of the motor is shown in Figure 4.5 [48][49].

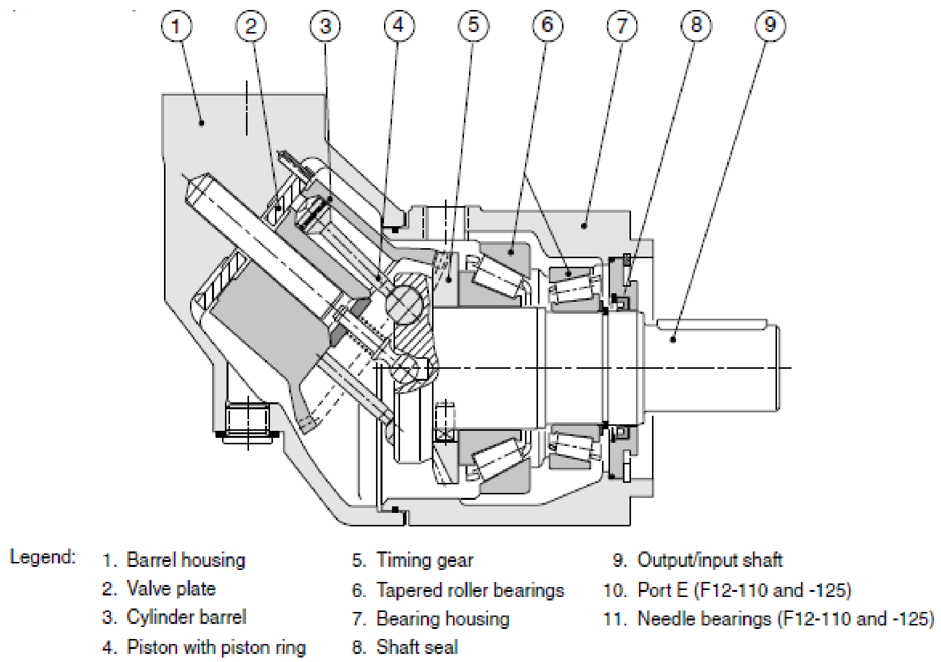

Figure 4-5 Parker F12 Hydraulic Motor [48]

In the specification for the F12-90 motor the capacity is given in displacement, cm^3/rev, which makes it necessary to calculate the corresponding flow, torque and power. In table 4.4, applied dimensions and given initial values are defined. The efficiencies η_v, η_{hm} and η_t are given relative to the shaft speed, pressure and type of hydraulic fluid. In general, the volumetric efficiency, η_v, increases with gained shaft speed and mechanical efficiency, η_{hm}, decreases under the same influence.

The variations in required torque of the applications generate a shifting differential pressure. The ratio between torque and differential pressure depends on the displacement capacity of the motor. An application with a large displacement is capable of generating a much higher torque than an application with a smaller displacement, and the ratio between torque and differential pressure is equally different in magnitude. The given maximum working pressure for the motor is 300 bars. Equations (4.2) are used for calculations of flow, torque and power.

Table 4.4 Applied Dimensions and Initial Values for Hydraulic Motor

Description	Ch.	Unit	Value
Displacement	D	[cm^3/rev]	93
Shaft speed	N	[RPM]	5000
Volumetric efficiency	η_v	[-]	0,98
Mechanical efficiency	η_{hm}	[-]	0,92
Overall efficiency	η_t	[-]	0,902
Differential pressure	Δp	[bar]	300

$$q = \frac{D \times n}{1000 \times \eta_v}$$

$$M = \frac{D \times \Delta q \times \eta_{hm}}{63} \tag{4.2}$$

$$P = \frac{q \times \Delta p \times \eta_t}{600}$$

Using the equations and the knowledge of the ratio between torque and displacement, the values in table 4.5 is obtained.

Table 4.5 Flow, Torque and Power

Description	Ch.	Unit	Value
Flow	Q	[l/min]	475
Torque	M	[Nm]	407
Power	P	[kW]	214

Figure 4.6 shows a picture of a hydraulic motor fixed on one of the slewing gears of a similar crane to the 50T AHC.

Figure 4-6 Hydraulic Motor Fixed on a Slewing Gear

4.2.3 Slewing Gear

The slew bearing is equipped with four slewing gears of the type RE2523 from Dinamic Oil where ‘3’ indicates that the pinion is categorized under stage 3. The stages refer to the exchanging ratio of the gear and higher stages represent higher ratios. It is important that it is consistency between the ratio and the strength of the pinion. A pinion with a large ratio needs a strong construction. This is ensured by strength calculations early in the decision process. Figure 4.7 shows the positioning of the gears relative to the slewing wheel.

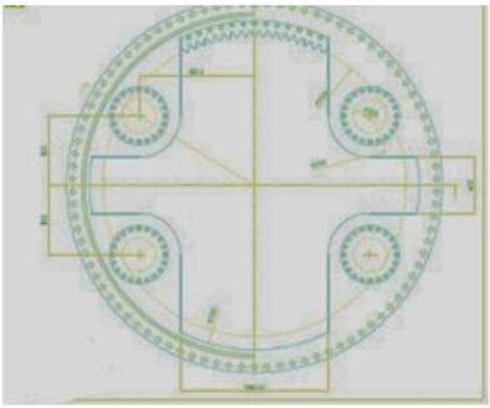

Figure 4-7 Positioning of Gears in the Slewing Ring [47]

The picture in Figure 4.8 displays the inside of the slew bearing of a similar crane to the 50T AHC. It is seen how the gear is paired with the slew bearing and that a hydraulic motor is positioned on top of the gear (barely visible). The motor is seen in Figure 4.6, Section 4.2.2.

Figure 4-8 Slewing Gear in Slewing Ring

The RE-series are gearboxes completed with enhanced output supports, integral pinion, a static negative disc brake and input coupling developed for hydraulic motors. The nominal output torque is available in a range from 4100 Nm to 60 000 Nm. For gear specifications, see table 4.6. The torque is given per pinion [45][46].

Table 4.6 Specifications of RE2523 DBS

Description	Unit	Value
Number	[-]	4
Number of teeth	[-]	14
Max dyn. torque	[Nm]	41 420
Max stat. torque	[Nm]	51 780
Swing torque	[tm]	150
Turning rate	[rev/min]	1.2
Ratio	[-]	128,7

In terms of installation of the gear system, the backlash has to be adjusted. The point of maximum deviation of the pitch circle from circularity is marked by three teeth colored green. The backlash is adjusted in this specific point. An alternative is to control each drive after the final assembly. It is expected that the backlash at the adjusting point is minimal 0.6 mm.

4.2.4. Encoder

The system is equipped with encoders in every joint. The encoders give feedback on the actual position of the different crane parts. In this way it is possible to compare the exact and true position to the intended. The error between the positions coordinates is caused by several factors, among these is the effect of backlash.

It is planned to install the new and improved all-electric drive system with the same encoders used for the hydraulically powered crane. See Section 5.3.6 for the function requirements of the encoder that is to be installed at the slew bearing.

4.3 All-electric Drive System

In this Section the chosen electric motor drive is briefly presented. The selection of electric drive is based on the calculations in Section 4.2 and the reasons stated in Chapter 3.

The chosen solution for the electric drive and motor, and how it is planned to operate and control the system drive is carefully reviewed in in Chapter 5. Figure 4.9 is a short introduction to this Chapter. The slew bearing is the mechanical rotational element that is to rotate the entire crane. The slewing ring will help move the load to the desired horizontal position and is intended equipped according to the Figure below. As for the existing hydraulic system, the slew bearing is to rotate by using four asynchronous motors powering each of the planetary gears. The main difference is the implementation of the frequency converter and the control system, in addition to the operating principle in used.

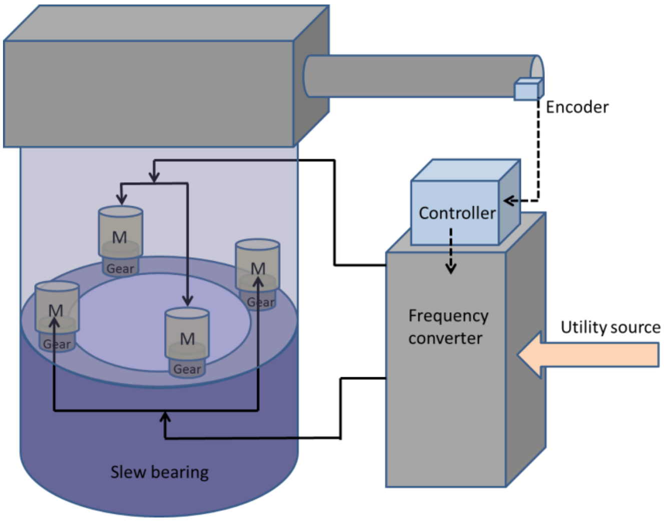

Figure 4-9 Illustration of Total System

4.3.1 AC Motor

The selected AC motor is a M3BP 200 MLC delivered by ABB with a rated output power of 45 kW. See appendix A for DriveSize report. The report is a confirmation that all necessary calculations are performed, and thus meets the claims of objectives 3 and 5.

The motor is a totally enclosed, three phase squirrel cage type and is designed to fulfill the IEC and EN standards. With IP class 56 the motor has the provided degree of protection. Motors conforming to other standards and sizes are available from ABB on request. This satisfies the requirements imposed in objective 3, 5 and the design criteria, and ensures scalability and portability. Table 4.7 gives the essential quantities of the AC motor.

Table 4.7 Applied Dimensions AC Motor

Description	Ch.	Unit	Value
Rated Output Power	P_N	[kW]	45
Rated Voltage	U_N	[V]	660
Rated Frequency	f_N	[Hz]	60
Rated Speed	n_N	[r/min]	1783
Nominal Speed	T_N	[Nm]	241
Moment of Inertia	J	[kgm^2]	0.366
Efficiency	η	[-]	0.944

In total four equal AC induction motors, shown in Figure 4.10, are used for the slewing of the crane. One motor is connected to each slewing gear. The applied swing gears are equal to those applied in the hydraulic system [50].

Figure 4-10 AC Motor from ABB [50]

4.3.2 Multi-drive

In this electric motor drive system an industrial multi-drive from ABB is selected. The multi-drive unit, see Figure 4.11, enables a single power entry as common supply for several drives. This comes in handy as this system actually is based on four separate drives, one for each gear, that form part of a single process.

Figure 4-11 Multi-Drive System [51]

The multi-drive supplied by ABB for this project comprises one rectifier unit and four inverter units.

The rectifying IGBT supply unit (ISU) provides the line-voltage in this multi-drive from ABB. Both motor and generator mode is achieved by the ISU and this is highly valuable as regenerative properties are requested. Specifications for the rectifier are given in table 4.8.

The inverters represent the drive units and are based on IGBTs. The inverters are to convert the temporary rectified link voltage (DC) into AC. The properties of the selected drive units in this multi-drive are specified in table 4.8.

Table 4.8 Properties of Selected Drive Unit

Description	**Rectifying supply unit**	**Inverter drive units**
Nominal voltage U_N	690 V	690 V
Nominal current $I_{CONT,MAX}$ (AC)	135 A (AC)	51 A (AC)
Nominal current $I_{CONT,MAX}$ (DC)	164 A (DC)	-
Nominal current I_{MAX}	245 A (DC)	68 A
Nominal apparent power S	161 kVA	-
No-overload use $P_{CONT,MAX}$	160 kW (DC)	45 kW
Light-overload use I_N	157 A (DC)	49 A
Light-overload use P_N	153 kW (DC)	37 kW
Heavy-duty use I_{HD}	122 A (DC)	34 A
Heavy-duty use P_{HD}	119 kW (DC)	30 kW
Heat dissipation	5,2 kW	0,8 kW

Chapter 5

Solution

The first two sections of this chapter mainly focus on the development process and the different alternatives for the solution design. Based on the choices made for the solution, there are several requirements to account for. Concerning the motor drive it is important to use structured design procedures, as it may be difficult to outline a good and unique design. The control system and drive for this particular configuration, in terms of design, mathematic modeling and block diagrams, are discussed and developed from Section 5.4 and on.

The arrangement of this chapter shows that the focus has been to design the drive system in a modular manner according to the specified design criteria in Chapter 1 and the methods for modular design and generic work described in Chapter 3. Each of the module blocks in Section 2.4.1 is now presented as separate sections in this chapter. The chosen propositions for each individual part are not dependent on each other. If necessary, and to ensure scalability and portability, the mechanical modeling, the electrical components and the proposed control scheme can easily be replaced independently. The project group can only think of one thing that especially must be taken into account. The properties of the different components and other involved parameters might need to be scaled up or down to match any new conditions.

5.1 Development Process

The design procedure for constructing electric motor drives and control systems are preferred by many to be analytical and should include the following steps to ensure a functional system:

- Performance specifications
- Conceptual design
- Mathematical modeling
- Model validation and parameter identification
- Analysis of the mathematical model
- Modification and iteration
- Constructing and testing

An essential part of this project is to model and examine the backlash in the connection between the slew bearing and the planetary gears. A model accounting for both the electric drive as well as the mechanical system must be constructed to achieve a good representation of the backlash. The model chosen to simulate the system and solve the problems is crucial for the resulting solution and the verification of the hypothesis. The result is highly dependent of the modeling and operation of the complete drive system.

5.1.1 Two-mass System Modeling

The first ideas for a backlash reducing system were to model the drive as a two-mass system; the motors represent one of the masses, whilst the load (including the slew bearing) is the other. The two masses are connected by the rotating gears and the backlash is a nonlinear element occurring when power is transmitted from one mass to the other. In this two-mass system all motors will accelerate, decelerate or maintain constant speed in the same direction at any time. The location of the motors and gears is as stated in Section 4.2 and the symmetric positioning remains the same throughout the different solution models tested/examined.

