Kinematic Control of Redundant Knuckle Booms
with Automatic Path-Following Functions

Björn Löfgren
Kinematic Control of Redundant Knuckle Booms with Automatic Path Following Functions

Doctoral thesis

This is an academic thesis which with the approval of the Department of Machine Design, Royal Institute of Technology, will be presented for public review in fulfilment of the requirements for a Doctorate of Engineering in Machine Design. The public review will be held at Kungliga Tekniska Högskolan, M3, Brinellvägen 64, Stockholm, at 13:00 on the 4th of December 2009.

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To stay competitive internationally, the Swedish forestry sector must increase its productivity by 2 to 3% annually. There are a variety of ways in which productivity can be increased. One option is to develop remote-controlled or unmanned machines, thus reducing the need for operator intervention. Another option—and one that could be achieved sooner than full automation—would be to make some functions semi-automatic. Semi-automatic operation of the knuckle boom and felling head in particular would create “mini-breaks” for the operators, thereby reducing mental and physiological stress. It would also reduce training time and increase the productivity of a large proportion of operators.

The objective of this thesis work has been to develop and evaluate algorithms for simplified boom control on forest machines. Algorithms for so called boom tip control, as well as automatic boom functions have been introduced. The algorithms solve the inverse kinematics of kinematically redundant knuckle booms while maximizing lifting capacity. The boom tip control was evaluated – first by means of a kinematic simulation and then in a dynamic forest machine simulator. The results show that boom tip control is an easier system to learn in comparison to conventional control, leading to savings in production due to shorter learning times and operators being able to reach full production sooner. Boom tip control also creates less mental strain than conventional control, which in the long run will reduce mental stress on operators of forest machines. The maximum lifting capacity algorithm was then developed further to enable TCP path-tracking, which was also implemented and evaluated in the simulator.

An evaluation of the fidelity of the dynamic forest machine simulator was performed to ensure validity of the results achieved with the simplified boom control. The results from the study show that there is good fidelity between the forest machine simulator and a real forest machine, and that the results from simulations are reliable. It is also concluded that the simulator was a useful research tool for the studies performed in the context of this thesis work.

The thesis had two overall objectives. The first was to provide the industry and forestry sector with usable and verified ideas and results in the area of automation. This has been accomplished with the implementation of a simplified boom control and semi-automation on a forwarder in a recently started joint venture between a hydraulic manufacturer, a forest machine manufacturer and a forest enterprise. The second objective was to strengthen the research and development links between the forestry sector and technical university research. This has been accomplished through the thesis work itself and by a number of courses, projects and Masters theses over the last three years. About 150 students in total have been studying forest machine technology in one way or the other.

**Keywords**

Hydraulic manipulator, redundancy, kinematic control, local optimization, knuckle boom, forest machine, forwarder, boom tip control, joystick control, simulations, path following

**Language**

English
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Björn Löfgren

November 2009, Stockholm
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**Paper A:**

**Paper B:**

**Paper C:**

**Paper D:**
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Chapter 1
Introduction

1.1 Introducing the problem

Of the total cost of wood, 85% is related to the operational supply that brings the raw material from the forest to the industries, and 15% is related to silviculture. This makes it clear that the supply chain plays a major role in the forestry economy (Rådström, 2008).

Thanks to improved productivity in logging, it has been possible to maintain satisfactory profits despite the downward pressures on raw-material prices (Brunberg, 2007) (see Figure 1.1).

![Figure 1.1: The cost of regeneration felling in Sweden in comparison with the consumer price index (CPI).](image)

To stay competitive on the international market the Swedish Forestry sector must increase productivity by 2 to 3% every year (Rådström, 2008).

Systems in operation today have been fully mechanized and the forestry sector is entering a new era in which automation will be introduced. This will involve automation of entire operations, knuckle boom manipulation and other parts of the process—such as the collection, transmission and reporting of information.

Another incentive for introducing automation is to improve the working conditions of the machine operator. A lot of work to reduce the physical stress to which operators are subjected has already been successfully done over recent years. However, the higher pace of work today, combined with the many qualified decisions that the operators must make under pressure, is imposing greater mental stress. This increases the likelihood of becoming stressed and having disorders such as repetitive strain injuries (RSI), which in turn can lead to sick leave, exclusion from work, and recruiting problems. As a result
operators could become bottlenecks in the advancement of productivity and the optimal utilization of raw materials.

Technological advances tend to occur in leaps and bounds, often making it possible to introduce numerous improvements also into the logging systems. Although it is difficult to predict what the next advance in technology will be, such predictions are critical for the forestry sector. Since most technological developments take place outside of this sector, the industry it is obviously dependent on advances that are made in the world at large.

There are a variety of ways in which productivity can be increased. One option would be to develop remote-controlled or unmanned machines, thus reducing the need for operator intervention. Another option and one that could be achieved sooner would be to make some of the functions semi-automatic. Semi-automatic operation of the knuckle boom and felling head in particular would create mini breaks for the operator and reduce the mental and physiological stress. An additional benefit is that it would also reduce training time and increase the productivity of a large proportion of operators.

Forest machine operators work under a great deal of mental pressure. They have to deal with a large volume of information and make crucial decisions while under considerable time pressure. Moreover, operating the machine controls with their hands is also quite demanding on the brain.

Automation of boom operation would, therefore, reduce mental pressure on the operator, not only giving the operator more time but also greater opportunity to concentrate on other tasks, such as tree selection, bucking options (bucking occurs when the tree is cut into logs suitable for transport and refinement while also maximizing the economical output of the tree), and assessing any damage or defects visible on the next tree.

The downside of automation of boom operation is that it could degrade the work of the operator, particularly those tasks characterized as “skillful” and those that represents valuable interventions. On the other hand, technological advances that give the operator wider scope for making qualified decisions could counter such risks. New technology should focus on removing the time constraints on decisions (i.e., reducing the need to make decisions at any given moment), and in making the operator more responsible for the qualified decisions that can increase yield values.

Semi-automation of the knuckle boom has been identified as the most important target for development (Eriksson & Thor, 1999). One solution in this regard would be to introduce boom-tip control, which would simplify the boom work and reduce the need for multiple, simultaneous control movements. Besides boom-tip control, the automation of different boom sequences is an important target.

The reason that simplified boom control or automated boom functions have not yet been implemented on forest machines is because the machines have not been sufficiently reliable to date. Up to now, the main objective for machine manufacturers has been to design forest machines that are reliable and highly productive units. Another reason for the lack of simplified boom control or automated boom functions is that no one knows
what these simplifications can provide from a productivity perspective and in terms of reducing mental and physical stress.

The timing for the introduction of simplified boom control and automatic functions is currently ideal since forest machines are now more reliable and have mature computer systems.

1.2 Purpose and objectives

The objective of this work has been to develop and evaluate algorithms for simplified boom control, including automatic boom functions on forest machines. This is accomplished by first conducting an analysis of how to control a redundant knuckle boom, and then developing, implementing and evaluating the corresponding algorithms in a simulator.

The overall long time purpose of this thesis is twofold:

- To provide the industry and the forestry sector with valuable, useable and verified ideas and results that can be used to introduce semi-automatic and automatic boom control.

- To strengthen the research and development (R&D) links between the forestry sector and technical university research.

1.3 Delimitations

In order to make the problem tangible and to keep the framework regarding resources for this work, the following delimitations have been made:

- Only kinematic solutions have been solved by the developed algorithms and no dynamic controls have been introduced in the algorithms.

- No implementation has occurred on a real forwarder for evaluation in a normal forestry application. Instead, a real-time forest machine simulator has been used to avoid the large costs involved in prototyping on a real machine.

1.4 Outline of the thesis

The thesis begins with an introduction to the research problem and a presentation of the purpose, objective and delimitations of the work (Chapter 1). Chapter 2 presents the research approach for this work. Chapter 3 describes the algorithms which have been developed and analyzed, and concludes with a discussion on why the chosen algorithm has been used throughout the work. Chapter 4 describes the use of simulators in forestry and in excavator applications, and includes a description of the simulation software and the dynamic forest machine simulator, as well as the problem of using simulators in research. A summary of the appended papers is included in Chapter 5. Chapter 6 presents results from a comparison between the simulator and reality. Finally, in Chapter 7, the thesis ends with suggestions for future research.
1.5 Forestry in Sweden

Sweden has a land area totaling 41 million hectares, with forests covering about half of this area. This translates into 2–3 hectares of forest per capita. Indeed, for the average Swede the forest plays a major role not only in his or her natural landscape, but also as a place for recreation. An ancient right entitles Swedes to enter and enjoy any forest freely, regardless of ownership.

Forestry is also of major importance to Swedish society in general and to the total value of exports, as Figure 1.2 demonstrates.

![Figure 1.2: Exports and imports of some product groups in Sweden 2008.](www.skogsindustrierna.se)

Being heavily dependent on the export market, Swedish industry has to ensure that its prices are competitive on the international market. To accomplish this, the forest industry has had to increase the efficiency of all its operations, from stump to dockside.

1.5.1 The forest of Sweden

The most common tree species in Sweden are Norway spruce (45%), Scots pine (39%) and hardwood species (16%). The National Forest Survey is responsible for regularly monitoring the standing stock, increment and cut. The annual cut, which is about 90 million m$^3$, has been lower than the annual increment during the last few decades except for 2005, due to the storm “Gudrun”. The standing stock is steadily increasing and now averages 120 m$^3$/hectare, although in mature stands the figures can more than double this.

The typical stand rotation is generally between 70 and 120 years. After the establishment, the stand is cleaned, thinned one to three times and, sometimes, fertilized. In contrast to natural forests, stands under long-term management contain large-diameter, even-aged and relatively sparse underbrush and few dead trees.
About 13% of Swedish forests are owned by the public sector, with another 37% belonging to forest companies. The remaining 50% is distributed among some 250,000 private woodlot holdings. The methods used in private woodlots often differ from those used in large-scale forestry. About one-third of private woodlot owners are members of a forest owner association, which looks after the interests of its members. Such associations represent about 50% of privately-owned, non-corporate forest land.

Given the importance of forestry in Sweden, forest legislation has since 1903 been in place to ensure that the country’s forest resources are properly managed.