The scope of this project is to reduce the backlash effect between gears and slew bearing. It is therefore important to exhibit how the all-electric motor drive influences the preciseness of the slewing. The backlash effect on the system must be simulated and the following are some of the backlash models that were considered for the two-mass system:

- Deadzone model
- Exact backlash model
- Describing functions method
- Hysteresis model
- Simple backlash model

Without entering the constructing details of the above mentioned models, it is concluded that all models are rejected. Building and implementing the models in the motor drive is too advanced and time-consuming. Mainly due to the limitations of Chapter 1 it was decided to skip further examinations and attempts to compose own advanced models for the backlash in the slew bearing. It was agreed to use a prebuilt backlash block from the SIMULINK Library.

However, this simplification yielded the modeling of the backlash phenomena only. The remaining issue was to reduce and/or remove the backlash effect from the system using the electric motor drive. The model structure of the electrical and mechanical sides of the two-mass system had to be optimized in relation to each other. Parallel to the development of the backlash model, it was concluded that a solution based on a three-mass-system best would reduce the backlash effect. The conclusion was a well-designed three-mass system with a master / slave configuration would solve the problem more elegant than the two-mass system. In an optimal design of the master / slave technique, one or more of the motors is always in contact with the slew bearing and prevents the backlash from affecting the system.

5.1.2 Three-mass System Modeling

The second and chosen scenario for the solution approach is the three-mass system, explained in details in Section 5.2. Using a three-mass system design also means exploiting the principles of master / slave operation technique. Initially it was decided to use a master / slave motor combination with a ratio 3:1.

- 3:1 master / slave combination
  For this scenario, the four motors (each connected to a gear) should provide a total of 120 kW, e.g. 30 kW each. According to performed calculations this amount of power is sufficient for the three-mass system with a 3:1 setup. The 3:1 configuration is an

excellent choice of solution since the negative effect of the backlash is removed as long as one motor is in contact with the slew bearing.

The first version of the three-mass system with a ratio of 3:1 was declined in favor of a ratio of 2:2. This was solely in order to simplify the simulations. The final solution for the three-mass modeling was chosen to be the 2:2 master / slave setup. See Figure 5.0.

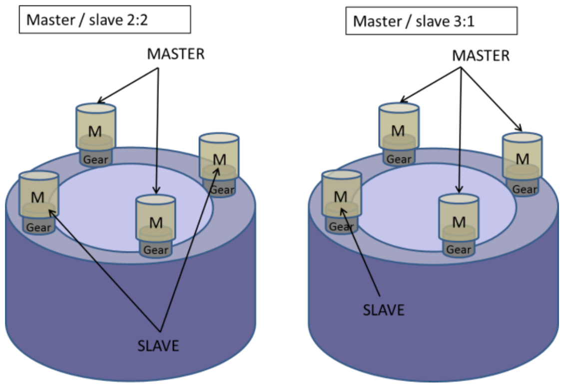

Figure 5-0 Master / Slave Solution with ratio 2:2 versus 3:1

- 2:2 master / slave combination
  To ease the simulations and keep a simple symmetry, it was chosen to model the three-mass system with a 2:2 motor configuration. If the pair of motors is to counteract, this would create an even behavior with respect to applied force and symmetry, as well as a simpler SIMULINK model. Still, using 2:2 instead of 3:1 will provide the same effect of counteracting motors, e.g. torque of opposite polarity to cancel out the backlash gap. This results in minimized oscillations due to backlash at the start and stop sequences of the slew bearing. With a 2:2 solution for the master / slave setup it was assessed that the motor size should be increased. The increase had to be in such matter that the two master motors would have a force equal to three motors in the 3:1 setup. It was therefore selected a motor size of 45 kW for this configuration. The specifications for the motor meet the requirements in the first part of OBJ. 2. The slave motors are only to provide an opposite torque during acceleration or deceleration and must in these cases contribute a smaller torque than the master motors. As the 2:2 master / slave technique is chosen as final solution, the procedure, model, parameters

and results are exhibited and discussed in Section 5.2. This Section also provides an explanation of how the system is intended to operate and more important; how it affects and reduces the effect of backlash.

5.2 Simulations

There is not found any documentation of master / slave technique used in the slewing of cranes earlier. The technique is earlier applied in several applications, such as laser, water and plasma jet cutting. In these applications there are no needs for fast direction changes. However, for the crane subject for this report, fast and precise direction changes of the slewing are an important property. Figure 5.1 shows the presence of backlash gap relative to driving direction. When the crane switches its rotational direction, the gap designated as *'Gap on lagging side'* occurs in front of the motion, on the leading side. This issue complicates the implementation of the master / slave system for this particular application.

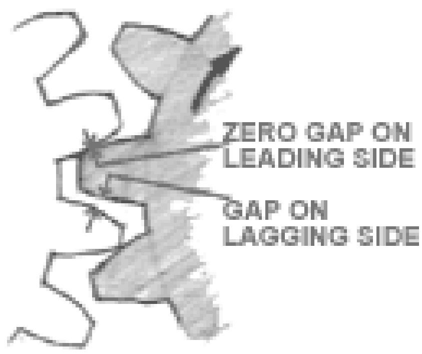

Figure 5-1 Backlash Gap Relative to Driving Direction [56]

5.2.1 The Main SIMULINK Model

The theoretical results are examined using SIMULINK for modeling and simulations. A three-mass system consisting of two motors and one load is composed. The parameters are adjusted to correspond with the actual values from the 50T AHC crane being the subject for this master thesis. As the chosen solution utilizes a master / slave operating strategy two equal motors acting on the load are integrated.

The SIMULINK model is shown in Figure 5.2.

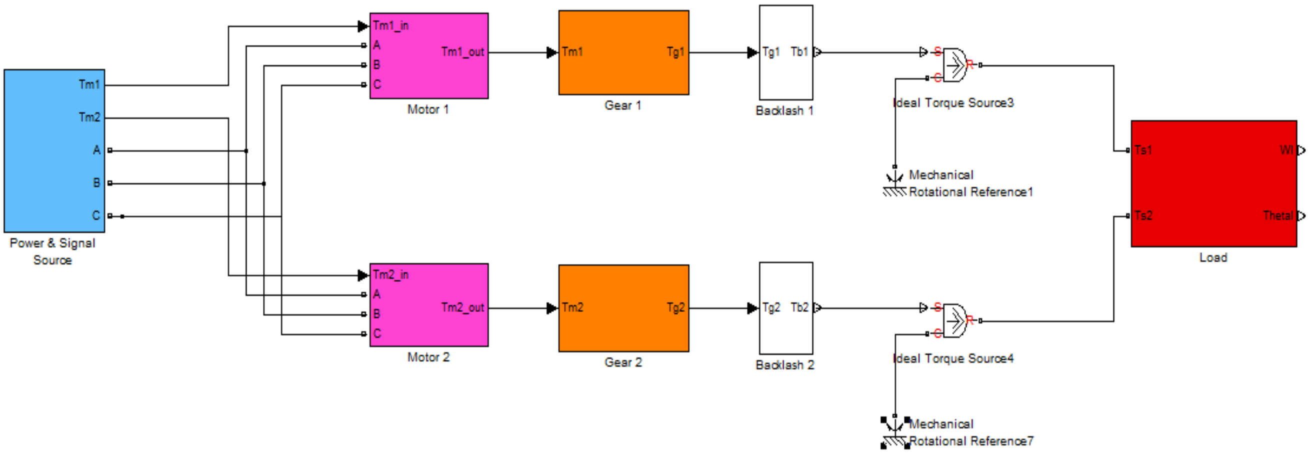

Figure 5-1 Three-mass System in SIMULINK

As stated in 3.2.2 it is decided to apply a master / slave driving technique to reduce the impact of backlash on the system. In light of analysis and testing of the technique, some modifications relative to the presentation in Chapter 3 are done. The crane has the property of changing the direction of motion. This creates the need for the motors to switch between master and slave operation mode. When the crane reaches the final position, the backlash gap is avoided due to the spring and break effect from the opposite directed torque created by the slave motors. When changing direction of the crane motion after a stop, the backlash gap changes position to the opposite side of the sprockets, see Figure 5.1. To counteract this behavior, the motors that until now acted as masters have to change to the properties of slave motors and counteract the main driving direction. See Figure 5.3. In this way the occurrence of backlash gap in any driving direction is controlled and effectively reduced and/or removed.

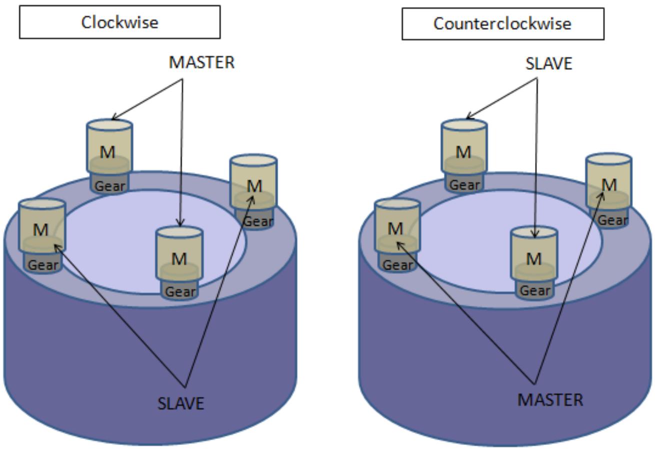

Figure 5-2 Definition of Master and Slave Motors in both Driving Directions

Figure 5.4.a is a draft of the applied torques from motor 1, M1, and motor 2, M2 (M1 and M2 representing two motors each). The sign of the torque depends on the direction of motion and the distance to endpoint. Figure 5.4.b shows the torque acting on the load and Figure 5.4.c is the corresponding load motion. The motion is defined as positive when directed clockwise and negative if acting in counterclockwise direction.

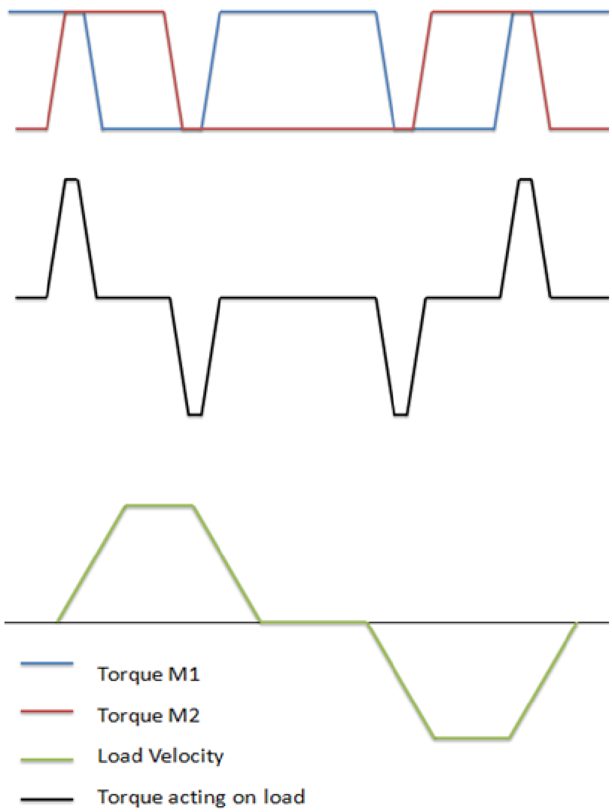

Figure 5-3 a) Applied Torque M1 and M2, b) Torque Acting on Load and c) Corresponding Load Velocity

The sign, υ, of the applied torque are decided by the relative error, e_{abs}, as described in 3.2.2. Depending on υ, an input torque, T_{mi}, is given to the motors. In earlier research, this method is proven to reduce the backlash impact smoothly and effectively, and is a good choice for practical applications. Since e_{abs} only depends on the relative position error, measurement of position angle for motor- and load side becomes unnecessary.

In the SIMULINK model the control structure of the input torque is replaced by the signal presented in Figure 5.4.a. This simplification is justified since the control design is regarded as a minor point compared to the analysis of the system performance during backlash influence. The structure of the signal is based on requirements given by HE. The request included a simulation of position change from an initial point to a given end position. In the end position, the motor is supposed to stop, change direction and drive back to the start position. Linear change in torque gives a smooth acceleration and deceleration effect and provides stability to the system.

5.2.2 Backlash Modeling

The backlash is represented by a prebuilt SIMULINK block. The block implements a change in output caused by a change in input. The change depends on the side-to-side-play in the system and the magnitude of the play is defined as the deadband. The deadband is set to 10 % of reference input. A relatively large deadband value is defined to emphasize the impact of the backlash. It also clarifies the improvement by introducing the master / slave driving technique. In 3.2.2 the dead zone model was introduced. The backlash block in SIMULINK uses the model as basis for its mode of operation and is therefore a good approximation of the backlash effect. Due to time limitations it is decided to take advantage the prebuilt block. A schematic presentation of the three-mass system with backlash is given in Figure 5.5. The G(s) block represents the load and the two loops with backlash elements represent motors and gears. The impact when the backlash gap closes is assumed to be inelastic.

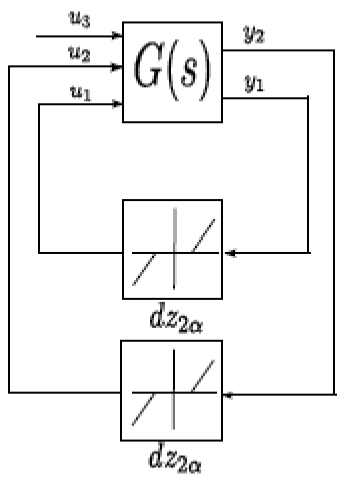

Figure 5-4 Schematic Presentation of the Three-mass System with Backlash

5.2.3 Gear Modeling

The gears are modeled using an ideal gearbox available in SIMULINK. The first block represents the gear ratio of 128.7. To account for energy dissipation and non-elastic effects, two gear mesh elements are introduced. The internal parameters in the mesh blocks are calculated on basis on the physical characteristics of the gear units. To represent the true stiffness of the system, two mesh elements for each motor is needed. The gear box referred to as *Pinion Rim Ratio Line 1*is an ideal 108/14 gearing between pinion and rim (slew bearing). Figure 5.6 shows the SIMULINK model of the gearing.