1.5.2 Logging

The annual cut is around 90 million m$^3$ in gross volume, of which more than 70% originates from regeneration felling. The traditional logging method in Sweden is the cut-to-length method (CTL), where the tree is processed (i.e., limbed and bucked in the stand) and subsequently extracted from the forest to the roadside in different lengths, normally 2.5 to 5.5 metres.

The major factors influencing this choice of method are:

- Relatively small and scattered operations, distributed along a private road network with many owners;
- Divergent flows (saw and pulpwood) are hauled to different locations;
- Difficulties in maintaining quality and efficiency in the thinning process when using other methods;
- The CTL method causes fewer disturbances to the site residual stand; and
- Tradition.

1.6 Forest machines

Today, large-scale logging operations are almost completely mechanized. The single grip harvester and the forwarder together constitute the system in total dominance.

The single grip harvester cuts and processes the trees, and the forwarder transports the logs from the stand out to the landing area, where the trucks transport the logs to the sawmill and the paper industry.

1.6.1 Single grip harvester

The single grip harvester controls both the felling and bucking processes with a harvester head, located at the tip of the boom (see Figure 1.3).
Harvesters come in different sizes, ranging from 8-ton machines used for thinning to 23-ton machines used for regeneration felling. The crane reaches a distance of up to 12 metres. The felling process starts when the operator moves the harvester head towards a tree, grasps it and cuts it off with a chain saw. The harvester head keeps the tree within the head with the help of the limbing knives and the feed rollers. The rollers start to feed the stem, simultaneously limbing them. The diameter and length of the stem is continuously measured and compared with a pricelist in the bucking computer to get an optimal economic outcome from the bucking of the stem. The same chain saw used in the felling process cuts the logs, which are then put into different sorting piles on the ground.

The harvester operator is subjected to mental and physical stress due to:

- The high pace of work, which is roughly one tree every 47 seconds, or 1000 trees per day (Brander & Nordén, 2004);
- The high number of required decisions, which is roughly 12 decisions per tree (Gellerstedt, 1993);
- Up to 50 functions (e.g., movements performed through joysticks, pedals and buttons) performed per tree, with an average of 24 functions per tree (Brander et al. 2004); and
- The risk of repetitive strain injuries, and no “micro-pauses” during work.
1.6.2 Forwarder

Forwarders (Figure 1.4), which vary in load capacity from 10 to 20 tons, carry the logs to the roadside. Smaller forwarders are used in thinning, and bigger ones in regeneration felling. The efficiency of the forwarder depends on a number of factors, including operator skill, extraction distance, planning of extraction, roads and the shape of the sorting piles.

Figure 1.4: A forwarder Valmet 860, at roadside. (www.komatsuforest.com).

The forwarder operator is also subjected to mental and physical stress, due to:

- The high pace of work;
- Whole body vibrations. At a typical pace of operation, almost every forwarder exceeds the maximum level of acceptable whole body vibrations (1.1 m/s²) as per the EU Physical Agents’ Directive 2002/44/EG;
- The combination of many functions due to a redundant knuckle boom. The operator loads and unloads on average a grapple with logs every 30 seconds, and handles about 2000 logs per day. The number of logs depends on the average stem volume and the transportation distance; and
- The risk of repetitive strain injuries, and no “micro-pauses” during work.
1.6.3 Interaction between operator and machine

The interaction between the operator and all of the functions he/she must perform to do the job has seen many improvements over the last 20 years (see Figures 1.5 and 1.6).

Figure 1.5: Left: the interior of a harvester cabin from the 1980s. Right: the interior of a forwarder cabin from the 1980s. (www.skogforsk.se).

Figure 1.6: Left: the interior of a harvester cabin in a modern Valmet 941. Right: the interior of a forwarder cabin in a modern Valmet 860. (www.komatsuforest.com).

Even though the environment in the cab has been significantly improved, the interaction between the operator and the machine today is of higher intensity than it was in the past. This is due to the fact that productivity has increased twofold over the last decade, as well as the implementation of new functions, such as GIS, GPS, and Internet. The complexity of the control system is depicted in Figure 1.7. Even though this figure shows the control system layout of a Valmet harvester, the overall structure is valid for almost every forest machine.
INTRODUCTION

1.6.4 Boom tip control

The forestry machines of today are controlled by two joysticks of different design. With a movement of one joystick in one direction, the operator controls a specific hydraulic cylinder on the boom (see Figure 1.8). This means that the operator has to combine different joystick movements to move the tip of the boom in the desired direction. In other words, the operator solves the inverse kinematics problem.

The concept boom tip control (BTC) (Löfgren, 2004, Paper A) means that the tip of the boom is controlled with only one joystick. Up/down on the joystick corresponds to up/down on the tip of the boom, out/in on the joystick corresponds to out/in on the tip of the boom, and left/right corresponds to left/right on the tip of the boom (see Figure 1.9).
INTRODUCTION

Figure 1.9: Boom tip control of a forest knuckle boom.

Potential advantages of boom tip control

A simplified manipulator control would provide the following advantages, compared to conventional manipulator control:

- less physical strain on the operator;
- decreased learning time; and
- longer manipulator lifespan.

Operators of forestry machines frequently have injuries to the neck, shoulders and back after several years of controlling the manipulator (Eriksson & Thor, 1999). Indeed, more than half of all operators have had problems with these areas of the body. These problems can, to a large extent, be attributed to manipulator control work.

It has been demonstrated that operators unconsciously tense themselves before manipulator control work, with increased stress as a consequence (Eriksson & Thor, 1999). A simplified manipulator control would most likely allow operators to control the manipulator in a more relaxed manner.

Work with the conventional controls of knuckle booms is very complicated since, knuckle booms are redundant, i.e. it is possible to reach every point within the knuckle boom’s workspace in many ways. Boom tip control would most likely make it easier to control the knuckle manipulator, since the tip would be controlled directly and since the redundancy would be resolved automatically.

Consequently, a simplified control system would also most likely mean that the learning time would decrease substantially (Suh & Hollerbach, 1987, Paper C).

There is a large difference between a skilled operator and a non-skilled operator (Siciliano, 1990). In forest operations, productivity variations of 25–40% are not uncommon. A non-skilled operator will control the manipulator in a jerkier manner, which affects the lifespan of the manipulator. With simplified controls, it would be
possible to eliminate many of the jerky movements and thereby increase the lifespan of the manipulator.

1.6.5 Semi-automation of boom tip sequences

Semi-automation means that parts of the boom sequences are made automatic, such as boom out, boom in, start up or parking the boom in the load space.

Operator opinions

In interviews with operators (Löfgren et al. 2002) from across Sweden, they were very positive about the automation of functions on forest machines and the automation of complex functions that today have to be controlled at the same time as other functions. They also stated that they would be happy to see the automation of repetitive boom functions. All of the interviewed operators were of the opinion that automation of functions should be designed not to affect the working rhythm of the other manually performed work, since this could create mental stress and decrease productivity. In studies of forwarders (Löfgren et al. 2002), it has been noticed that the boom out and boom in functions are sequences that could benefit from automation. Another candidate for automation could be setting and returning to the last position of the boom or a predetermined position.

Tests with unskilled operators

In Brander et al. (2004), students from a forestry vocational school took part in a test that involved both conventional controls and semi-automated functions of a single grip harvester. As Table 1.1 shows, the usage of the joystick and push-button controls was much lower when functions were semi-automated than when functions were performed via conventional operator control, which do not give the operator sufficient time for “micro–breaks”. The results also pointed towards an increase in productivity. Even though only a few semi-automated functions were introduced, the findings clearly demonstrated the potential for introducing semi-automation in harvester operations. The performance of the students was also compared to that of a highly skilled operator (see Table 1.2). The result shows that the difference in the time needed to perform the job between the skilled operator and the students was decreased by approximately 70% when using the semi-automated functions.

<table>
<thead>
<tr>
<th>Swing function</th>
<th>Boom up/down</th>
<th>Telescope in/out</th>
<th>Rotation function</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average reduction of joystick activations in automation</td>
<td>63 %</td>
<td>62 %</td>
<td>60 %</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Conventional control</th>
<th>The students needed 75% more time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Automation</td>
<td>The students needed 21% more time</td>
</tr>
</tbody>
</table>

Tests with a skilled operator

In Brander & Nordén (2004), a more detailed study using a dynamic forest simulator was performed to see how automation affects a skilled operator in his/her work. The
operator (in this case a highly skilled operator) performed a final cutting with a single
grip harvester in a very well-defined forest stand (a copy of a real stand). Five studies
were carried out in the following order: Conventional Control, Automation, Automation,
Automation and Conventional Control. As can be seen in Figure 1.10, the total working
time decreased by 33% between the first and the fifth study, which in both cases were
conventional control.

![Figure 1.10: Total working time for each study.](image)

The decrease in time can be explained by a different use of the dynamic forest simulator
compared to the real machine. In the first study, the operator acted as he would in a real
forest machine. According to the operator, during the test he learned how to make
shortcuts in the dynamic forest simulator that were not possible in a real machine. Even
though the operator learned how to make shortcuts, between each study, the automation
did not affect productivity negatively—despite the operator being highly skilled.

1.7 Estimation of productivity and economic potential
In Löfgren et al. (2002), a qualitative estimate was made of how automation could affect
productivity and the mental and physical stress of the operators (see Tables 1.3 and 1.4).

<table>
<thead>
<tr>
<th>Activity</th>
<th>Productivity</th>
<th>Mental and physical stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boom work</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Boom out</td>
<td>+</td>
<td>++</td>
</tr>
<tr>
<td>Grasping</td>
<td>+</td>
<td>+</td>
</tr>
<tr>
<td>Boom in</td>
<td>+</td>
<td>++</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Activity</th>
<th>Productivity</th>
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<tr>
<td>Boom work</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Boom out</td>
<td>+</td>
<td>++</td>
</tr>
<tr>
<td>Grasping</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Boom in</td>
<td>+</td>
<td>++</td>
</tr>
<tr>
<td>Unloading</td>
<td>0</td>
<td>+</td>
</tr>
</tbody>
</table>

Table 1.3: Estimated effect of automation on productivity and on mental and physical stress of the operator during felling activity in a harvester.