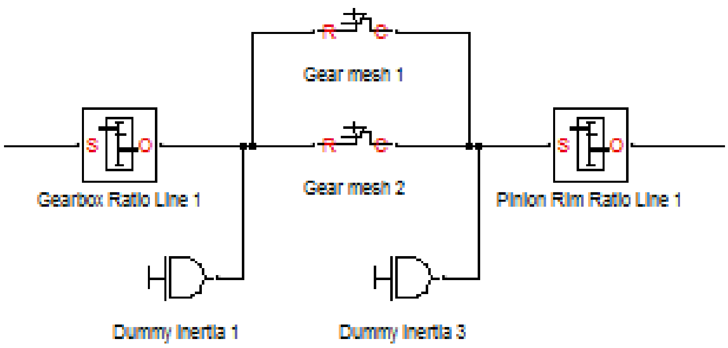

Figure 5-5 SIMULINK Model of Gear System

5.2.4 Load Modeling

The load consists of both the slew bearing and the specific loading. Further this is referred to as the common "load" and is modeled according to Figure 5.7.

The load is divided into two parts; the lower and upper part of the crane tower. The tower receives the rotational energy and to simplify, the tower is given the inertia of the total mass to rotate. The inertia of the load is collected from Inventor at the working radius of 10 m. The lower part of the tower contains approximately 25 % of total inertia, whilst the upper part contains approximately 75 %.

For the system to experience a steady state resistance, friction and damping must be included in the slew bearing. These elements ensure losses within the system and provide steady state. The total loss due to friction and damping constitutes the term T_d, specified by equations in Section 3.2.2.

Regarding the torsional stiffness and damping coefficient of the tower, these values are calculated by means of a simple torsional analysis when considering the crane tower as a beam with a circular base.

All coefficient parameters should be subject for further tuning when the complete model is calibrated to best match the actual construction and operation of the crane.

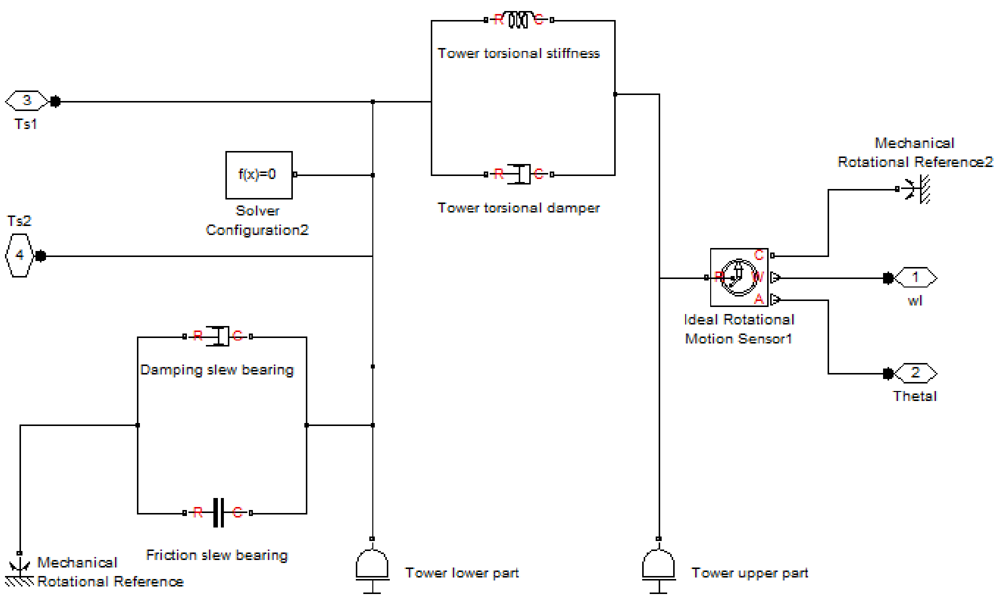

Figure 5-6 Load Model in SIMULINK

5.3 Solution Components for the Electric Drive

This Section is an examination of the selected system components. The result is a SIMULINK-model of the final drive (see Section 5.3.5), which in the end consists of the DTC, the frequency converter, the induction motor and various other required electronics. Generally, the arrangement of this Section is in accordance with the circuit in Figure 5.8.

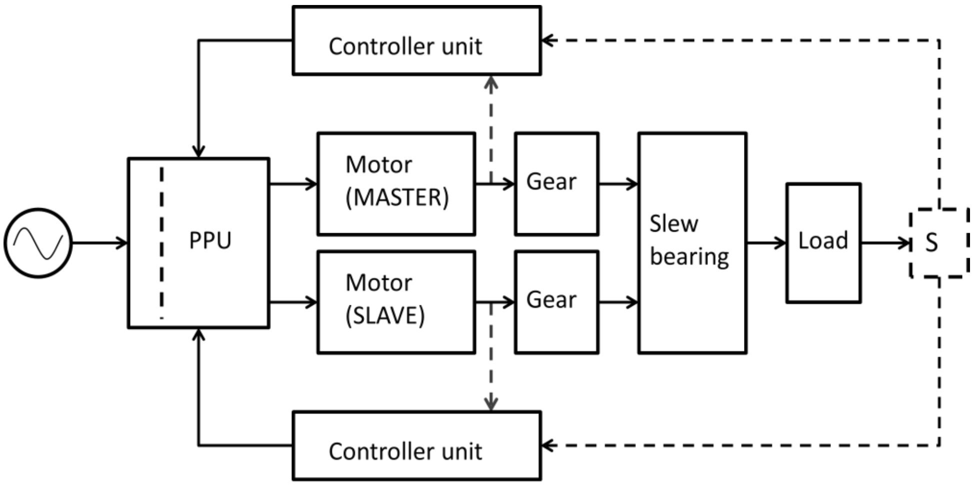

Figure 5-7 Model of Final Drive Assembly

5.3.1 Utility Source

A transformer is included as part of the power supply, see dashed line in the PPU in Figure 5.8. As the electric drive is placed on board an offshore vessel, a transformer is a necessary component. The transformer steps down the supplied voltage, in addition to providing the electrical (galvanic) isolation that is required for offshore conditions. A dry isolation transformer with conditions outlined in table 5.1 is included in the circuit to provide the necessary power [52][53].

Table 5.1 Transformer Conditions

Description	**Unit/type**
Transformer	DTE 400 A8S
Network short circuit power	200 MVA
Primary voltage	21 000 V
Secondary voltage	660 V
Frequency	60 Hz
Apparent power S_N	400 kVA
Impedance Z_K	6 %

5.3.2 The Power Processing Unit

The chosen converter is a multi-drive from ABB. Figure 5.9 shows the multi-drive specified for this project, as the separate drives provide each gear with mechanical power.

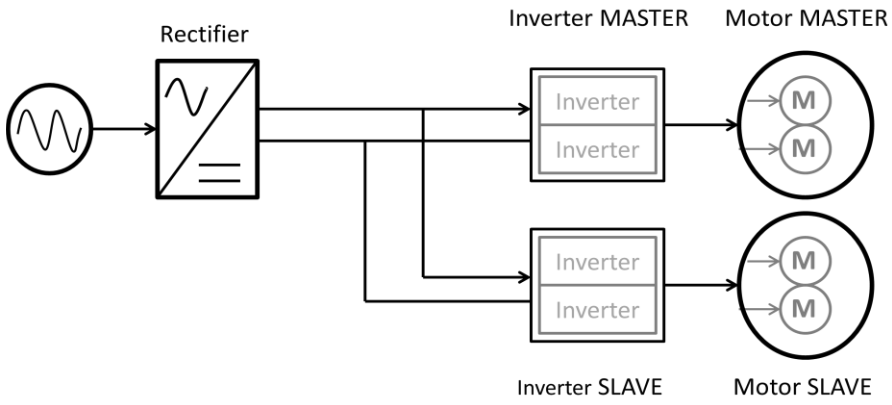

Figure 5-8 Multi-Drive

The converter consists of a phase-controlled rectifier and two main inverter/motor-configurations (keep in mind the 2:2 master / slave setup), all based on IGBT technology. As this regenerative converter operates in all four quadrants, the multi-drive contains no separate braking unit and the excessive energy is stored in a capacitor at the DC-link.

A rectifying IGBT supply unit (ISU) provides the line-voltage in the multi-drive from ABB. Both motor and generator mode is achieved by the ISU and this is highly valuable as regenerative properties are requested. Specifications for the rectifier are given in table 5.2.

The inverters, four in total, represent the drive units and are based on IGBTs. The inverters are to convert the temporary rectified link voltage (DC) into AC. The properties of the selected inverters in this multi-drive are specified in table 5.2.

Table 5.2 Specifications for Rectifier and Inverter Units

Description	Rectifying supply unit	Inverter drive units
Nominal voltage U_N	690 V	690 V
Nominal current $I_{CONT,MAX}$ (AC)	135 A (AC)	51 A (AC)
Nominal current $I_{CONT,MAX}$ (DC)	164 A (DC)	-
Nominal current I_{MAX}	245 A (DC)	68 A
Nominal apparent power S	161 kVA	-
No-overload use $P_{CONT,MAX}$	160 kW (DC)	45 kW
Light-overload use I_N	157 A (DC)	49 A
Light-overload use P_N	153 kW (DC)	37 kW
Heavy-duty use I_{HD}	122 A (DC)	34 A
Heavy-duty use P_{HD}	119 kW (DC)	30 kW
Heat dissipation	5,2 kW	0,8 kW

Rectifier

The supply unit is a phase-controlled rectifier built up of IGBT's. The reason for implementing transistors is to ensure a regenerative effect in addition to the motoring mode. In this project it is chosen a system design where both acceleration and deceleration are requested and this is the most suited solution for the rectifier-part of the converter.

When turning the rectifier on and off at a various number of sequences, errors in the PF variations and high levels of harmonics are produced in the power to the DC side. The highest power quality is assumed to be when the PF lags the most and the total harmonic current distortion (TDD) is the lowest. A reduction in the switching frequency ripple is achieved by installing a line inductor at the AC input and a harmonic filter at the DC-link.

Inductor

It is in all simplicity chosen a basic L-filter as the AC-side inductor. The design of the filter is not essential for verification of the hypothesis of backlash reduction, so a simplification is performed to save time.

To ease the design procedure, it is customary to set inductors to approximately 5 % of the rated line inductance. The three-phase base inductance of this electrical utility is given by (5.1):

$$L_b = \frac{V_b^2}{S_b * \omega_n} \tag{5.1}$$

Where

L_b Base inductance

With line-to-line base voltage of 660 V, apparent power S of 400 kVA and line frequency of 60 Hz, it is obtained an approximated base inductance of 2900 μH according to (5.1). If choosing an estimation of 5 %, the filter inductance becomes roughly 145 μH.

Capacitor

The capacitor at the DC-link handles the remaining disturbances in the circuit, ensuring good power quality. When placing a capacitor at the DC-link, the drive becomes a voltage source drive and resonance is introduced to the circuit. As a restriction to the capacitor size, the resonant frequency should be less than half the switching frequency, yet ten times greater than the line frequency. The capacitor, as well as the inductor, should be adjusted when tuning the complete electrical circuit. The capacitor is initially set to 15 μF in this case. However, this must definitely be increased when tuning is performed. It may also be relevant to place multiple small capacitors in parallel in order to achieve a lower electric series resistance (ESR). This will better manage the ripple current. In either case, it is important that the final capacitor is a high-quality component with sufficient voltage ratings.

Inverter

The system design is divided into two main inverter drives; one separate inverter drive for the master motors and one for the slave motors. The master / slave operation is discussed in Section 3.2.2. The two motors and associated inverters, for both master and slave configurations, are combined and represented using a single inverter and a single motor and scaled up to match the total power.

Both master and slave inverter are based on IGBT technology. The inverters contain transistors and the switching control is as described for the rectifier. Since the control circuit is not, and will not be, completely outlined, the gate signal for the inverter is not optimal. If good control is crucial and the control system is emphasized to a greater extent, the signal should be tuned. Due to limitations set in Chapter 1, tuning and further design of the control system is neglected and it is simply displayed how the backlash of the slew bearing is affected. For this reason, the signals into the converter (yielding both the rectifier and the inverters) are manipulated in relation to the requested torque output. See Section 5.2.

Snubber circuits estimation

When simulating the chosen PPU in SIMULINK, a number of snubber characteristics for the associated IGBT's are requested as input for the converter. Values to be particularly aware of are the snubber capacitance and resistance. Initially, these values were not available to the public and solely for internal use within the ABB group. However, the project group came to a special agreement with ABB and the data sheets for the relevant IGBT's were handed over. Technical values are collected from the product data sheet for the different IGBTs involved and the required snubber values are reproduced in table 5.3.

Table 5.3 Technical Data and Specifications for IGBT Rectifier and Inverter

Description	Parameter	Rectifier IGBTs (FS300R17KE3)	Inverter IGBTs for master & slave motors (SKM145GB176DN)
Rail voltage	V_{rail}	1700 V	1700 V
Snubber capacitance	C_s	247 nF	3,8 nF
Snubber resistance	R_s	40 Ω	2656 Ω
Internal resistance	R_{on}	2,5 Ω	0,01 Ω
Forward voltage device	V_f	2 V	5,8 V
Forward voltage diode	V_{fd}	1,8 V	1,6 V
Fall time	t_f	0,18 µs	0,145 µs
Tail time	t_t	0,63 µs	0,485 µs

5.3.3 Switching

This Section concerns the switching of the converter. However, as the task description clearly states that the control circuit is not to be developed, only sketched, this Section does not give a detailed explanation of the converter switching. Due to this limitation, the necessary switching signals for the converter are missing. As the signals are not generated in correct manner, it will simply be suggested a switching scheme appropriate for the situation.

To illustrate the proposed switching and ensure an optimal design of the converter circuit, this Section also reviews necessary filters and other passive components.

In short, the planned design starts with filtering the utility of most of its disturbances. By means of an inductor at the AC-side, the source is smoothened before entering the rectifier. After the rectifier, at the DC-bus, a capacitor attenuates noise by absorbing the energy and smoothens the output even more. See Figure 5.10.

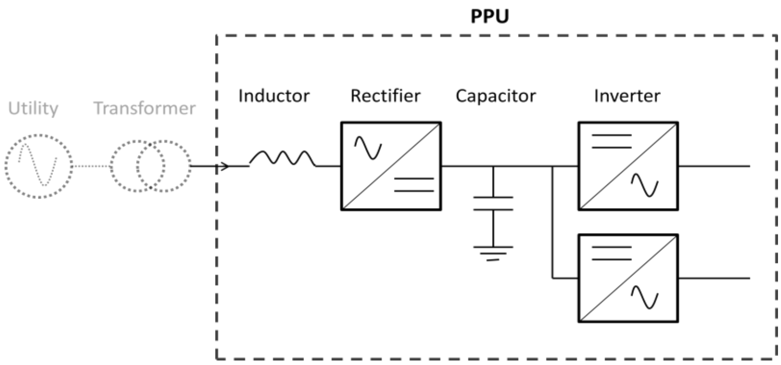

Figure 5-9 PPU with filtering components

To ensure sufficient voltage and torque to the motor/load, it is planned to turn the switches of the converter on and off by means of pulses from the controller. Thus, the decision is to propose a pulse-width-modulated (PWM) switching scheme.

Switching topology

The transistor switches of this converter are controlled by signals from the controller unit.

The switching is performed by conventional hard-switched PWM. In PWM the voltage across and the current through the IGBT is abruptly connected or disconnected. Hard switching stresses the components and results in a certain amount of switching losses, depending on design values and the switching frequency. Ringing is another problem associated with stress and high frequency noise. Ringing is a significant problem occurring at each wave and contributes to additional stresses and oscillations to the load. See Figure 5.11.

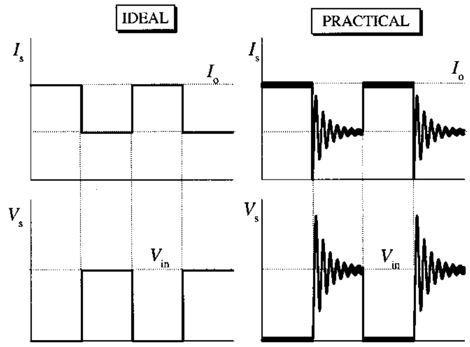

Figure 5-10 Ringing Effect Contributing to Stress and Oscillations on the Load [54]

As explained in Chapter 1, the design procedure of the drive system is to follow certain guidelines and restrictions in order to complete on schedule and present a satisfactory solution. It is therefore selected a hard-switched PWM topology for the frequency converter, despite its drawbacks. By choosing the simpler PWM, the controller design is easier executed and it also makes sure that an early-draft proposal is provided within this project. It is emphasized that in the event of the realization of this project the control system must be subject for continued design for optimal results.

The hard-switched PWM converter circuit is as depicted in Figure 5.12. Keep in mind Figure 5.10. The schematic diagram is extended by the complete schematic of both the rectifier and the two inverters.

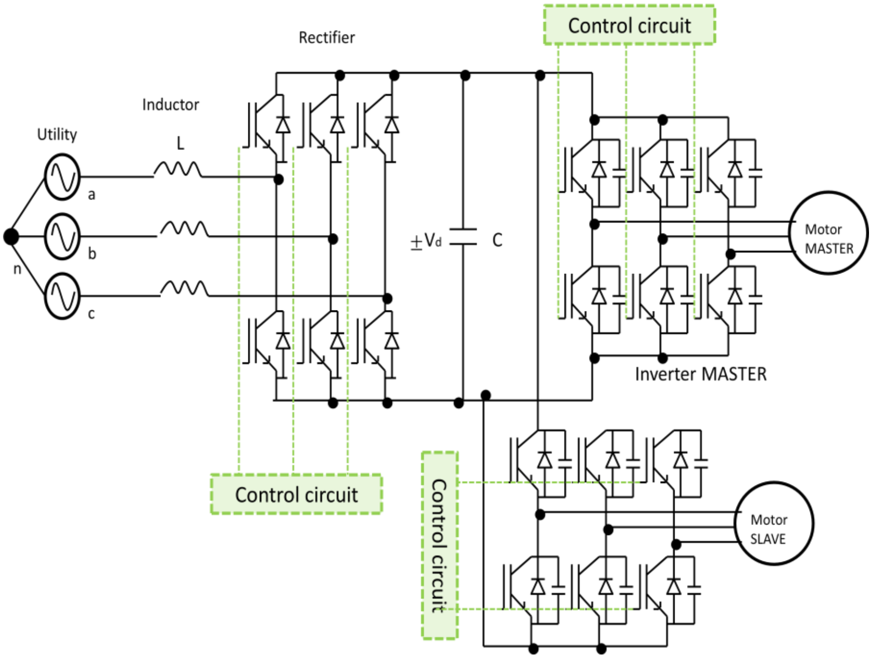

Figure 5-11 Hard-Switched PWM Converter Circuit

5.3.4 The AC Induction Motor

The asynchronous machine is available as a pre-built model in SIMULINK. The block operates either in motor or generation mode, depending on the sign of the mechanical torque. Positive T_m yields motor operation, whereas a negative T_m indicates a regenerative mode of operation. The pre-built block, containing several subsystems and masks, requires a set of clearly defined parameters. In addition, the mechanical torque T_m or motor speed ω_m is asked as direct input to the block. The output of the block is a vector containing 21 different signals.

The three-mass system with master / slave ratio of 2:2 includes four 45kW motors.

5.3.5 The Complete Motor Drive

Figure 5.13 displays the final proposal for motor drive and motor. Each of the two inverters (four units) and associated motor are connected to a common rectifier unit. The controller units in Figure 5.13 demand, as mentioned, additional design and tuning for optimal functions. The subsystems of the controller blocks in Figure 5.13 are shown in details in Figure 5.14 and correspond to the outlined control system proposed in Section 3.5.

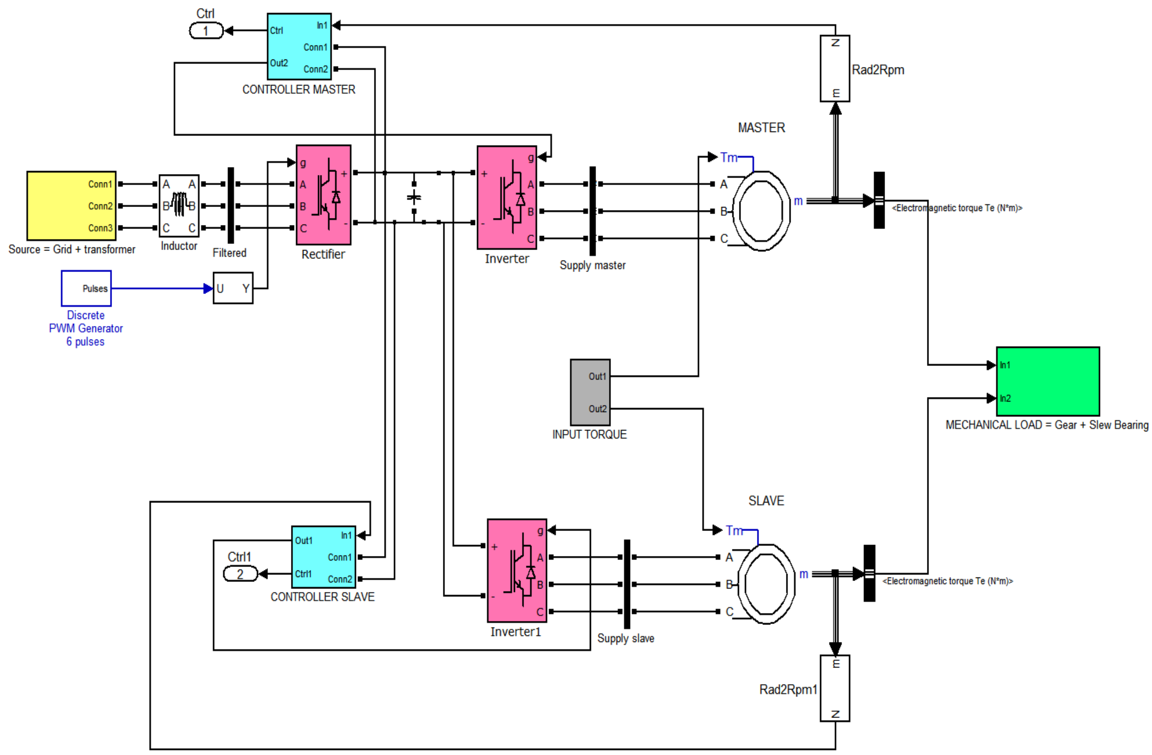

Figure 5-12 Final Solution for Motor Drive and Motor

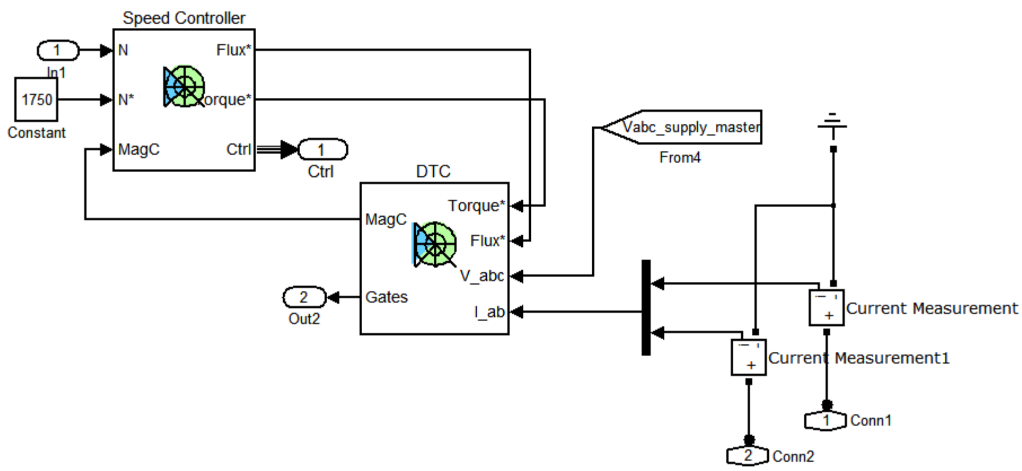

Figure 5-13 Controller

5.3.6 Encoder

Encoders are presumed to be present at each of the crane's joints to keep track of the complete movement of the crane.

An encoder at the slew bearing is installed to monitor its position, in order to maintain an accurate slew. The selected encoder continuously measures the position and motion of the slew bearing relative to its origin/initial position.

As it appears in the discussion of Chapter 7, future design of the solution is proposed to contain an encoder for speed measurements. However, the selected solution currently consist a position encoder and is the basis for the rest of this Chapter. The reason for choosing a position encoder is that previous research shows that this solution gives the best outcome.

Reference values for position are used to generate the proposed switching signal for the master / slave configuration. See Section 5.4.2. The master / slave setup is programmed to turn on when the backlash gap is within a certain range relative to an endpoint reference. It is thus critical with precise data of the position. The solution for the all-electric system retains the previous selected encoders for the hydraulic driven 50T AHC crane. Working principles of encoders are described in Section 4.2.4.

The encoder is to register the position of the slew bearing. When the position is measured, it is sent as feedback to the control circuit. The control circuit, if completely developed (not executed in this project), compares the exact position to the requested position, before supplying signals of more or less power to the motor in order to correct the deviation. If the master/slave-system is to function optimally, there must be active encoders to indicate where the slew bearing is positioned at all times. This is mainly to specify when the separate master and slave motors are to provide contributing or counteracting torque in relation to the main driving direction.

Since the task of the project only includes the proposing of a control circuit, the motors achieve pre-defined input signals. The output torque must be of the necessary magnitude to rotate the slew bearing according to a specific path. As determined, the slew bearing should start the rotational movement at 0°, accelerate to a top-speed of 1 rpm while rotation towards the desired position of 45° (positive direction is defined to the left). When approaching the final position, the slew bearing is to decelerate and come to a stand-still at the exact required position. After reaching 45° the slew bearing should rotate back to the original position. To achieve this in practice, active encoders have to constantly give feedback about the location (angular position) of the slew bearing. From this the motors are told when to accelerate, decelerate or maintain constant speed. In reality it is important to handle and process the obtained position data and create input signals. The input signals are generated according to the process described in Section 5.2. If the entire system was developed as explained in Section 5.4, the control circuit and frequency converter would receive the appropriate and correct signals needed to drive the motors and slew bearing optimally in practice. The scope of the simulations is to show the master / slave configuration influences of the backlash effect and thus encoder measurements are not exploited and assessed any further. By using the desired path of motion, the input signals to the motor drive are manipulated and manually constructed before applied to the control circuit and frequency converter. This is further described in Section 5.4 [31] [55].

5.4 Control System

OBJ. 4 states that only a proposal of the control system is given as part of the solution. Key limitations clearly restrict the solution to only contain a sketch of the control circuit. With respect to step two in Section 5.1, the following Section outlines the structural arrangement of the control system. A simple conceptual, cascaded design is proposed.

When designing the controller it is important to determine the control variables. In this case the variables are torque and speed (from a position encoder), and the main controller thus consists of a torque and a speed loop. Additionally, there is a need for a separate controller for the master / slave setup for the backlash reduction. When the three-mass system was selected as the solution system the issue of the direction change of the motors became a concern. The control circuit must tell the two different drive systems when to act and counteract each other. A suggestion for this controller is briefly presented in Section 5.4.2.

5.4.1 Speed and Torque Control

The methods mostly used for torque control in adjustable speed drives of asynchronous motors are field-oriented control (FOC) and direct torque control (DTC).

In view of the control method for this asynchronous application, the control design is required to follow guidelines given by ABB. The chosen multi-drive from ABB uses DTC as motor control. Thus, DTC is the best approach to control this asynchronous motor drive.

The proposed control of the asynchronous drive is in accordance with Figure 5.15. The circuit is clearly divided into two fundamental Sections; the speed loop and the torque loop.