Table 1.4: Estimated effect of automation on productivity and on mental and physical stress of the operator during forwarder activity.
An analysis was also conducted to determine the potential impact on profit if productivity was increased by 10 or 20% (see Tables 1.5 and 1.6). The calculations are based on Swedish forestry industry statistics and an annual cut of 38 million m³ solid per year, with a distribution of 28 million m³ solid in regeneration felling and 10 million m³ solid in thinning.

Table 1.5: Regeneration felling.

<table>
<thead>
<tr>
<th>SEK/m³ solid</th>
<th>Increased productivity by</th>
<th>Profit in million SEK/year</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boom time %</td>
<td>Harvest head time %</td>
<td>10%</td>
</tr>
<tr>
<td>Felling</td>
<td></td>
<td>42</td>
</tr>
<tr>
<td></td>
<td></td>
<td>51</td>
</tr>
<tr>
<td>Terrain transport</td>
<td></td>
<td>31</td>
</tr>
<tr>
<td>Total profit</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 1.6: Thinning.

<table>
<thead>
<tr>
<th>SEK/m³ solid</th>
<th>Increased productivity by</th>
<th>Profit in million SEK/year</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boom time %</td>
<td>Harvest head time %</td>
<td>10%</td>
</tr>
<tr>
<td>Felling</td>
<td></td>
<td>85</td>
</tr>
<tr>
<td></td>
<td></td>
<td>51</td>
</tr>
<tr>
<td>Terrain transport</td>
<td></td>
<td>46</td>
</tr>
<tr>
<td>Total profit</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

In summarizing the estimations and calculations, it is likely that boom automation would be favorable to the operator, and would increase profit if introduced.
Chapter 2
Research Approach

There are several approaches to consider when designing a research project and its constituent studies. In applied engineering studies, research tools such as experiments and correlation research are common. Qualitative and empirical methods are used in this thesis. The purpose of this chapter is to introduce the main research method used. As described below, the research consists of five parts, all of which made individual contributions to the overall aim and scope.

2.1 Problem identification, solution and hypothesis

Chapter 1 identifies problems with productivity and the control of the boom on forest machines. Automation of the boom is an initiative that shows significant potential for addressing these problems in the near term. In the long run, the forestry sector can foresee forest machines becoming completely autonomous and controlled by operators far away from the actual work site.

2.1.1 The solution

The solution, which will be dealt with in this thesis, is a simplified boom control and automatic boom sequences on a forwarder.

2.1.2 Hypothesis

Introducing automation and enabling the forest machine to perform most of the repetitive work by itself will create potential for increased productivity and increased product value, while simultaneously reducing the negative working environment effects on the operator. The operator can thereby focus on tree quality and environmental concerns, and also reduce wear and tear on the forest machines.

2.2 Simulations

A major approach in this research has been the use of simulations for the evaluation of suggested boom algorithms and automatic path-following functions. Two types of simulations have been used: own-developed kinematic simulation software (hereafter called KSS), and a dynamic forest machine simulator (hereafter called DFMS).

2.3 Industrial partners

Industrially applied research greatly depends on the researchers forging a productive and a long-term relationship with industrial partners, in this case manufacturers in the forestry sector, all of which had experiences with forest machine problems. Part 1 of the research focused on cranes and only companies developing and manufacturing cranes were approached. In the latter parts of the research, forest machine manufacturers, forest enterprises, forest machine entrepreneurs and researchers were approached.
2.4 Research partitioning

The work could be described as having five parts. Problem identification and theory development took place in Part 1, with Paper A being written in this phase. The fidelity analysis of the DFMS was performed in Part 2. In Part 3, an evaluation of the proposed simplified boom control algorithm was done using the DFMS. The identification, design and analysis of automatic boom functions, using both the KSS and the DFMS, were performed in Part 4. Finally, a verification of the capability of the DFMS to mimic a real machine was performed in Part 5. Paper B was written in Part 2, Paper C in Part 3, Paper D in Part 4, and finally, this doctoral thesis in the fifth and concluding part. Four main empirical studies have been performed, one each in Parts 2, 3, 4 and 5.

2.4.1 Part 1

In this first part, a theoretical study was undertaken to analyze three different algorithms on how the redundancy of the knuckle boom could be solved. The algorithms, described briefly below, were evaluated in the own-developed kinematic simulation software.

Maximum velocity

In some applications, velocity is of major importance. The kinematic control law that was applied to give maximum velocity is based on a weighted pseudo inverse of the Jacobian. The results were reported in the licentiate thesis (Löfgren, 2004).

Maximum lifting capacity

In some applications, static lifting capacity is essential. The corresponding kinematic control law is based on an optimization study of the lifting capacity. The results were reported in condensed form in Paper A.

Dynamic programming

In some manipulator applications the start point and the end point are known. By using dynamic programming it is possible to find the minimum time between two points in the workspace. The results were reported in the licentiate thesis (Löfgren, 2004).

2.4.2 Part 2

In the second part, an empirical study was conducted to investigate the fidelity of the DFMS, which was to be used later for verifying the benefits of the developed algorithms. A time study was conducted in a real stand, where a harvester operator has cut down approximately 500 trees and where data from the stand such as tree diameter, height, position and height to first live branch and also tree type was measured. Also the terrain was measured. The same stand and terrain data was then implemented into the DFMS and the same operator performed the same work again. The results were reported in condensed form in Paper B.
2.4.3 Part 3
In the third part, an empirical study was conducted on the DFMS to investigate the maximum lifting capacity algorithm for the boom tip control that was used throughout the work in Part 1. The purpose of the study was to investigate how two types of boom control, conventional control and boom tip control affected inexperienced operators in their learning capabilities. The participants in the study were from a vocational school for natural resource use and between 15 and 19 years of age. Altogether, sixteen students (fourteen male and two female) voluntarily participated in the study. None of the students had previously participated in any simulator or real forest machine studies. The results are reported in condensed form in Paper C.

2.4.4 Part 4
In the fourth part, an empirical and a simulation study were undertaken to implement automatic boom functions for the boom out and boom in motions. In the empirical study, a skilled operator was analyzed in terms of how he used the knuckle boom and its different parts during normal forwarder work. The position of each joint on the boom was measured. From the empirical study, motion trajectories of boom out and boom in functions were determined. These solutions were then analyzed in the DFMS. The boom in and boom out functions were evaluated in the KSS from Part 1. The KSS was further developed to handle different pre-defined paths. The results are reported in condensed form in Paper D.

2.4.5 Part 5
In the fifth part, an empirical and a simulation study were undertaken to investigate if the results achieved in Part 4 were reliable. In the empirical study, a skilled operator moved a real crane along pre-defined paths (similar to those defined in Part 4) for the boom out and boom in motions. The position of each joint and the pressure in the cylinders for each joint on the boom were measured. From the measured position and pressure, the speed and force used for each joint was calculated. The speeds from each joint were repeated in the DFMS. The results from the speeds of each joint and the corresponding force in the DFMS were then compared with the results from the test on the real forwarder. This part represents the final deliverable of this PhD project, and includes a cross analysis of the previously reported work.
Chapter 3
Kinematic control of redundant knuckle booms

This chapter describes three different algorithms for addressing the redundancy of the knuckle boom. The algorithms were developed, evaluated by an own-developed kinematic simulation software and reported in a previous licentiate thesis, Löfgren (2004).

3.1 BASIC EQUATIONS

The knuckle boom on a forwarder is velocity controlled by means of a 3 DOF (Degrees Of Freedom) joystick, operating in a cylindrical \((r, \theta, z)\) or Cartesian \((x, y, z)\) coordinate system (Figures 3.1 and 3.2)
In both coordinate systems the z-axis coincides with the pillar of the knuckle boom. In the Cartesian coordinate system the swing of the manipulator is included in the kinematic control.

When working in the cylindrical coordinate system, the control of the $\theta_0$ - joint (Figure 3.3) is separated from the control of the other joints.

![Manipulator geometry](image)

**Figure 3.3: Manipulator geometry.**

When working in the Cartesian coordinate system, with commanded $\dot{x}$, $\dot{y}$ and $\dot{z}$, it is possible to find the commanded $\dot{\theta}_0$ by differentiating $\theta_0 = \arctan(y/x)$ and using $r = \sqrt{x^2 + y^2}$.

$$\dot{\theta}_0 = (-s_0 \dot{x} + c_0 \dot{y}) / r$$

where $s_0 = \sin \theta_0$, $c_0 = \cos \theta_0$

The commanded $\dot{r}$ is found by differentiating $r = \sqrt{x^2 + y^2}$

$$\dot{r} = c_0 \dot{x} + s_0 \dot{y}$$

The $\dot{x}$ and $\dot{y}$ increments are now expressed as increments of $\dot{\theta}_0$ and $\dot{r}$, and the study can concentrate on the kinematic control in the r-z plane, where the redundant DOF is used.

A vector $\underline{\theta}$ is defined:

$$\underline{\theta} = [\theta_1, \theta_2, d_3]^T$$

The tool center point (TCP, see Figure 3.3) coordinates r and z defines the vector $x$: 

20
\[ x = [r, z]^T \]  
\[ (3.4) \]

Fig. 3.3 gives the following relationship:
\[ r = d_1 s_1 + (d_2 + d_3)s_{12} \]  
\[ (3.5) \]
\[ z = d_0 + d_1 c_1 + (d_2 + d_3)c_{12} \]  
\[ (3.6) \]

where \( s_{12} = \sin(\theta_1 + \theta_2) \), \( c_{12} = \cos(\theta_1 + \theta_2) \).
\[ \dot{r} = d_1 c_1 \dot{\theta}_1 + (d_2 + d_3)c_{12}(\dot{\theta}_1 + \dot{\theta}_2) + s_{12} \ddot{d}_3 \]  
\[ (3.7) \]
\[ \dot{z} = -d_1 s_1 \dot{\theta}_1 - (d_2 + d_3)s_{12}(\dot{\theta}_1 + \dot{\theta}_2) + c_{12} \ddot{d}_3 \]  
\[ (3.8) \]

or in matrix form:
\[ \dot{x} = J \ddot{\theta} \]  
\[ (3.9) \]

where the Jacobian J (2x3 – matrix) has elements \( j_{11}, j_{12}, ..., j_{23} \):
\[ \begin{align*}  
  j_{11} &= d_1 c_1 + (d_2 + d_3)c_{12} \\
  j_{12} &= (d_2 + d_3)c_{12} \\
  j_{13} &= s_{12} \\
  j_{21} &= -d_1 s_1 - (d_2 + d_3)s_{12} \\
  j_{22} &= -(d_2 + d_3)s_{12} \\
  j_{23} &= c_{12} 
\end{align*} \]  
\[ (3.10) \]

Since the Jacobian J is non square, the matrix cannot be directly inverted. The problem can be solved by introducing a constraint.

Three different control strategies will be studied in Sections 3.2, 3.3 and 3.4, and as introduced in the following sub-sections.

### 3.1.1 Maximum velocity

In some applications, velocity is of major importance. However, it is not possible to have maximum velocity as a constraint since the operator will not use maximum velocity at all times, but will utilize the highest possible velocity only when he/she finds it necessary. The used kinematic control law is based on a weighted pseudo inverse of the Jacobian, plus extra calculations when one or two joints are at their speed or position limits. This strategy is non-conservative—when repeating the same work cycle several times, the joint configurations will not be repeatable and the manipulator can run into unfavorable configurations. The strategy is non-conservative in the sense that a closed
path of the TCP in rectilinear space does not yield a closed path of the joints in joint space.

3.1.2 Maximum lifting capacity

In other applications, velocity is of minor importance, but static (low velocity) lifting capacity is essential. For this case, the kinematic control is based on an optimization study of the lifting capacity (Löfgren, 1989), based on the force or torque characteristics and the geometrical arrangements of the cylinders.

3.1.3 Dynamic programming

In some manipulator applications the start point and the end point are known, and at the same time the boom is free to use the whole workspace without concern about obstacles. By applying a global approach it is possible to reach an optimal solution. By using dynamic programming, the minimum time between two points in the workspace is identifiable. This approach may be useful when more semi- or fully automatic functions are introduced on forestry machines.

3.2 MAXIMUM VELOCITY

3.2.1 Kinematic Control Law

High velocity is essential in some applications, providing a low cycle time. This means that some of the joints will work at their velocity limits, when maximum velocity is desired. However, since the operator does not always want maximum velocity, a constraint is needed to solve the problem with the non-invertible Jacobian when all joints are working below their velocity limits. The quite popular pseudo inverse solution will be used:

Equation (3.9) can be written:

\[
\dot{\theta} = J^\# \dot{x}
\]

with

\[
J^\# = J^\top \left( J J^\top \right)^{-1}
\]

The pseudo inverse matrix \( J^\# \) implies that a cost function \( C \) is minimized (Noble, 1975):

\[
C(\dot{\theta}) = \dot{\theta}^\top \dot{\theta} = \dot{\theta}_1^2 + \dot{\theta}_2^2 + \dot{d}_3^2 = \text{minimum}
\]

This is not a reasonable cost function, since, the angular speed \( \dot{\theta}_1, \dot{\theta}_2 \) and the prismatic speed \( \dot{d}_3 \) are treated as equivalent, even though they do not even have the same dimension. A symmetric, positive definite weighting matrix \( W \) is therefore introduced (Chen et al., 1995):
\[ C(\dot{\theta}, W) = \dot{\theta}^T W \dot{\theta} \] (3.14)

A diagonal matrix is a sound choice here since the joints are independent of each other:

\[
W = \begin{bmatrix}
    w_{11} & 0 & 0 \\
    0 & w_{22} & 0 \\
    0 & 0 & w_{33}
\end{bmatrix}
\] (3.15)

giving

\[ C(\dot{\theta}) = w_{11} \dot{\theta}_1^2 + w_{22} \dot{\theta}_2^2 + w_{33} \dot{\theta}_3^2 \] (3.16)

The choice of diagonal elements in \( W \) is in this case chosen according to equation (3.17):

\[ w_{11} \dot{\theta}_{1\text{max}}^2 \approx w_{22} \dot{\theta}_{2\text{max}}^2 \approx w_{33} \dot{\theta}_{3\text{max}}^2 \] (3.17)

which gives the three joints equal weight when they are working at their maximum velocities \( \left( \dot{\theta}_{1\text{max}}, \dot{\theta}_{2\text{max}}, \dot{\theta}_{3\text{max}} \right) \).

The weighting matrix \( W \) can be position and/or angle dependent:

\[ W = W(x, \theta) \] (3.18)

This approach has not been used in the simulations but can be used in a real case.

The weighting matrix \( W \) modifies equation (3.11) and (3.12):

\[ \dot{\theta} = Q \ddot{x} \] (3.19)

with (Chen et al., 1995)

\[ Q = W^{-1} J^T \left( JW^{-1} J^T \right)^{-1} \] (3.20)

We find, after some calculations:

\[ \left( JW^{-1} J^T \right)^{-1} = \frac{1}{D} \begin{bmatrix}
    d_{11} & d_{12} \\
    d_{21} & d_{22}
\end{bmatrix} \] (3.21)

with
\[
\begin{align*}
\sum_{j} j_{i,j}^2 &= w_{11} + j_{22}^2 / w_{33} \\
\sum_{j} j_{2,j}^2 &= -(j_{11} j_{21} / w_{11} + j_{12} j_{22} / w_{13} + j_{13} j_{23} / w_{33}) \\
\sum_{j} j_{3,j}^2 &= w_{11} + j_{22}^2 / w_{33} + j_{33}^2 / w_{33} \\
D &= d_{11} d_{22} - d_{12}^2
\end{align*}
\]

(3.22)

The joints have maximum velocities \(\dot{\theta}_{i_{\text{max}}}^{\text{max}}\), \(\dot{\theta}_{2_{\text{max}}}^{\text{max}}\), and \(\dot{\theta}_{3_{\text{max}}}^{\text{max}}\), respectively. If a joint, such as joint no.1, receives a command signal \(\dot{\theta}_{1c}\), with \(\dot{\theta}_{1c}^{\text{max}} > \dot{\theta}_{1_{\text{max}}}^{\text{max}}\), the velocity limitations will cause a position error. This problem is solved in the following way:

Introduce:

\[
\alpha_i = \frac{|\dot{\theta}_i|}{\dot{\theta}_{i_{\text{max}}}}; i = 1,2,3
\]

(3.24)

In a practical case, the geometrical arrangement of the \(\theta_i\)- and \(\theta_2\)-joints may cause \(\dot{\theta}_{1_{\text{max}}}^{\text{max}}\) and \(\dot{\theta}_{2_{\text{max}}}^{\text{max}}\) to be functions of \(\theta_1\) and \(\theta_2\), respectively, and the directions. In this case joint speeds are set equal in both directions. This is due to the fact that for each machine, the application of the hydraulic pressure and flow are different. As a result, the joint speeds are set to suit one typical pressure and flow.

The largest \(\alpha_i\) value can be found, \(\alpha_{\text{max}}\):

\[
\alpha_{\text{max}} = \max \{\alpha_i\}
\]

(3.25)

If \(\alpha_{\text{max}} \leq 1\), there are no velocity limitations, and \(\dot{\theta}_{1c}\), as determined by equations (3.11) and (3.19), are used.
Cylindrical Coordinate System

*Two DOF*

Assume first, that the joystick is working in the cylindrical coordinate system, and that the \( \theta_0 \) joint is separately controlled.

Assume \( \alpha_1 = \alpha_{\text{max}} > 1 \)

Choose \( \dot{\theta}_1 = \dot{\theta}_{\text{max}} \) if \( \dot{\theta}_1 > 0 \) and \( \dot{\theta}_1 = -\dot{\theta}_{\text{max}} \) if \( \dot{\theta}_1 < 0 \). Since one DOF is lost, making the system non-redundant, the calculation of \( \dot{\theta}_2 \) and \( \dot{d}_3 \) is straightforward (\( r_c \) and \( z_c \) are commanded velocities):

\[
\begin{pmatrix}
\dot{r}_c \\
\dot{z}_c
\end{pmatrix} = J \begin{pmatrix}
\pm \dot{\theta}_{\text{max}} \\
\dot{\theta}_{2c} \\
\dot{d}_{3c}
\end{pmatrix}
\]  

(3.26)

giving

\[
\begin{pmatrix}
\dot{\theta}_{2c} \\
\dot{d}_{3c}
\end{pmatrix} = \frac{1}{D_1} \begin{pmatrix}
j_{23} & -j_{13} \\
-j_{22} & j_{12}
\end{pmatrix} \begin{pmatrix}
\dot{r}_c \\
\dot{z}_c
\end{pmatrix} + \begin{pmatrix}
\dot{\theta}_{\text{max}} \\
\dot{\theta}_{\text{max}}
\end{pmatrix}
\]

(3.27)

with

\[
D_1 = j_{12}j_{23} - j_{13}j_{22}
\]

(3.28)

*One DOF*

To find out if the new values for \( \dot{\theta}_{2c} \) and \( \dot{d}_3 \) given by (3.27) will exceed their maximum velocities, \( \alpha_2 \) and \( \alpha_3 \) must be calculated again giving \( \alpha'_{2c} \), \( \alpha'_{3c} \) and \( \alpha'_{\text{max}} \). If \( \alpha'_{\text{max}} \leq 1 \), control law (3.27) is used. If \( \alpha'_{\text{max}} > 1 \), two joints, \( \theta_1 \) and \( \theta_2 \) or \( d_3 \) must be working at their maximum velocities and hence there is only one DOF, and two DOFs are necessary to follow a commanded path in the r-z plane.

Assume that \( \alpha'_{\text{max}} = \alpha'_{2\text{max}} \), i.e. \( |\dot{\theta}_2| = \dot{\theta}_{\text{max}} \). Choose \( \dot{\theta}_2 = \dot{\theta}_{\text{max}} \) if \( \dot{\theta}_2 > 0 \) or \( \dot{\theta}_2 = -\dot{\theta}_{\text{max}} \) if \( \dot{\theta}_2 < 0 \). The commanded velocity must be scaled by a factor \( \beta < 1 \) since the commanded path cannot not follow with two joints at their maximum speeds, due to the loss of two DOFs:
\[
\beta \begin{bmatrix} \dot{r}_c \\ \dot{z}_c \end{bmatrix} = \begin{bmatrix} j_{11} & j_{12} & j_{13} \\ j_{21} & j_{22} & j_{23} \end{bmatrix} \begin{bmatrix} \pm \dot{\theta}_{1\text{max}} \\ \pm \dot{\theta}_{2\text{max}} \\ \dot{d}_3 \end{bmatrix} \tag{3.29}
\]

Equation (3.29) has two unknown, \( \beta \) and \( \dot{d}_3 \).