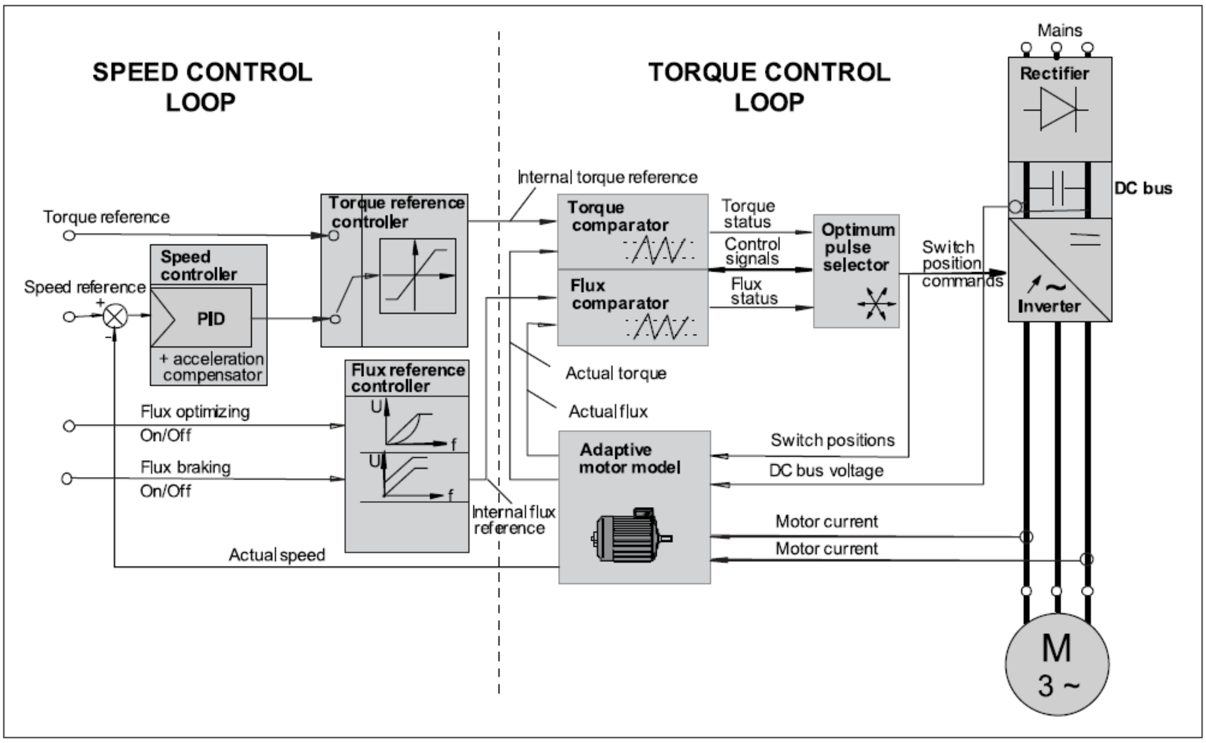

Figure 5-14 Sketch of the Propose Control of the Asynchronous Drive [55]

Starting with the innermost torque loop shown in Figure 5.15, the main actions are as follows: 1) Voltage and current measurements, 2) An adaptive motor model, 3) Torque and flux comparator and 4) Optimum pulse selector.

In step 1, phase currents, DC bus voltage and switch positions are measured while a comparison between actual and reference values for torque and flux are performed in step 3. The control signals for the transistors are produced in step 3. The task of step 2 is identification of motor parameters and tuning the drive. Step 4 utilizes the information from the preceding steps to determine the switch combination for the inverter.

The speed loop of Figure 5.15 is the outermost loop of the control system and consists of three main steps. Step 1 is the torque reference controller that bounds the speed control output by the limits for torque and DC bus voltage. The actual speed controller is step 2. In Figure 5.15, the speed controller is depicted as a PID-controller, whereas it in this case is decided sufficient with a PI-controller. The position from the encoder (see Section 5.3.6) is converted into speed and transferred to the PI-controller where error signals are produced by the difference in reference and actual values for the motor speed. Step 3 is the flux controller, which gives the flux comparator an exact reference value of the flux.

As advanced control is not a part of this project, the control parameters are not further tuned and the sketch in Figure 5.15 is considered to be the final solution for the control system.

5.4.2 Master / Slave Control

The master / slave controller must be designed to indicate when the motors are to switch from master to slave operational modes and vice versa. Clear references for the magnitude of the counteracting force must also be supplied. The amount of torque provided by the different motors is determined from encoder readings. Regardless of how the backlash is modeled, when the slew bearing approaches the requested position and the backlash is likely to occur, the motors must be told when to change directions and counteract each other in order to cancel out the backlash gap. The best solution to obtain a measurable backlash system is to implement a more precise and extensive model in SIMULINK, yielding the mechanical conditions for backlash. This gives the opportunity to collect necessary measurements to use as input signals for the master / slave controller unit. Although not performed, due to limitations stated in Chapter 1, it would be appropriate to use a dead zone model with describing functions for the backlash modeling of this project. This could detect the exact backlash angle. When the angle approaches a certain value, the control device for the master / slave operation achieves and processes these values. Next, the controller should specify the magnitude of the torque required in the opposite direction to counteract and ensure a minimized, preferable reduced, backlash gap. The signal from the master / slave control circuit must also set the condition of which motor pair that should act as master and which pair to act as slave. The motor pairs are also specified to swap the roles of master and slave operational mode. It is also necessary for the controller to specify the amount of power for the counteracting motors, as these must be far less than the master pair.

However, such a control system is beside the scope of this project and is not discussed any further. Nevertheless, the project group imagines that the control of such a system can be according to the proposal above. It is referred to Chapter 3 for the theoretical background of the backlash modeling.

The project group recommends that the suggested control scheme for master / slave, speed and torque controllers forms the basis of the control system. If an electric drive is selected for the operation of the slew bearing, the control scheme must be further developed.

Chapter 6

Results

This chapter presents the results from the system development and corresponding simulations. The simulations were carried out to verify the theory described in Chapter 3.

6.1 Final System

Figure 6.1 shows a block diagram of the proposed system design. The load represents the crane including the slew bearing. Both the electrical and the mechanical system are modelled in SIMULINK and simulated separately. In this situation, the electrical system is a collective term referring to the AC source, frequency converter, transformer and filters.

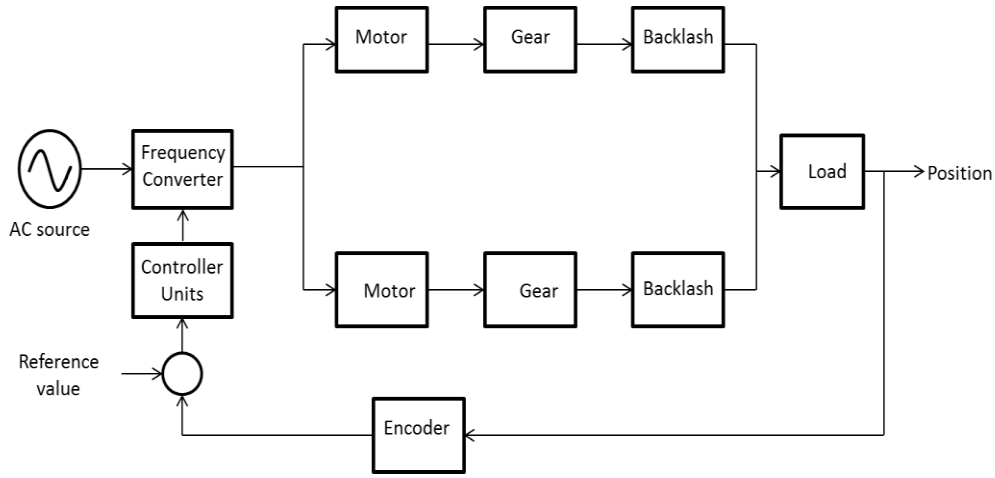

Figure 6-1 Block Diagram of Proposed System Design

Simulations showed that the electrical and the mechanical systems both functioned when simulated separately. The electrical system, where the frequency converter and controller units are the most important components, was proved to function and provided a suitable AC source for the motor.

However, due to a general deadline for the execution of simulations, the interconnection of the electrical and mechanical system is not completed. It was not found sufficient time to adjust the control, - drive and motor parameters as necessary. Consequently, it could not be done adequate measurements of angular position or velocity of the load for use in a feedback system for the controllers. This would close the control loop, as shown in Figure 6.1. Still, if connecting the two models and closing the loop, the results would be inaccurate. The complete closed loop shown in Figure 6.2 is thus only simulated as an open loop. It is nevertheless believed that Figure 6.2 shows a well-functional system design.

All further work is focused on the mechanical part represented by motors, gears, backlash elements and load. Additional, an electrical three-phase AC source is included as input to the motors.

The complete system design, based on the block diagram of Figure 6.1, is modeled in SIMULINK as shown in Figure 6.2.

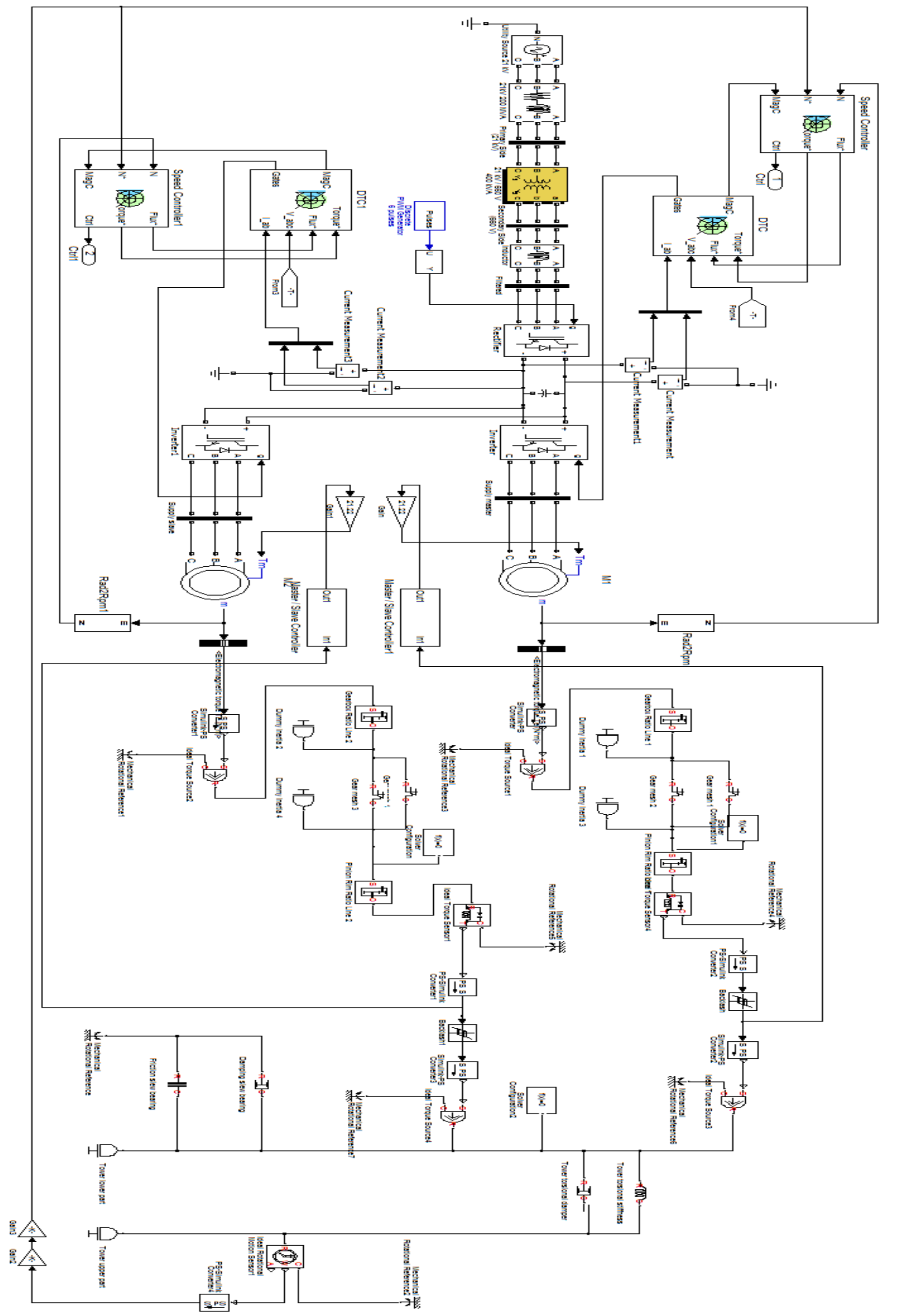

Figure 6-2 SIMULINK Model of the Complete System

6.2 Controller System

In accordance with OBJ.4, a sketch of the control system is proposed. A cascaded design with PI-controllers and a combination of a feed-forward and feedback system is recommended as solution. Torque and speed are chosen as control variables and are represented by their separate controllers in the cascaded structure shown in Figure 6.2. The proposed control system is outlined in Section 5.4 and also depicted in the complete drive assembly shown in figures in Section 5.3.5.

As mentioned in Section 6.1, the controller units are not connected to the mechanical system during simulations. If it was found time to connect the models and adjust and simulate a closed control loop, it is believed that the system is proved to work as intended and described in Section 5.4.

6.3 Electrical Components

According to OBJ. 2, a suitable electric motor and drive is selected. In order to meet the objective, the existing system is studied and multiple calculations are conducted. After completing this, several vendors are contacted. The final decision was ABB, as they presented the most complete package for the solution. During frequent correspondence between the project group and ABB throughout the working process, an adequate motor and a suitable drive system is selected.

The selected asynchronous motors have a power rating of 45 kW and a nominal voltage of 660 V. The efficiencies of the motors are set to 94,4 %. The variable frequency drive was chosen to match the size of the motors and consists of one single rectifier and four inverters. The rectifier has a nominal power of 160 kW and each of the inverters has the power rating of 45 kW. Nominal voltage rating for the converter is 690 V. The remaining components are as described in Chapter 5.

The execution of OBJ. 2 has been a time consuming process and a large amount of work is put down in order to ensure that the chosen option was the most appropriate for this application.

6.4 SIMULINK Results

The main simulations are executed with respect to the mechanical system represented by the block diagram in Figure 6.3.