It can be found that:

\[
\beta = \frac{j_{23}k_1 - j_{13}k_2}{j_{23}\dot{r}_c - j_{13}\dot{z}_c} \tag{3.30}
\]

and

\[
\dot{d}_3 = \frac{\beta \dot{r}_c - k_1}{j_{13}} \tag{3.31}
\]

where

\[
k_1 = \pm j_{11}\dot{\theta}_{1\text{max}} \pm j_{12}\dot{\theta}_{2\text{max}} \tag{3.32}
\]

\[
k_2 = \pm j_{21}\dot{\theta}_{1\text{max}} \pm j_{22}\dot{\theta}_{2\text{max}} \tag{3.33}
\]

If, instead \( \alpha'_{\text{max}} = \alpha'_{3\text{max}} \), similar calculations will apply.

Similar calculations will also apply if there is a speed limitation in \( \theta_2 \) or \( d_3 \), i.e., \( \alpha_2 = \alpha_{\text{max}} \) or \( \alpha_3 = \alpha_{\text{max}} \).

Cartesian Coordinate System

A first step is to determine if \( \alpha_o > 1 \). If so, the velocities of all the joints are scaled down by dividing by \( \alpha_o \). \( \alpha_o > 1 \) means that the commanded velocity cannot be reached.

Then proceed exactly as in Section 3.2.3, but it is necessary to multiply \( \dot{\theta}_{oc} \) by \( \beta \) if \( \beta < 1 \).

### 3.2.3 Mechanical Limits

If one of the \( \theta_1 \)-, \( \theta_2 \)- or \( d_3 \)-joints reaches a mechanical limit, the redundant degree of freedom is lost, but the boom can still follow a desired path (until a second joint reaches a mechanical limit). To solve this, \( \dot{\theta} \) is first calculated by means of equation (3.19). If the \( \theta_1 \)-joint is at a mechanical limit and equation (3.19) shows that the joint should pass through the mechanical limit, a second calculation would have to be performed.
exactly as in Section 3.2.2, but with $\dot{\theta}_{\text{max}}$ replaced by 0. Similar calculations are made for $\theta_2$ and $d_3$ when they reach their mechanical limits.

In order to avoid large transients when approaching a mechanical limit, the maximum velocity used for calculations is decreased for that joint.

### 3.3 MAXIMUM LIFTING CAPACITY

#### 3.3.1 Kinematic Control Law

When working with heavy loads, the extra degree of freedom can be used for maximizing lifting capacity. The kinematic control algorithm for this is based on computer studies of lifting capacity (Löfgren, 1989) as a function of $\theta_1$, $\theta_2$, and $d_3$, using the hydraulic cylinder characteristics and the geometrical arrangements of the hydraulic cylinders. From the studies it is possible to analyze how lifting capacity is dependent on the prismatic function $d_3$.

Figure 3.4 shows how $d_3$ should be chosen for maximum lifting capacity for a specific manipulator. Except for the lower left and upper middle part of the work area, the three curves $d_3 = (0, 50 \text{ and } 100\%)$ can be approximated by circles. In other cases where the
$d_3$-curves are more complicated, a look-up table for $d_3 = d_3(r, z)$ plus interpolation can be used.

In order to avoid unnecessarily large accelerations in $d_3$, some “smoothing” of the optimal $d_3(r, z)$ function may be introduced, especially in areas in the r-z plane where the optimum is flat, i.e., where the effect of $d_3$ on the lifting capacity is small. The proposed kinematic control function of $d_3$ is shown in Figure 3.5.

\[
\rho_{\text{max}} = d_3 = d_{\text{max}}
\]

Figure 3.5: $d_3$ as a function of TCP’s position.

Figure 3.5 gives:

\[
\rho = \sqrt{(r - r_c)^2 + (z - z_c)^2}
\]  

(3.34)

where $r_c$ and $z_c$ are the coordinates for the centre point of the circles in the proposed kinematic control function.

The approach to describe the transition between $\rho_{\text{min}}$ and $\rho_{\text{max}}$ should be expressed as a “smooth” function. The choice of the coordinates for $r_c$ and $z_c$ can be done according to Figure 3.4, Figure 3.5 and the geometrical data from a specific boom.

### 3.3.2 Smoothing function

In the zone where $d_3$ is active it should vary smoothly without large accelerations. The chosen function is given in equation (3.34) according to Figure 3.6. This function has $\frac{dd_3}{d\rho} = 0$ for $\rho = \rho_{\text{min}}$ and for $\rho = \rho_{\text{max}}$, thus avoiding jumps in the linear joint’s velocity.
Figure 3.6: $d_3$ as a function of $\rho$.

The smoothing function depicted in Figure 3.6 is given by equations (3.35) to (3.37).

$$d_3 = \frac{d_{3\max}}{2} \left[ 1 + \frac{3(\rho - p)}{2q} - \frac{(\rho - p)^3}{2q^3} \right]$$

(3.35)

with

$$p = \frac{\rho_{\max} + \rho_{\min}}{2}$$

(3.36)

$$q = \frac{\rho_{\max} - \rho_{\min}}{2}$$

(3.37)

Differentiating equation (3.35) gives

$$\dot{d}_3 = f(r, \rho)\dot{r} + g(z, \rho)\dot{z}$$

(3.38)

with

$$f(r, \rho) = c(r - r_3) / \rho \quad \rho_{\min} \leq \rho \leq \rho_{\max}$$

$$g(z, \rho) = c(z - z_3) / \rho \quad \text{for} \quad \rho_{\min} \leq \rho \leq \rho_{\max}$$

$$f(r, \rho) = g(z, \rho) = 0 \quad \rho < \rho_{\min}; \rho > \rho_{\max}$$

(3.39)

where

$$c = \frac{3d_{3\max}}{4q} \left[ 1 - \frac{(\rho - p)^2}{2q^2} \right]$$

(3.39)
Independent of the characteristics that are used for \( d_3 \), the notations can be simplified:

\[
 f(r, \rho) = f, \quad g(z, \rho) = g.
\]

Using \( \dot{d}_3 \) from equation (3.38) in equation (3.9) gives:

\[
\begin{bmatrix}
\dot{r} \\
\dot{z}
\end{bmatrix} =
\begin{bmatrix}
 j_{11} & j_{12} & j_{13} \\
 j_{21} & j_{22} & j_{23}
\end{bmatrix}
\begin{bmatrix}
 \dot{\theta}_1 \\
 \dot{\theta}_2
\end{bmatrix}
\begin{bmatrix}
 fr + gz
\end{bmatrix}
\]

(3.40)

Rewriting equation (3.40) gives:

\[
\begin{bmatrix}
 j_{11} & j_{12} \\
 j_{21} & j_{22}
\end{bmatrix}
\begin{bmatrix}
 \dot{\theta}_1 \\
 \dot{\theta}_2
\end{bmatrix} =
\begin{bmatrix}
 1 - j_{13}f & -j_{13}g \\
 -j_{23}f & 1 - j_{23}g
\end{bmatrix}
\begin{bmatrix}
 \dot{r} \\
 \dot{z}
\end{bmatrix}
\]

or

\[
\begin{bmatrix}
 \dot{\theta}_1 \\
 \dot{\theta}_2
\end{bmatrix} =
\begin{bmatrix}
 j_{11} & j_{12} \\
 j_{21} & j_{22}
\end{bmatrix}^{-1}
\begin{bmatrix}
 1 - j_{13}f & -j_{13}g \\
 -j_{23}f & 1 - j_{23}g
\end{bmatrix}
\begin{bmatrix}
 \dot{r} \\
 \dot{z}
\end{bmatrix}
\]

(3.41)

(3.42)

Equations (3.38), (3.42) and (3.10) give, after some calculations, the kinematic control law:

\[
\dot{\theta} = P\ddot{x}
\]

(3.43)

where the (3x2) P-matrix has the following elements:

\[
\begin{bmatrix}
p_{11} = (s_{12} - f) / (d_1 s_2) \\
p_{12} = (c_{12} - g) / (d_1 s_2) \\
p_{21} = [d_1 f (c_2 f - s_1) / (d_2 + d_3) - s_{12}] / (d_1 s_2) \\
p_{22} = [d_1 g (c_2 g - c_1) / (d_2 + d_3) - c_{12}] / (d_1 s_2) \\
p_{31} = f \\
p_{32} = g
\end{bmatrix}
\]

(3.44)

A singularity appears for \( \theta_3 = 0 \) (i.e., a “straight elbow”), and is caused by the applied constraint on \( d_3 \). With a kinematic control law using the pseudo inverse matrix, this singularity will not exist within the workspace, only on the workspace envelope.
3.3.3 Velocity limitations

If one of the joints reaches its maximum velocity, a simple “scaling” (cf. Section 3.2.2) of the command signals is necessary, if a specified path in the workspace is to be followed.

3.3.4 Mechanical limits

If one of the $\theta_1$, $\theta_2$ or $d_3$-joints reaches a mechanical limit, the redundant degree of freedom is lost, but can still follow a desired path until a second joint reaches a mechanical limit (see Section 3.2.3).

3.4 DYNAMIC PROGRAMMING

In some manipulator applications the start point and the end point are known, and at the same time the boom is free to use the whole workspace without concern about obstacles. By using a global approach it is possible to find an optimal solution.

Bellman (1957), the idea is presented with dynamic programming, which is a problem-solving approach that proceeds by combining the solutions to sub-problems that are not independent of each other (i.e., they share common sub-sub-problems). A dynamic programming algorithm stores the optimal solutions to sub-sub-problems, thereby avoiding computing them several times. If these ideas are transformed to the manipulator example where the manipulator TCP moves from one known point to another known point in the workspace (Figure 3.7), a short time between two points is possible to find.

Since dynamic programming is based on numerical calculations, the straight path in the work area is discretized into $N$ steps, $(\Delta r, \Delta z)$ and $N+1$ stages, $S_0, S_1, \ldots, S_N, S_0$, representing the positions $(r, z), (r + \Delta r, z + \Delta z), \ldots, (r + N\Delta r, z + N\Delta z)$; see Figure 3.7. In addition, one of the DOFs will be discretized into $M$ steps. Choose $d_3$ with the values $d_{3\text{min}}, d_{3\text{min}} + \Delta d_3, \ldots, d_{3\text{min}} + M\Delta d_3 (= d_{3\text{max}})$. Between two stages, $d_3$ can change $-q\Delta d_3, -(q-1)\Delta d_3, \ldots, 0, \ldots, (q-1)\Delta d_3$ or $q\Delta d_3$, where $q$ is a given integer. The selection of $\Delta d_3, \Delta r, \Delta z$ and $q$ is important. Small increments $(\Delta d_3, \Delta r, \Delta z)$ and a large $q$ are necessary for reaching the optimal solution, but can lead to a large computational time. Therefore, empirical tests will give the necessary values of $\Delta d_3, \Delta r, \Delta z$ and $q$.

For pedagogical reasons, $q = 1$ has been used in Figure 3.7, where only a few nodes within the discretized workspace are shown.

Each stage $S_1, \ldots, S_N$ is associated with $M+1$ nodes. In every node, $T_{ij}$, $i = 0, 1, \ldots, n, \ldots N; j = 0, 1, \ldots, m, \ldots M$ the coordinates of the three DOFs must be calculated:

$$d_3(n, m) = d_{3\text{min}} + m\Delta d_3 \quad \text{(independent of stage n)}$$
$\theta_1(n,m)$ and $\theta_2(n,m)$ are determined by means of equations (3.5) and (3.6) and tedious calculations:

$$\theta_2(n,m) = \tan^{-1} \left( \frac{\sin \theta_2(n,m)}{\cos \theta_2(n,m)} \right)$$  \hspace{1cm} (3.45)

where

$$\cos \theta_2(n,m) = \left( \frac{(r + n\Delta r)^2 + (z + n\Delta z)^2 - d_1^2 - (d_2 + d_{3\min} + m\Delta d_3)^2}{2d_1(d_2 + d_{3\min} + m\Delta d_3)} \right)$$  \hspace{1cm} (3.46)

$$\sin \theta_2(n,m) = \sqrt{1 - \cos^2 \theta_2(n,m)}$$  \hspace{1cm} (3.47)

$$\theta_1(n,m) = \tan^{-1} \left( \frac{a(r + n\Delta r) + b(z + n\Delta z)}{-b(r + n\Delta r) + a(z + n\Delta z)} \right)$$  \hspace{1cm} (3.48)

with

$$a = d_1 + (d_2 + d_{3\min}m\Delta d_3)\cos \theta_2(n,m)$$  \hspace{1cm} (3.49)

$$b = (d_2 + d_{3\min} + m\Delta d_3)\sin \theta_2(n,m)$$  \hspace{1cm} (3.50)

Now, examine at one “transition”, such as from $T_{0,m-1}$ to $T_{1,m}$:

$\theta_1$ and $\theta_2$ will have increment

$$\Delta \theta_j(0,m-1;1,m) = \theta_j(1,m) - \theta_j(0,m-1) ; j = 1,2$$  \hspace{1cm} (3.51)

$d_3 : s$ increment is

$$\Delta d_3(0,m-1;1,m) = \Delta d_3$$  \hspace{1cm} (3.52)

The shortest time for the three transitions $\Delta \theta_1, \Delta \theta_2$ and $\Delta d_3$ will be

$$t_j(0,m-1;1,m) = \frac{\Delta \theta_j(0,m-1;1,m)}{\dot{\theta}_{j\max}} ; j = 1,2$$  \hspace{1cm} (3.53)

$$t_3(0,m-1;1,m) = \frac{\Delta d_3(0,m-1;1,m)}{\dot{d}_{3\max}}$$  \hspace{1cm} (3.54)
The shortest time for the transition from $T_{0,m-1}$ to $T_{1,m}$ is determined by the slowest of the three DOFs:

$$t_{\min}(0,m-1;1,m) = \max_{j=1,2,3}(t_j(0,m-1;1,m))$$  \hfill (3.55)

By calculating in a similar way $t_{\min}(0,m;1,m)$ and $t_{\min}(0,m+1;1,m)$, the minimum time is finally found for the transition from stage 0 to node $T_{1,m}$:

$$t_{\min}(0;1,m) = \min_{j=0,1,2,3}[t_{\min}(0,m+j;1,m)]$$  \hfill (3.56)

This time is assigned, in the memory, to node $T_{1,m}$.

Similar calculations are made for all nodes in stage $S_1$ and memorized.

In the calculations for transition from stage $S_1$ to stage $S_2$, the minimum transition time assigned to each node is added, for example:

$$t_{\min}(1;2,m) = \min_{j=0,1,2,3}[t_{\min}(0;1,m+j) + t_{\min}(1,m+j;2,m)]$$  \hfill (3.57)

The calculations are repeated until stage $N$, giving $t_{\min}(N-1;N,m)$ for $m = 0,1,...,M$.

The optimal (minimum) time for the transition from $S_0$ to $S_N$ is

$$t_{\min} = \min_{m=0,1,...,M}[t_{\min}(N-1;N,m)]$$  \hfill (3.58)

The optimal path from stage $S_0$ to each node must be memorized such that the final optimal path from $(r,z)$ to $(r+N\Delta r, z + N\Delta z)$ can be achieved.
If $\Delta d_3$ is larger than the increment of the TCP, in most cases $d_3$ will not move. We have chosen the value of $\Delta d_3$ to one-tenth of the increment in the TCP path, i.e. from one stage to the next.

### 3.5 CONCLUSION

Algorithms for the computation of the inverse kinematics of kinematically redundant hydraulic manipulators have been investigated. The manipulator used in the simulation study is a 4 DOF hydraulic forestry machine manipulator. The aim was to compare kinematic control algorithms that would resolve the redundancy and control the manipulator arm.
Three methods resolving the manipulator redundancy at the velocity level were examined: maximum lifting capacity, maximum velocity (local optimization) and dynamic programming (global optimization).

The maximum velocity algorithm gives shorter cycle times than the maximum lifting capacity algorithm, but this is not a general conclusion. In other applications, the two methods should be carefully compared, since there are many parameters affecting the result. In the maximum lifting capacity method, the parameters $r^c$, $z^c$, $\rho_{\min}$, $\rho_{\max}$ are affecting the result. In the maximum velocity method, the parameters $w_{11}, w_{22}, w_{33}$ are affecting the result.

Dynamic programming demonstrates the shortest possible cycle times when relaxing the requirements of constant TCP velocity along the path, but the start and the end position of the trajectories must be known.

A detailed description of the algorithms and also an analysis of their functionality are presented in Löfgren (2004) and in condensed form in Paper A.
Chapter 4
Simulations

The use of computer simulations for the developed algorithms in Chapter 3 is essential for the work in this thesis. This chapter describes the use of simulators in forestry and in excavator applications, the simulation software and the dynamic forest machine simulator, and explores the problem of using simulators in research.

4.1 The use of human-in-the-loop simulators

What follows is a short description of how the human-in-the-loop simulators have been used in the past and how they are being used today. In a simulator, a human error in a complicated process or any malfunction of the simulated vehicle/process will not have any other consequence besides the trainee or vehicle/process developer learning from the mistake or complicated situation. In addition, the costs of testing new concepts are lower in a simulator than by building a prototype of a new machine, car, airplane, etc.

The first known simulator of a kind similar to what is discussed here was a mechanical flight simulator developed in the US in 1929 (Alm, 2007). In the 1960s computers became involved, enabling the development of more elaborate scenarios. Simulators for use in design activities appeared in the 1970s and thereafter simulations for research purposes started. In the late 1970s, simulator activities for more design-oriented activities were established. In the 1980s and 1990s, the use of simulators became important for marketing issues. By the late 1990s, simulators were used to verify new functions and subsystems, instead of verifying them using a real application. A summary of the history of simulation can be seen in Figure 4.1.

![Figure 4.1: A historical perspective on the evolution of flight simulator usage and of the simulator's technical level. (Alm, 2007)](image)

With the introduction of powerful and cheap personal computers to replace expensive workstations, it became possible for companies, universities and research institutes to
invest in simulators. This, in turn, led to breakthroughs in the use of simulators in many different areas.

Human-in-the-loop simulators can be found in many areas of human activity, such as space, automobiles, trucks, flight, train, medicine (surgery), harbor cranes, fork trucks, forestry, excavators, tractors, combines, motorcycles, graders, wheel loaders, and so on. A full discussion on all of these areas is beyond the scope of this thesis. The focus in this brief overview is on forestry and excavator simulators, since they use booms to perform their work.

4.2 Commercial human-in-the-loop simulators

Most commercial human-in-the-loop simulators for forestry and excavators are simulators used for training purposes. Some of them are listed below.

4.2.1 Excavator simulators

Caterpillar, John Deere and VCE (see Figure 4.2) sell simulators in addition to their regular line of excavation products. All of them describe simulators as a cost-effective, safe and efficient way of training new excavator operators in a risk-free environment. The training simulators feature real-world situations, job site hazards, safety violations, hand signals, equipment damage, budget-based scoring, and replica machine controls. There are several detailed and realistic training activities that teach proper operator techniques, machine controls, and safe operation in a virtual job site.


Figure 4.2: Example of three different excavator simulators. Left: the Caterpillar excavator simulator (www.caterpillar.com). Center: the John Deere excavator simulator (www.johndeere.com). Right: the VCE excavator simulator (www.volvo.com).

4.2.2 Knuckle boom simulators

TYA sells simulators that use redundant knuckle booms (see Figure 4.3). TYA is the Vocational Training and Working Environment Council (Transport Trades), a not-for-profit association established by the labour market parties within Sweden’s transport sector.
4.2.3 Forestry simulators

Two different forestry machine simulators are shown below.