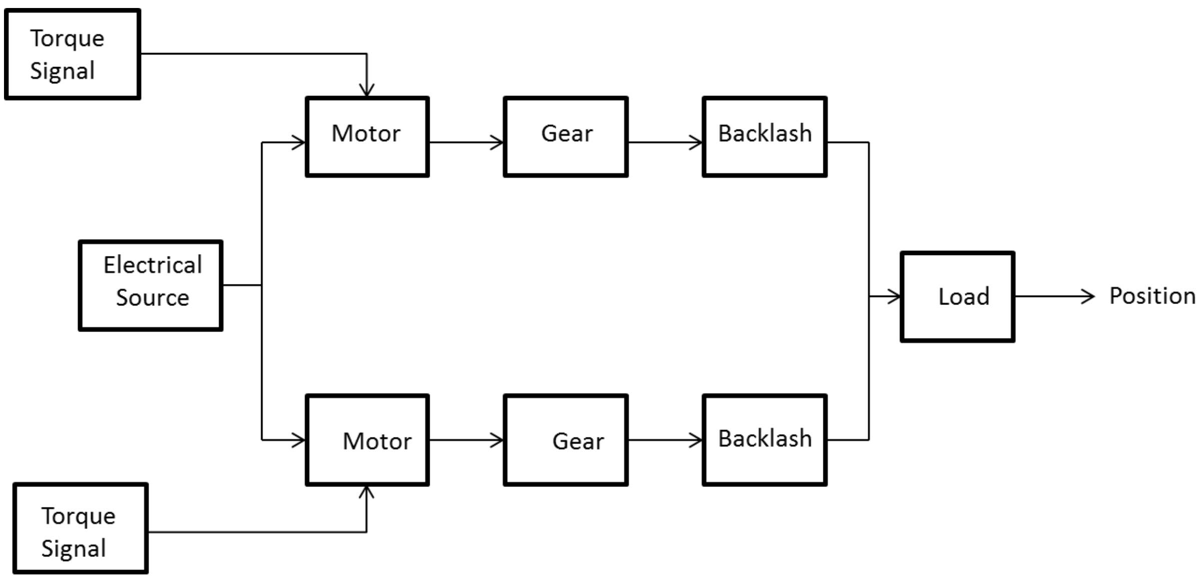

Figure 6-3 Block Diagram Representing the Main SIMULINK Model

Several SIMULINK models are developed. Two three-mass models are made; one including the master / slave driving technique and one without. In addition, it is created a reference model without both backlash and the master / slave configuration for the drive. The crane is to rotate 45°, equivalent to π/4 radians, stop and hold the position and then rotate back to the start position. Figure 6.4 displays the velocity and position graphs for the ideal scenario without backlash influence or the master / slave setup.

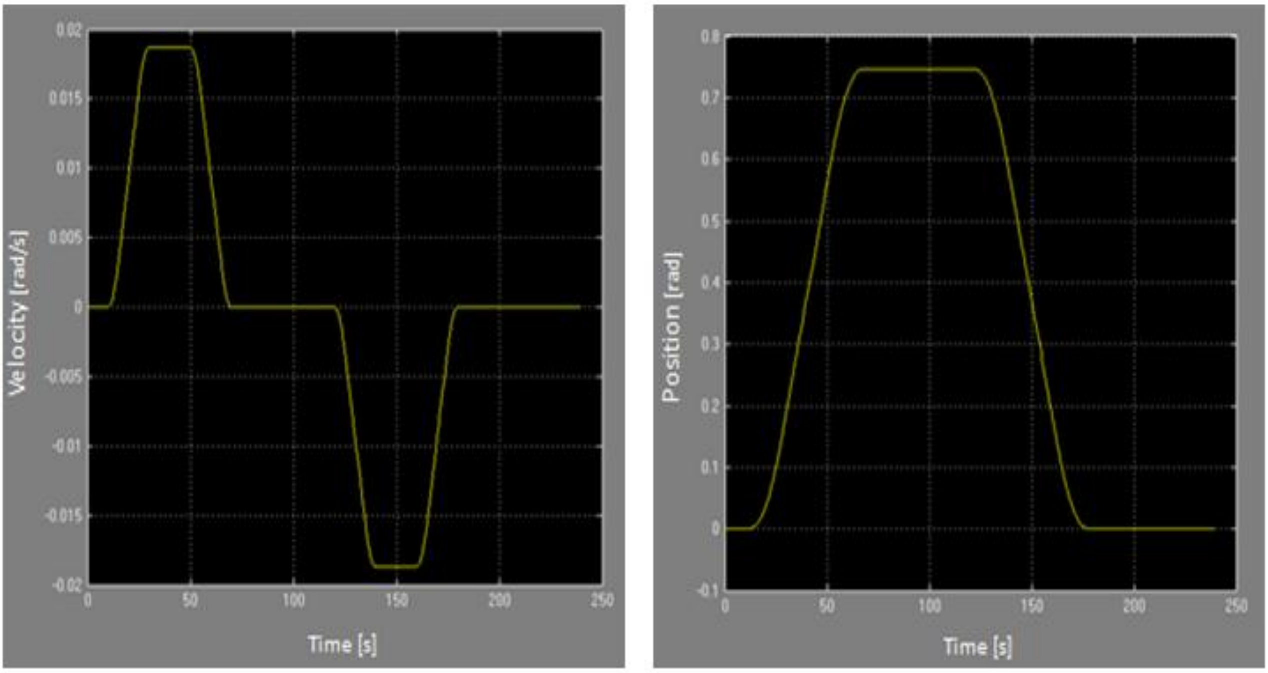

Figure 6-1 Reference Graphs, Velocity to Left and Position to Right, without Backlash

The main goal of the Master's thesis is to develop a system that effectively reduces the presence of backlash and results in graphs identical to the one presented in 6.4. In Figure 6.5 the corresponding system with backlash, but without master / slave driving strategy is displayed.

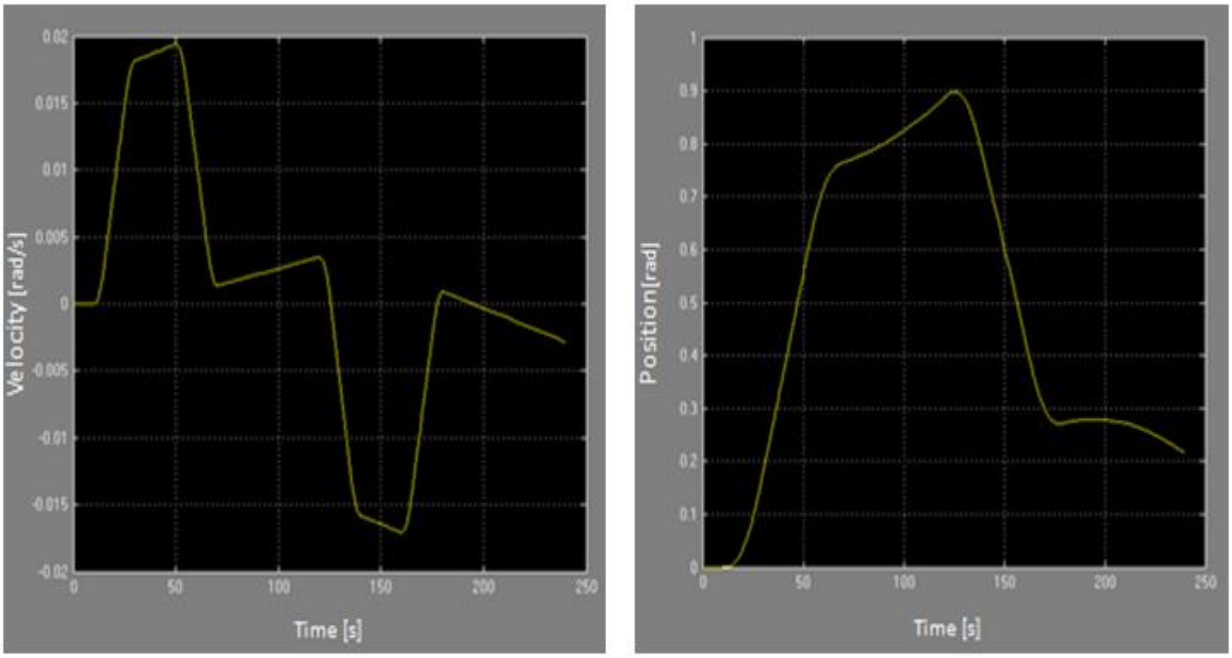

Figure 6-2 Graph of Velocity to Left and Position to Right. System with Backlash, without Control System

These graphs illustrate the influence of the backlash on the system and the resulting disturbances on the load velocity and position. The offset is approximate 10 % of the reference value.

The graphs in Figure 6.6 represent velocity and position after implementing the master / slave driving technique. By utilizing the slave motors as spring and break as described in Section 3.2.2, the backlash influence is drastically reduced and the resulting position error is approximate 1 %.

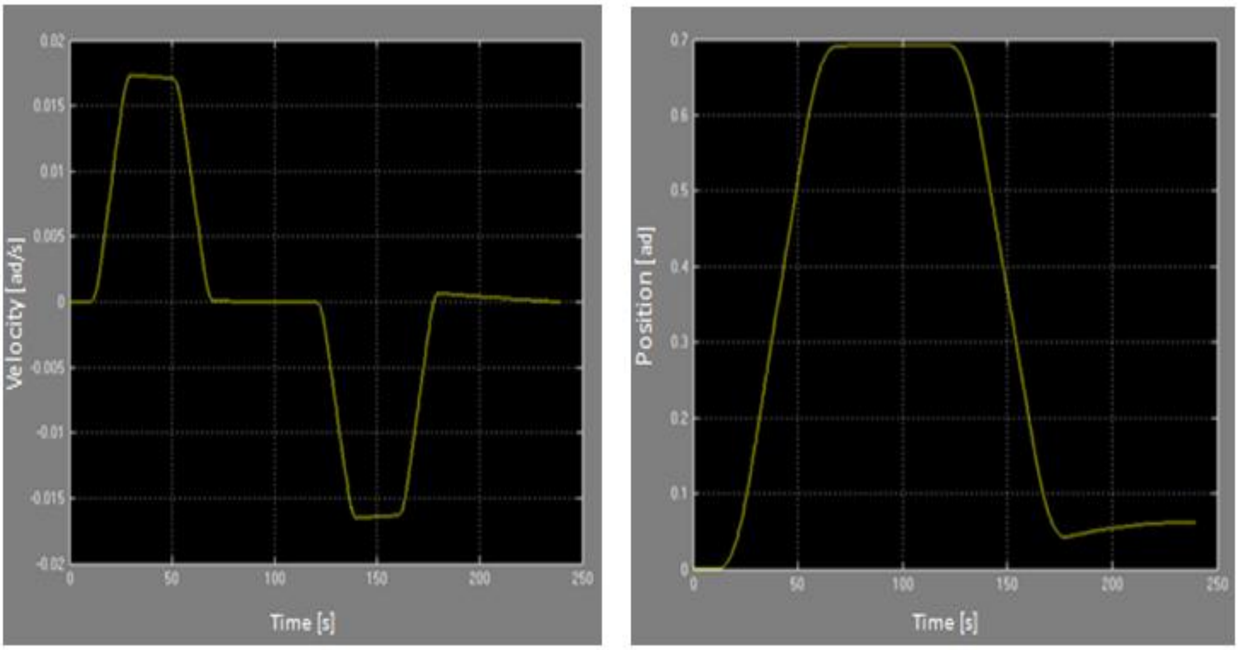

Figure 6-3 Velocity to Left and Position to right. With Master / Slave Driving Technique

As the controller system is not included in the SIMULINK model and the torque input to the motors are defined by a signal block, the load is not forced back to the reference position after backlash influence. This is particularly seen after 175 sec when the slew bearing is to be back at the origin position. At this point the approximated error position of 1 % is accumulated. Due to the absence of a functional control system, the error of 1 % increases for each change in the torque signal used as input to the motor.

Attached to a fully developed controller system and feedback from the encoders, the system would effectively rotate back to the reference position at $\pi/4$ radians. This would provide output similar to figure 3.13 in section 3.5.2, illustrating the relationship between acceleration and overshoot. The current situation makes it is impossible to optimize the relationship between acceleration and overshoot. It is also impossible to determine time of steady state after backlash disturbance.

The master / slave driving technique reduce the position error from 10 % to approximate 1 %, which is within the offset tolerance band given in Section 3.5.2. Based on this, it is legitimate to assume that if the controller unit was included in the SIMULINK model, steady state would be achieved within an acceptable period of time.

Figure 6.7 is a magnified view of the area where the crane motion stops at the given position of π/4 rad. The purple line represents the reference value of π/4 and the yellow line is the cranes angular position. The deviation is reduced considerably compared to Figure 6.5.

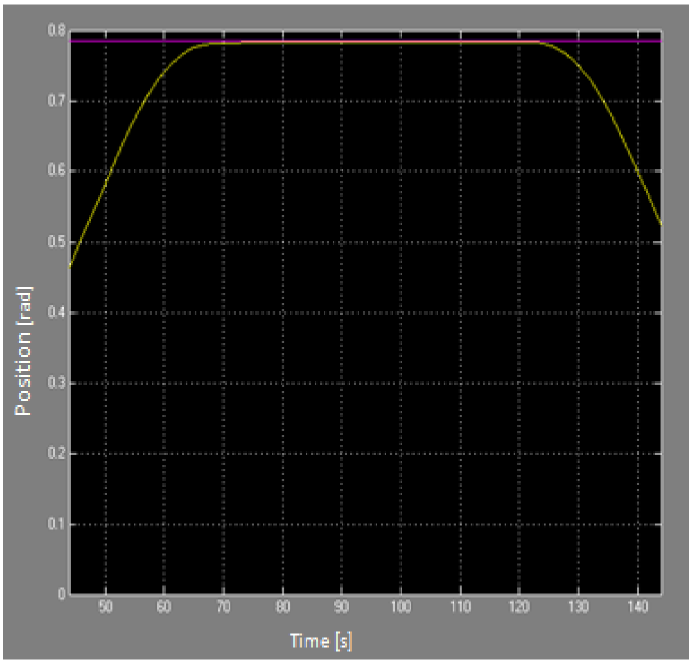

Figure 6-7 Reference and Actual Angular Position

Chapter 7

Discussion

In this chapter the solution and results are discussed relative to objectives, limitations, assumptions and design criteria defined in Chapter 1.

7.1 General Discussion

There are several aspects to account for when discussing the project results.

Control system and simulation results

Based on OBJ.4 and the limitation regarding the control system, a simple control design is proposed. This model is not fully developed and is not intended to be part of the simulation model. From Chapter 5 it is known that the input torque on the motor side is manipulated.

Due to the absence of a control system the load is never forced back to the reference position. A control system would guide the load back to the reference position by adjusting the motor input. This would have resulted in a graph with little deviation from Figure 3.13 in Section

3.5.2. Since the automatic correction never occurs in this system it is impossible to investigate the systems settling time and adjust the trade-off between acceleration and overshoot. After introducing the master / slave configuration the offset magnitude is approximately 1 % and it is therefore appropriate to assume that steady state will occur within a short period of time. This is also within the allowed offset requirement of $\pm$ 1% specified in Section 3.5.2.

Control parameters

e_{abs} is suggested as the control signal for the slave motors. For a fully automatic system where the end position is registered before every new motion, this would be the most effective solution. The crane subject for this Master's thesis, however, is manually controlled with a joystick and the end position is not preset. The crane operator has at all times the possibility to change the final position and using e_{abs} as the control parameter would not give an adequate result. It is therefore decided to suggest a new control structure utilizing speed changes as control variable. From research [24] it is known that nonlinearity mostly appears close to end points and during motion changes. In these situations the crane will have a relatively slow speed. This knowledge is used as basis when suggesting a speed controlled master / slave configuration. A reference value is defined and as long as the actual speed is larger than the reference speed, all motors are contributing to the main driving direction. When the speeds drops below the reference value, the slave motors starts to counteract the main motion.

Another topic to consider is the situation occurring during horizontal load stabilization. This is a situation where the crane operator manually tries to counteract external forces acting on the load. Since the steady state errors of the system are not determined it is not possible to evaluate if the mechanical system will come to a steady state between each direction change.

Overspeed

OBJ. 3 requires an analysis of overspeed handling in the motor drive. Overspeed is an important aspect to consider for all rotational, mechanical systems and it is important to assess the issue and how to avoid it in this exact case.

The finalized system will contain an overspeed protection system. As long as the required power for the load does not exceed the rated power of the motor during overspeeding, no significant problems will occur for the drive. This indicates that when the motor speed is above 1781 rpm, the power demand for the load must be less than the rated power of the four motors.

$$P_{load} \leq P_{nom} \leq 180 \text{ kW}$$

During all normal operational modes and situations, overspeeding will not be a problem. The motor supplier gives a motor speed of 4500 rpm as maximum speed without causing any mechanical damage to the motor system. Still, this is not a desired situation as the motor will not generate any significant torque at this speed.

Energy consumption

It is beneficial to reduce the energy consumption of the system. One way to do this is to examine the possibility to change to a 3:1 ratio of the master / slave configuration. With a total power contribution of 120 kW for the four motors a 3:1 ratio is an adequate approach. Both the negative effect of the backlash and the energy consumption will be reduced.

Excess Energy

The proposed solution design includes motors that at some given times act as brakes. This will develop a great amount of excess energy in form of heat. This heat is important to dispose of or use for a purpose.

If disposal of the excess energy is desirable, this can be performed by means of an air- or liquid cooled system.

If reuse is the purpose, the heat is proposed used for indoor heating or thermal processes, or even for distillation of waste water / salt water into fresh water [57]. This is provided that a waste heat recovery system is already installed at the offshore vessel the crane is mounted on. In relation to crane and slew bearing, a large amount of waste heat from the master / slave process is produced. Still, relative to the rest of the vessel the crane is mounted on, this heat is only a small fraction. It is therefore not appropriate to install a separate recovery system for the crane. The excess heat from this specific motor drive application is only to contribute in the already installed recovery system.

Safety and reliability

OBJ. 7 states that the new system has to undergo a basic analysis to consider the safety and reliability of the design. The electric system is based on the mechanical construction of the existing system, and thereby it is assumed that it complies with standards referred to in Section 3.6. Further factors to consider are; mean time between failures, total outage time, the system life time, demands for maintenance, reliability and environmental requirements.

Exact values for overspeed, overload and various specific limits on the capacity of the operating system, must be specified after significant testing when the new system is fully developed. At this stage of the development process, it is assumed that the same safety and reliability demands as the hydraulic system are valid, with reservations of necessary corrections. Nevertheless, brief assessments of how safety and reliability is maintained within the new solution must be performed, as these are very important aspects.

- The topic of overspeed has already been handled.
- With regards to overload, calculations with the DriveSize program (appendix A) from ABB shows that the system can withstand an overload of 150 % of the specified nominal load of the motor.
- The IP class is 56.

The electric multi-drive delivered by ABB is supplied with a predicted MTBF of more than 64.3 years. This means that during a period of 64.3 years, ABB expects only one inherent, internal structural failure of the system operation. High reliability is a key feature, as the repair of cranes is often complex and can be at hard to access locations offshore.

Design criteria

The design criteria state how the solution is to be obtained. Modular design is used throughout the entire work process to ensure scalability. By using ABB as supplier and by building SIMULINK models that are easy to adapt to different motors and drives, the system is transferable to other crane projects.

7.2 The Hypothesis

In Section 3.2 the following hypothesis is defined;

'The backlash will be reduced by a transition from hydraulic to electric motor drive of the slew bearing of a crane'

Results obtained during the work process, points out the electric motor drive to be the preferred choice over the existing system. An electric motor and motor drive combined with simple instrumentation provides the opportunity for very precise control of the mechanical system. As opposed to a hydraulic system, a controlled electrical system is less affected by, for example, temperature. Changes in temperature affect the compressibility of the hydraulic oil and thereby the oil flows through the pipes.

Compared to a hydraulic system, an all-electric system with induction motors has considerably less hardware such as hoses, pipes, valves and pumps. This leads to substantially reduced maintenance requirements and more effective installations. In all, the induction motor is preferred due to its reliability, ease of operation and low life cycle costs. The all-electric system also makes the solution more flexible as changes in the system can be made by software applications. This provides the ability for easy changes between speed- and torque control. In addition the absence of hydraulic oil removes the risk of environmental emissions.

The disadvantages with an all-electric system include a slightly higher purchase price and the requirement for power to the electrical machinery components and good cooling systems.

Based on findings obtained during the work process the hypothesis is verified.

Chapter 8

Conclusion

The work completed in this project aims to verify or falsify the hypothesis *'The backlash will be reduced by a transition from hydraulic to electric motor drive of the slew bearing of a crane'*. Based on the project results, it is concluded that the new concept design including an all-electric system, will reduce the backlash effect and improve the precision of the slewing. Although the design is in an early stage, the results are very promising. A list of potential future work that will improve the design is presented in the last section of this chapter.

8.1 General Conclusion

This thesis is motivated by Huse Engineering's constant strive to improve its products and gain competitive advantages. The main goal for this Master's thesis was thus to improve the slewing motion of an offshore crane by exploiting the new opportunities arising when changing its motor drive from hydraulic to electric.

Although some work remains to refine the system, it has been shown that an all-electric concept design reduce the backlash occurring in the slew bearing.

The main consequence of changing to an all-electric motor and drive is that an effective system control is more easily achieved, and that the system will respond faster and more precisely. This reduces the effect of backlash and provides a smoother crane motion with better accuracy.

The possibility to combine a functional control structure with a master / slave configuration is another advantage occurring when changing the drive system. The operational idea of the master / slave drive is that torque of a certain magnitude is applied in the opposite direction of the main driving direction. This provides a braking effect and improves the positioning and stiffness of the mechanical system. In addition to the improved control system, this is the main advantage of an electric motor drive. A well-designed motor drive system with a master / slave configuration, where the motor pairs are counteracting each other in given situations, should in theory be able to remove the effect of backlash entirely. Techniques for applying this type of solution to the existing hydraulic system have not been found.

It has also been shown that implementation of a good control system in combination with an electric variable speed drive significantly reduces the energy consumption significantly. Energy efficiency is in focus within all industries including the offshore market. By reducing the system’s energy consumption, Huse Engineering will gain a competitive advantage.

Environmental benefits include removing the risks of hazardous emissions from hydraulic oil. In addition, conditions regarding safety and reliability will be improved, as the asynchronous motors are solid machinery with a long service life and little need of manual maintenance.

Through this Master’s thesis it is found that an electric drive system with a master / slave configuration will provide a reduction of backlash in the slew bearing. The result will be a smoother and more precise rotation of the crane and a smaller deviation of the load’s position.

These findings verify the hypothesis.

8.2 Further Work

The intention of this work was to develop a new and advanced electric drive concept for the slew bearing. A master / slave configuration combined with a control circuit is proposed as the solution. Considering that there are not found any previous research available for comparison and verification of the new design it is necessary for the solution to undergo further testing and development.

- The proposed control scheme is a good approach. However, it needs further work and preparation before implementation onto a crane. Different controller values should be tested and the technique of soft switching of the drive should also be considered.
- The electric motor and motor drive need further simulations and testing. External forces such as wind and wave motion have yet to be accounted for, and simulation with different loads and conditions occurring during load stabilization should be further examined. It is assumed that the solution will function under these conditions, yet some performance parameters, such as the properties of the motor, multi-drive or gears may need to be changed to adapt to harsher conditions.
- A study of the possibilities of a 3:1 ratio of the master / slave configuration to reduce the energy consumption should be performed. Further work concerning the optimal magnitude of counteracting torque is also recommended.
- R-R is currently developing a PM motor. Despite the shortcomings of PM motors, such as the risk of overheating, and weaker magnets and reduced efficiency over time, the implementation of a permanent magnet motor would be beneficial. The PM motor is in general more powerful with a simpler and compact design than the induction motor. It would therefore be in both HE’s and R-R’s interest to investigate the benefits of implementing a PM motor to the new system design.

Bibliography

[1] Det norske veritas (www.dnv.no), 23. May 2012

[2] Ethernet: An Engineering Paradigm, Huang, Miller, Sun, Oji, 1998

[3] Designing and managing a software project (http://www.designingprojectmanagement.com/SoftwareProcessModels.html), 23. May 2012

[4] Introduction to research methodology (http://www.infotech.oulu.fi/GraduateSchool/ICourses/2006/phd/lecture3-oulu.pdf), 23. May 2012

[5] QSR (http://www.qsrinternational.com/what-is-qualitative-research.aspx), 23. May 2012

[6] Qualitative research methods: A data collector’s field guide, Family health internat ional (http://www.ccs.neu.edu/course/is4800sp12/resources/qualmethods.pdf), 23. May 2012

[7] Essentials of quality with cases and experiential exercises, Sower, Ph.D., C.Q.E., 1999

[8] ISO 9001 (http://www.dnvba.com/no/sertifisering/systemsertifisering/Kvalitet/Pages/ISO-9001.aspx), 23. May 2012

[9] IPCC good practice guidance and uncertainty management in national greenhouse gas inventories, Mangino,

[10] Project management tips (http://pmtips.net/project-quality-assurance-role/), 23. May 2012

[11] Software and hardware quality assurance: towards a common platform for high reliability, Martin, Mathur, 1990

[12] Visitask (http://www.visitask.com/project-quality.asp), 23. May 2012

[13] Designing and building parallel programs, Foster, 1995

[14] msdn (http://msdn.microsoft.com/en-us/library/ff921069(v=pandp.20).aspx), 23. May 2012

[15] Rother erde (http://www.rotheerde.com/gb/produkte_gwl.shtm), 23. May 2012

[16] SKF (http://www.skf.com/portal/skf_gb/home/literature?contentId=100019), 23. May 2012

[17] Brevini Norge (http://www.brevininorge.no/produkter/gear/17-svinggear-svinggir-sjaltegir-reduksjonsgear-vinskelgir-), 23. May 2012

[18] Sensor magazin, February 2010

[19] Gear design simplified, Jones, 1961

[20] Controlling mechanical systems with backlash—a survey, Nordin, Gutman, 4. March 2002

[21] Ericson Yachts (http://www.ericsonyachts.org/infoexchange/showthread.php?10136-Orienting-a-2-speed-winch-for-Load-Direction), 23. May 2012

[22] Boston Gear, Free gear theory (http://www.bostongear.com/index.asp), 23. May 2012

[23] Definition of backlash (http://www.qtcgears.com/Q410/PDF/techsec14.pdf), 23. May 2012

[24] Dual motor control for backlash reduction, Schiffer, June 2009

[25] Motion Control of Systems with Backlash, Bruns, Diepstraten, Schuurbiers, Wouters, August 2006

[26] A literature survey of control of automotive powertrain with backlash, Lagerberg, 2001

[27] efunda (http://www.efunda.com/formulae/fluids/overview.cfm), 23. May 2012

[28] SNL (http://snl.no/hydraulisk_motor), 24. January 2012

[29] digioo (http://www.digoo.info/industri/2011/01/Hva-er-en-hydraulisk-motor.html), 23. May 2012

[30] New energy and fuel (http://newenergyandfuel.com/http:/newenergyandfuel/com/2010/02/09/the-best-electric-vehicle-motor/), 23. May 2012

[31] Electric drives- An interactive approach, Mohan, 2003

[32] Electric motors, Woodford, 2012

[33] In Tech (http://www.isa.org/InTechTemplate.cfm?template=/ContentManagement/ContentDisplay.cfm&ContentID=76997) , 23. May 2012

[34] Four quadrant operation (http://www.wiki-kollmorgen.eu/wiki/tiki-index.php?page=Four+Quadrant+Operation), 23. May 2012

[35] Global spec (http://www.globalspec.com/reference/42927/203279/four-quadrant-operation-and-regenerative-braking), 23. May 2012

[36] Larsen & Toubro (http://www.automation4me.com/Products/backtobasics2.html), 23. May 2012

[37] Læringsportalen (http://arsplan.no/PraksisoppgaverVg2/AUA/Vg2_AUA_010/tabid/400/Default.aspx), 23. May 2012

[38] Buildings experts (https://www.buildingexperts.info/no/norske/paper/title/energibesparende-takket-vaere-turtallsregulerte-motorer-med-frekvensomformere.html), 23. May 2012

[39] Enova (http://naring.enova.no/sitepageview.aspx?articleID=252), 23.May 2012

[40] ABB (http://www05.abb.com/global/scot/scot234.nsf/veritydisplay/251b9bf8727aee9cc125784f0037fed7/$file/effektive%20motorerer%20og%20riktig%20turtall.pdf), 23. May 2012

[41] Automatic overload protection (http://www.seatrax.com/PDFS/AOPS.pdf),23. May 2012

[42] Cranes today (http://www.cranestodaymagazine.com/story.asp?storycode=2021291), 23. May 2012

[43] Opplyst (http://opplyst.no/IP-klasser), 23. May 2012

[44] The energy toolbox (http://www.engineeringtoolbox.com/ip-ingress-protection-d_452.html), 23. May 2012

[45] Rothe Erde; Slewing Bearings (http://www.rotheerde.com/download/info/V01_2_RE_ILM_W.pdf), 23. May 2012

[46] Dinamic Oil ,applications, technical sheet- slewing gearboxes, 12.