![Forestry simulators](image)

Figure 4.4: Left: the John Deere forest machine simulator ([www.johndeere.com](http://www.johndeere.com)). Right: the Ponsse forest machine simulator ([www.ponsse.fi](http://www.ponsse.fi)).

4.3 Human-in-the-loop simulators in research

Most of the research on simulators that can be found are in the field of flight simulators, car simulators and medical simulators, and a few in forest or excavator simulators. It is very difficult or almost impossible to obtain any information from machine manufacturers on their research work in simulators.

When simulators became more accessible by the end of the 1990s, the research began to examine if the simulators where good enough for training of forest machine operators. Lapointe (2000) performed a comparative study on the results between traditional training (where students go directly from the classroom to real machine operation in the woods) and a new virtual reality (VR)-augmented training. The results indicate that the addition of 25 hours of hands-on VR training increases the volume of wood harvested by 23%, and reduces repair and maintenance costs by 26% during the first month of operation in the forest. The use of VR also allowed the precise recording and monitoring of the evolution of trainees’ performance during their training sessions, showing learning curves that decreased with time for all of the defined performance criteria (execution time, error rate and precision). Ovaskainen (2009) shows that productive working techniques taught via simulator can increase productivity by 10 to 15%. The
results also indicated that the virtual harvester simulators are applicable for harvester training when the trainees are conscious of the limitations of the simulators. The ProForSim project Ranta (2009) shows that the trainees felt a strong connection to the authentic harvester work and that alternating use of the simulators and the real harvester was meaningful.

No research was found on simulation of semi-automation of forest machines or excavators in a human-in-the-loop simulator, except for studies performed at Skogforsk (Brander et al., 2004; Egermark, T. 2005, Egermark & Löfgren, 2005).

At the 2009 Elmia Wood Fair John Deere Forestry presented a boom tip control in their forest machine simulator. The boom tip control was based on the maximum lifting capacity algorithm. John Deere wanted to assess interest in the boom tip control among the experienced operators that visited the fair.

4.4 Kinematic simulations

Three types of simulations have been performed during this thesis work. One used the kinematic simulation software (KSS), where the kinematic equations have been analyzed. In another, operators have been testing the developed algorithms in the DFMS (Dynamic Forest Machine Simulator). In the third simulation, the KSS has been connected to the DFMS to test automatic boom functions. The DFMS is a real-time simulator, meaning that all boom movements are based on the input from an operator and not on any pre-implemented sequences. All movements of the forest machine in the DFMS are based on physical data such as weight, geometry, friction, moment of inertia etc.

Simulations were used to evaluate boom tip control algorithms and automatic boom functions. In Part 1, only the KSS was used; in Part 4, the KSS was used in conjunction with the DFMS.

4.4.1 The Kinematic simulation software

The KSS was developed in two steps, in Parts 1 and 4 respectively.

STEP 1

It was important to verify the simplified boom control presented in Part 1 (Paper A). Therefore, a kinematic simulation program was developed to evaluate different algorithm approaches. A special Graphical User Interface (GUI) was also developed (Figure 4.5).
Figure 4.5: The GUI menu (in Swedish) of the kinematic simulation software in Part 1.

With the GUI, the following functions are available:

- Evaluate three different algorithms, maximum lifting capacity, maximum speed and dynamic programming;
- Define any length of the boom parts, to enable analysis of any type of knuckle booms;
- Define any start point in the working area of the boom;
- Define limitations of:
  - minimum and maximum joint angle; and
  - maximum joint speeds and accelerations.
- Set parameters for velocity ramps at the end position of each joint; and
- Graphical presentation of the results.

STEP 2

The proposed automatic functions in Part 4 (Paper D) were added to the kinematic simulation program. The GUI was further developed, as shown in Figure 4.6.
Automatic functions were added to the GUI. In addition to the functions in Part 1, the following functions are available:

- Start the automatic *boom in* and *boom out* motion functions;
- Remember the last position outside the load space of the forwarder;
- Automatically go to the last position outside the load space of the forwarder;
- Remember the last position in the load space of the forwarder;
- Automatically go to the last position in the load space of the forwarder; and
- Put the boom at transport position.

There is also a network connection between the KSS and the DFMS. In practice this means that the automatic functions are performed in the KSS and the output in the form of velocity references are then sent to the DFMS for activation of the boom (see Figure 4.7).
4.5 The dynamic forest machine simulator

The DFMS used in this thesis work is from Komatsu Forest AB. This simulator is made by Oryx Simulations AB. In addition to the forest machine simulators, Oryx Simulations also produces other simulators, such as those identified in Chapter 4.2.

4.5.1 Simulator setup

Operator seat setup

The setup of the DFMS at Skogforsk is a driver seat with all the electronics used in a real harvester and forwarder, (see Figure 4.8 and compare with Figure 1.7).

Figure 4.8: The simulator seat with electronic control units.

The simulator has no hydraulics such as in the forest machines, where the speed signals from the joysticks are transferred to the crane computer and then on to the hydraulic valves, to the left in Figure 4.9. Instead, the simulator has an electronic unit taking care of the speed signals from the joysticks, with the signals then being transferred to the simulator computer (see the right portion of Figure 4.9).
Figure 4.9: Left: Description of how the speed signals from the crane computer is transferred to the hydraulic valves. Right: Description of how the speed signals from the crane computer in the simulator chair are transferred to the simulator computers.

**Screen setup**

The simulator at Skogforsk has three screens with back projection, which provides a good field of view and makes the simulator suitable for different test applications (see Figure 4.10).

Figure 4.10: The Oryx Simulations AB dynamic forest machine simulator at Skogforsk.
Computers

The simulator at Skogforsk has four computers, one for each of the three screens, and one for calculations.

4.6 The dynamics in the simulator

To briefly understand how the dynamics in the simulator has been implemented by Oryx Simulations. A full discussion of how the method for computing the constraint forces is beyond the scope of this thesis.

Lacoursière’s thesis (2007) presents novel numerical methods for computing trajectories of classical mechanical systems of constrained multibodies subject to contacts, and a variety of constraints, kinematic or otherwise. The thesis also presents the interactive physics involved and how the equations are solved. Interactive physics is described as the combination of a numerical simulation with multimodal input devices driving multisensory output devices with short response times. This allows seeing, hearing, touching and steering a numerical simulation as it executes.

Numerical time integration of the equations of motion of a given physical system is the heart of interactive physics. The software implementing physics models and the time integrator is called a “physics engine”. An interactive physics application is typically a distributed, soft real-time system. Multimodal inputs from a user or a script are polled at various frequencies and communicated to the physics engine. The engine integrates the simulated system forward in time by a fixed amount and the new state is then to generate output signals for multisensory feedback. Soft real-time system means that the flow of time in the simulation is nearly identical to flow of wall clock time, and that response to inputs is quick and even, although not subjected to hard guaranteed bounds as in the case of a hard-time systems used in control engineering.

The linearized version of the regulated and stabilized stepping scheme for mixed systems with both holonomic and nonholonomic constraint is called the SPOOK method. A novelty of the SPOOK method (Lacoursière, 2007) used in the simulator is that it applies time discretization directly to analytical mechanics and therefore preserves physical symmetries and laws, such as conservation of energy, even for large time steps, which happens to be particularly important in interactive simulators. Many traditional approaches instead apply discretization to the continuous differential equations, but then the information about symmetries and conservation laws is lost unless they are introduced in an ad hoc manner. The method also prescribes a way of introducing constrained mechanics into the discrete variational framework, and a method for computing the corresponding constraint forces. The result is a discrete stepping scheme for positions and velocities.

The velocities are updated according to
SIMULATIONS

\[ v^{k+1} = v^k + hM^{-1}f + M^{-1}G^T \lambda \]  

(4.1)

where \( v^{k+1} \) is the new vector of velocities, \( v^k \) the old velocity, \( h \) is the size of the time step, \( M^{-1} \) is the inverse inertia (mass matrix) of the system, \( f \) is the sum of all external forces, \( M^{-1}G^T \lambda \) is the constraint force, with \( \lambda \) being the magnitude of the constraint force (referred to as the Lagrange multiplier of the constraint) and \( G^T \) is a matrix providing information about the constraint (called the Jacobian matrix). For example, when modeling a hinge constraint, this will provide an entry in a certain structure in the matrix \( M^{-1}G^T \) with information about the symmetry of the constraint, and which bodies are involved in the constraint. In fact, \( \lambda \) is actually an impulsive force since it is applied instantly (there is no \( h \) in front of it).

The formalism uses generalized system coordinates, so \( v \) is considered to be a vector of translational and rotational velocities of all bodies in the system and \( f \) and \( \lambda \) thus also contain entries describing torques. Correspondingly, the mass matrix describes both masses and moments of inertia of all bodies. Thus, the system coordinates are

\[
q_k = \begin{bmatrix}
    r_1 \\
    Q_1 \\
    r_2 \\
    Q_2 \\
    . \\
    . \\
    . \\
    r_n \\
    Q_n
\end{bmatrix}, \quad v_k = \begin{bmatrix}
    v_1 \\
    \omega_1 \\
    v_2 \\
    \omega_2 \\
    . \\
    . \\
    . \\
    v_n \\
    \omega_n
\end{bmatrix} \quad \text{and} \quad f_k = \begin{bmatrix}
    f_1 \\
    \tau_1 \\
    f_2 \\
    \tau_2 \\
    . \\
    . \\
    . \\
    f_n \\
    \tau_n
\end{bmatrix}  
\]  

(4.2)

Given the new velocities, new positions can be computed as

\[ q_{k+1} = q_k + hv_{k+1} \]  

(4.3)

As mentioned, \( q \) is a state vector and describes both position coordinates and angular coordinates. Assuming that there are no constraints, this scheme can be identified as the Velocity/Leapfrog stepping scheme. The main ingredient of SPOOK is thus that it is a method for computing the constraint forces \( \lambda \) used in the above stepping equations.

The simulator software computes the constraint forces of classical mechanical systems of constrained multibodies.