[47] H.E; Drawings Slewing Bearing

[48] Parker; Catalogue HY30-8249/UK, Hydraulic Motor/ Pump, Series F11/ F12, 23. May 2012

[49] H.E; Technical specification – 50 T 8 m Construction Crane

[50] ABB, M3BP 200 MLC 4- data sheet

[51] ABB, multi-drive (http://www.abb.com/product/seitp322/48504a9c976e1843c125747a002d3d32.aspx), 23. May 2012

[52] OpAmp Electronics (http://www.opamp-electronics.com/tutorials/three_phase_ transformer_ circuits_2_10_06.htm), 23. May 2012

[53] MathWorks (http://www.opamp-electronics.com/tutorials/three_phase_transformer_circuits_2_10_06.htm), 23. May 2012

[54] The data stream (http://thedatastream.4hv.org/gdt_leakage.htm), 23. May 2012

[55] ABB, Motor control with DTC, 2007

[56] Nandan Motors, http://www.nandanmotors.com/camgear.html,_23. May 2012

[57] Offshore Technology, 23. May 2012 http://www.offshore-technology.com/contractors/separation/maxim-evaporators/

Appendices

Appendix A – DriveSize Report

Appendix B – Slew Bearing & Gear Dimensions

Appendix A

DriveSize Report

Project Data Sheet		ABB ref. no.	ID
Customer:	Universitetet i Agder	Date:	22.02.2012
		Handled by:	Ivar Dyrendahl
Project name:	Huse Engineering sving 50	File name:	Drive size standard motorer 12 04 2012
Customer ref.	Nina Moe	Case:	1
Location:		Revision:	A

Network:	Short circuit power	200 MVA	Ambient conditions:		
Transformer:	Type	CTMP 24 H_ 500-500		Altitude motor	1000 m
	Primary voltage	21000 V		Temp. motor	45 °C
	Secondary Voltage	660		Altitude drive	1000 m
	Frequency	60 Hz		Temp. drive	45 °C
	kVA rating (Sn)	500 kVA		Altitude transformer	1000 m
	Impedance (zk) %	4,4		Temp. transformer	40 °C

Item No.	Drive name	Motor q'ty	Type designation Motor	Drive q'ty	Type designation Drive
1.1.0	[undefined]	-----	-----	-----	ACS800-207-0175-7
1.1.1	Drive 1	1	M3BP 200 MLC 4	1	ACS800-107-0050-7
1.1.2	Drive 2	1	M3BP 200 MLC 4	1	ACS800-107-0050-7
1.1.3	Drive 3	1	M3BP 200 MLC 4	1	ACS800-107-0050-7
1.1.4	Drive 4	1	M3BP 200 MLC 4	1	ACS800-107-0050-7

Figure A 1 Project Data Sheet

ABB

Line Supply Unit Technical Data Sheet					
Item No.	1.1.0		ACS800-207-0175-7		
Specifications			**Catalogue data**		
Name	[undefined]				
Type	ISU cabinet		Voltage [V]	690	
Cooling	Air		Drive power [KVA]	161	
IP Class	IP21		Icont max DC [A]	164	
			Ihd DC [A]	122	
			Imax DC [A]	245	
			Pcont max DC [kW]	160	
			Frame type	R7i	

Line supply unit load		**Calculations**			
		Power kW			
Motoring Idc cont [A]	90	mot cont	83.7	160	91 %
Motoring Idc max [A]	131	mot max	122	239	96 %
Regen. Idc cont [A]	0	gen cont	0	160	0 %
Regen. Idc max [A]	0	gen max	0	239	0 %
		DC current [A]			
		cont motoring	89.7	164	83 %
		max motoring	131	245	87 %
		cont generating	0	164	0 %
		max generating	0	245	0 %
		Temperature			
		mot temp			38 %
		gen temp			38 %

Figure A 2 Rectifier Data Sheet

ABB

Inverter Technical Data Sheet				
Item No.	1.1.1		ACS800-107-0050-7	
Specifications			**Catalogue data**	
Name	Drive 1			
Inverter amount	1		Voltage [V]	690
Type	Auto selection		Nominal power [kW]	37
			Nominal current [A]	46
			Imax [A]	68
			Ihd [A]	34
			Frame type	R5i
			Icont max [A]	51

Inverter load			**Calculations**			
Load type	Constant torque		Limit			
Overload type	Simple cyclic		Icont [A]	27.4	51	N/A
Icont [A]	27,4	590s	Imax [A]	34.9	68	95 %
Imax [A]	34,9	10s	Temperature			30 %
cosfii						
			ACS800-107-0050-7			

Figure A 3 Inverter Data Sheet

ABB

Motor Technical Data Sheet			
Item No.	1.1.1	M3BP 200 MLC 4	
Specifications		**DOL Catalogue data**	
Name	Motor 1	Product code	3GBP 202 033 (SE)
No.of motors	1	Voltage [V]	660
Motor type	Process performance	Frequency [Hz]	60
FrameMaterial	Not specified	Power [kW]	45
Family	Not specified	Poles	4
Polenumber	4	Speed [rpm]	1781
Design	Not specified	Max mech.speed [rpm]	4500
Connection	Not specified	Current [A]	49.2
IP class	IP55	Torque [Nm]	241
IC class	IC411 self ventilated	Tmax/Tn	3.3
IM class	IM1001, B3(foot)	Power factor	0.85
Max. speed rule	Standard	Efficiency [%]	94.2
Temp. rise	F (<105 K)	Temperature rise class	B
Tmax margin	43 %	Insulation class	F

Motor load		**Calculations**			
Load type	Constant torque	Torque [Nm]			
Overload Type	Simple	n min	104	182	76 %
n min [rpm]	0	n base	104	242	134 %
n base [rpm]	1750	Power [kW]			
n max [rpm]	1750	n min	0	0	0 %
Pbase [kW]	19	n base	19	44.4	134 %
Olbase [%]	150	Overload [Nm]			
Olmax [%]	150	n min	156	557	258 %
Temperature [°C]	45	n base	156	557	258 %
Altitude [m]	1000				
Ol definition:	No RMS				
0-1750 rpm / 100 %	590s				
0-1750 rpm / 150 %	10s				

Figure A 4 Motor Data Sheet

Project Technical Data Sheet		ABB ref. no.	ID
Customer:	Universitetet i Agder	Date:	22.02.2012
		Handled by:	Ivar Dyrendahl
Project name:	Huse Engineering sving 50	File name:	Drive size standard motorer 12 04 2012
Customer ref.	Nina Moe	Case:	1
Location:		Revision:	A

Item No.	Drive name	Motors per drive	Load type	Speed range min [rpm]	Speed range base [rpm]	Speed range max [rpm]	P [nbase] [kW]	Overload [nbase] [%]	Overload [nmax] [%]	Motor nominal values: Type designation	Un [V]	fwp [Hz]	Pn [kW]	Poles	In [A]	cos phi	eff. [%]	Tmax /Tn	Motor calculation result: Temp.rise/ Insul.	Tcont margin	Tmax margin	Icont [A]	Imax [A]	Drive q'ty	Frequency converter: Type designation	In [A]	Imax [A]	Inu calculation: Icont margin	Imax margin
1.1.1	Motor 1	1	Constant torque	0	1750	1750	19	150	150	M3BP 200 MLC 4	660	60	45	4	49	0,85	94	3,3	B/F	76 %	258 %	27	35	1	ACS800-107-0050-7	46		N/A	95 %
1.1.2	Motor 2	1	Constant torque	0	1750	1750	19	150	150	M3BP 200 MLC 4	660	60	45	4	49	0,85	94	3,3	B/F	76 %	258 %	27	35	1	ACS800-107-0050-7	46		N/A	95 %
1.1.3	Motor 3	1	Constant torque	0	1750	1750	19	150	150	M3BP 200 MLC 4	660	60	45	4	49	0,85	94	3,3	B/F	76 %	258 %	27	35	1	ACS800-107-0050-7	46		N/A	95 %
1.1.4	Motor 4	1	Constant torque	0	1750	1750	19	150	150	M3BP 200 MLC 4	660	60	45	4	49	0,85	94	3,3	B/F	76 %	258 %	27	35	1	ACS800-107-0050-7	46		N/A	95 %

Item No.	Drive name	Load power motoring Pcont [kW]	Load power motoring Pmax [kW]	Load power regenerating Pcont [kW]	Load power regenerating Pmax [kW]	Line supply unit: Type designation	Un [V]	motoring Sn [kVA]	motoring Idc [A]	regenerating Sn [kVA]	regenerating Idc [A]	motoring Pcont [kW]	motoring Pmax [kW]	regenerating Pcont [kW]	regenerating Pmax [kW]	Line supply unit calculation: motoring Icont margin	motoring Imax margin	regenerating Icont margin	regenerating Imax margin
1.1.0	[undefined]	83,7	122	0	0	ACS800-207-0175-7	690	161	164	161	164	160	160	160	160	83 %	87 %	0 %	0 %

Figure A 5 Overall Technical Data Sheet

Appendix B

Slew Bearing & Gear Dimensions

2.1 Basic data

Main dimensions and weight			
Outer diameter	DA	2700,00	mm
Inner diameter	DI	2140,00	mm
Height	H	231,00	mm
Weight	app.	2724	kg
Material data			
Gear rings		42CRMO4V	
Rings without gear		42CRMO4V	

2.2 Information regarding the fixing of the slewing bearing

The bolt connection has been dimensioned based on the data hereinafter and the load data of item 3.2.

	Bolts Outer ring	Bolts Inner ring
Hole circle diameter	2620,00 mm	2330,00 mm
Number of bolts	90	90
Bolt size	M 36	M 36
Borehole distribution	equal	equal
Clamping length	264 mm	180 mm
Property class of bolts DIN EN ISO 898-1: 2009	10.9	10.9
Nom. preload	90 %	90 %
Tightening procedure	Stretch method	Stretch method
Tightening factor Alpha-A	1,4	1,4
Loctite necessary between the contact surfaces	no	no
Coefficient of friction in the thread + under the bolt head support	0,14	0,14
Thread of the bolts rolled after heat treatment	yes	yes
Assembly pretensioning force	665,28 kN	665,28 kN

The bolt design with the a.m. data is based on a torsion-resistant and bending-stiff companion structure.

We assume that no bolts with particular characteristics (coating, cross-sections) are used on your part.

You are kindly requested to check whether a sufficient space has been taken into consideration for the tightening tools used by you.

Figure B 1 Structural Data

2.3 Gearing data

		Wheel	Pinion	
		Internal gear	External gear	
Module		20		mm
Number of teeth	z	108	14	
Addendum modification	xm	-10,00	10,00	mm
Addendum reduction	km	0,00	0,00	mm
Tooth width	b	166,00	171,00	mm
Gearing quality	DIN	12	10	
Theoretical centre distance	a	940,000		mm
Pressure angle	α	20 °		
Helix angle	β	0 °		
Helix direction				
Basic rack tooth profile		DIN 867: 1986		
Number of drives	4	equally spaced		

2.4 Pinion requirements

2.4.1 Tip Relief

Rothe Erde requires a tip relief in form of an involute.

The Radius R must blend into the addendum flank without forming an edge.

Ca	=	0,2	mm
h	=	8 -12	mm
R	=	2 –3	mm

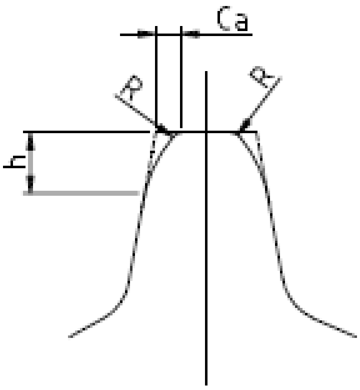

Figure B 2 Gear Da

3.2 Raceway calculation

For dimensioning the loads indicated hereinafter have been used, which have been determined based on your details.

The calculated static safety in the following table refers to the specifically most loaded raceway.

This calculation is subject to a bearing installation into a torsion-resistant and bending-stiff companion structure as well as to the compliance with our Instructions for Installation, Lubrication and Maintenance.

3.2.1 Raceway loads

Load case	Description	Axial load Fa [kN]	Radial load Fr [kN]	Tilting moment Mk [kNm]	Speed n [1/min]	Amount of time [%]	Kind of rotation	Static raceway safety	
								Actual	Required
1	Max. working load	700,00	150,00	9700,00	1,00	100,00	Slewing motions	1,60	1,45
2	Incl. 25% overload	750,00	150,00	10700,00	1,00	0,00		1,45	1,45
3	Incl. load factor 2.0	1500,00	350,00	12000,00	0,00	0,00		1,31	1,00

Figure B 3 Raceway Data