4.7 Simulator as a research tool

Normally the DFMS is used for the training and education of new forest machine operators. In this thesis, the DFMS was used as a research tool for evaluating simplified boom control and automatic boom movements. As a result, the problems that can occur when using a simulator for evaluation must be identified.
4.7.1 Controlled studies

The opportunity to conduct controlled studies is perhaps the greatest advantage of performing research with a simulator. Many external factors that would otherwise influence the studies can be minimized or avoided. When performing forestry machine studies in a simulator, disturbing factors such as weather, wind, terrain and stand variations are avoided. This means that tests can be repeated again and again with variation only of the most interesting variables or parameters, one at a time. By keeping the external variables constant, it is possible to study the main influencing factors.

In general, simulator studies are more efficient than field studies. Fewer operators are needed when repeated studies in the simulator are performed, since identical conditions are used for all operators. This has been concluded in a car simulator (Nilsson, 1993).

It is often difficult to perform comparative studies when there are different stand conditions, and it is practically impossible to find comparable stands. In the simulator, however, the same stand conditions can be employed over and over again.

Another factor is safety. New concepts or methods can be tested without harming the machine or the operator. The simulator provides the ability to evaluate simplified or automated functions. Yet another factor is that it is simpler to collect data from the simulator than in the field, particularly data on the operator.

However, there are other factors that may influence the results from studies performed in the simulator. A number of these factors are described below.

4.7.2 Physical limitations and realism

A simulator with an operator seat or cabin that does not move cannot reproduce vibrations from the boom movements and from traversing terrain. As such, there are limitations in the normal operator work in the simulator until the operator experiences the simulator as unrealistic. Another factor is that, for example, quick joystick movements that do not correspond to boom movements could cause simulator sickness for the operator. Nilsson (1993) found connections between simulator sickness and rapid movements and accelerations; driving on simulated terrain without the normal vibrations and slopes, for example, can encourage overly fast driving. Moreover, in a real forest machine there are normally delays in the machine’s system (e.g., in the hydraulic system), which must also be present in the simulator.

It is therefore very important that the operators who perform tests in the simulator are also given the opportunity to practice and learn the limitations of the simulator before performing their tests.

A perfect representation of the reality in a simulator is impossible to realize, but this is not often necessary to obtain reliable results (Alm, et al. 2006; Nilsson, 1993; Nählinder 2002). Depending on the research question, even a low fidelity, PC-based table
Simulator can be able to provide reliable and valid results; in other cases even a full-scale simulator with an advanced motion system will not necessarily produce valid test results.

Tests of new design solutions are not always best evaluated in a full-scale simulator using experienced operators as test persons. These operators comprehensively understand existing technology and can often find new design solutions worse, even though the new design is objectively superior. Therefore, new designs are preferably tested with operators with limited experience.

4.8 Simulator sickness

An important simulator limitation that has been seen in flight and car simulators is simulator sickness. The feeling of sickness in connection with extensive use of simulators has been reported in the literature. Casali (1996) has found literature that deals with simulator sickness in a helicopter simulator as early as 1957 (Havron & Butler). In the beginning, these phenomena were treated as motion sickness or the result of whole body vibrations at very low frequencies. Both motion sickness and simulator sickness can produce a number of symptoms, such as overstrain of the eyes, a feeling of sickness, sleepiness, pallor, headache, disorientation, sleepiness, and an inability to work. Vomiting, however, is rare.

Simulator sickness is not the same as motion sickness (e.g., sea sickness), even if in many contexts it is described that way (Nilsson, 1993). Motions are fundamental for motion sickness, whereas simulator sickness can arise even if there are no motions in the simulator platform. Simulator sickness tends to be less vigorous than motion sickness and occur less frequently. It is necessary, therefore, to make a distinction between simulator sickness and motion sickness. According to LaViola (2000), the human being can experience imperative self-motions even if the operator remains still relative to the environment. This effect is called vection, which can occur in natural environments such as when you look through a car window and experience motion even if the car itself is not moving. It has been shown that the visual projection (the process), such as experienced vection, can also contribute to simulator sickness (Kennedy et al., 1988). Similar effects have been seen in virtual environments, especially those with a wide field of view or helmet-mounted displays with few references to the static world. Simulator sickness has many complex contributing factors that in turn are manifested by several different symptoms. A description of many factors that influence simulator sickness can be found in Kolasinski (1995). Kennedy and Folwkes (1992) claim that simulator sickness arises from several factors and that no specific factor can be identified as a generic factor without dependence on other factors.

The consequences of simulator sickness on studies can be serious if not being taken into account (Casali, 1986). The simulator sickness influences the studies in many different ways, primarily in a negative way on operator performance. Other problems include incorrect use of the simulator, and discrepancy from real experiences.

Optic flow is created by the movement of objects when an individual moves in its environment (Goldstein, 1989). An example of optic flow is when a driver fixes his/her
eyes on a direction. All objects in the field of view tend to move from the centre to the periphery (see Figure 4.11). The point from where optic flow begins is called the “point of expansion” (POE). Simulator sickness normally occurs in the beginning of simulator use (when optic flow is large), and can continue up to six hours following use of the simulator (Goodley et al., 1999), although duration of this length is unusual. Symptoms normally disappear almost immediately when one’s eyes are shut. The opposite occurs with motion sickness—when closing one’s eyes, sickness ensue.

![Figure 4.11: An example of optic flow. Mollenhauer, 2004.](image)

**4.8.1 The influence of simulator parameters on simulator sickness**

Kolasinski et al. (1995) and Mollenhauer (2004) have described a number of factors regarding the simulator hardware that can cause simulator sickness, such as the field view of the simulator (operator’s view), the flicker, resolution and frame rate of the display, the frame rate of the computer, the environment of the simulator, motions of the platform, visible references and visual background.

**Lag**

Lag in the simulator is the time between reading an input from the operator, processing the input and then presenting the changes on the simulator display. If there is a lag in the system, it is considered to add divergence in the signals between the operator’s sense of balance and vision. This divergence can lead to simulator sickness. Attempts to reduce the effect of sluggishness/lag have been focused partly on the visual system and partly on the motion platform.

**Field of view**

A simulator with a wider field of view is considered to increase simulator sickness (Kennedy et al., 1988; Casali, 1996; Kolasinski et al., 1995; Pausch et al., 1992), since a wider field of view increases the potential for more peripheral vision. This increases the risk of simulator sickness for two reasons. First, a wider field of view increases the feeling of apparent movement, or vection. An increased feeling of apparent movement increases the signal conflict between vision and sense of balance. Secondly, peripheral...
vision has a higher threshold for flicker, meaning that flicker can be perceived at a higher frame rate than the display.

**Flicker of the display**

The experience of flicker on the display is influenced by a number of factors, such as the computer’s frame rate, field of view and brightness. Flicker and increased brightness increase the possibility of simulator sickness. The critical frequency for noticing flicker is 40-60 Hz; the exact limit is difficult to identify since it depends on brightness and light in the surrounding environment. In general, the darker the room the lower the frequency is before flicker occurs.

**Resolution of the display**

The resolution of the display may influence the task performed and can also cause simulator sickness. A healthy eye is able to distinguish a feature that corresponds to about one arc minute on the macula part at the retina. The view angle corresponding to a specific feature is dependent on the number of pixels on the display and the distance from the operator to the display. The result of limited resolution may lead to the operator missing important aspects of his work, and could create overstraining of the eye since it is not possible to focus. No connection between the resolution of the display and simulator sickness has been found in the literature.

**Frame rate of the display**

The frame rate is the frequency at which the computer is able to make new calculations based on the operator’s use of functions and then show it on the display. The frame rate is strongly dependent on computer capacity, software and the graphical capacity of the computer. Too low a frame rate creates behavior that is more sluggish than the real movements of the machine. Based on experience, a system with a sample time of 0.1 seconds (10 Hz) is perceived as slow. The response time for the eye/brain to process a new picture is about 20 milliseconds (50 Hz).

**Frame rate of the computer**

If the computer has a frame rate of 30 Hz and the projector/display has a frame rate of 60 Hz, each picture is presented twice. If the frame rate of the computer varies there is a potential risk of simulator sickness (Kolasinski et al., 1995). Since frame rate is normally at 60 Hz or more (and also very stable), the risk of simulator sickness related to the frame rate of the computer is minimal.

**Motion platform**

By using a motion platform, it is possible to move and imitate real motion and thereby obtain increased realism, increased validity of operator input and less simulator sickness. In the literature there are numerous examples of experiments on how to improve the validity of the simulator by introducing motion platforms. However, this is not guaranteed to reduce or eliminate simulator sickness. There are examples where the motion platform has reduced simulator sickness, as well as examples where it has increased it (Casali, 1986). There are many factors that determine if the outcome is going to be positive or not; in general the opinion is that an accurate, connected motion platform reduces simulator sickness. In the best-known study (Sharkey & McCauley, 1992), a platform with and without motion was compared. The test drivers performed
the same task with and without motion, and the platform with motion did not eliminate simulator sickness. The same result was achieved by Watson (1995). Moreover, a motion platform does not guarantee more valid experiments (Alm et al., 2006; Nählinger, 2002, 2006).

The simulator environment

A number of physiological reactions arise as a consequence of simulator sickness, such as changes in heart frequency, blood pressure, breathing and skin temperature. Many of these reactions can be stress related. Increased temperature, which is experienced as uncomfortable, can contribute effects related to body temperature, heart frequency, etc. Increased blood pressure can be connected to increased stress levels, but also to a lack of water, which can occur during long periods in the simulator. As a general rule, it is recommended to have good ventilation and temperature control.

Adaptation to the simulator

It is generally accepted that simulator sickness increases along with the time that a person spends in the simulator on a single occasion, but that this risk decreases as the number of occasion increases. Watson (1997) showed that simulator sickness decreased by two-thirds between the first and third simulations. It is important that the operators participating in the tests have the opportunity to practice and adapt to the simulator; at the same time, it is important for those evaluating the results to understand how exposure and adaptation influence the results.

4.8.2 The simulator at Skogforsk

The simulator that was used in all tests was developed by Oryx Simulations AB. It consists of equipment from a real Valmet forwarder or harvester such as a seat, pedals, joysticks, bucking computer and machine computers controlled via a CAN-bus, and three displays. Since the simulator acts on responses to the input from an operator and then makes calculations based on physical data of a real forest machine, the properties and responses are very similar to a real forest machine. This has been evaluated over a period of many years by professional operators. The simulator simultaneously provides a good dynamic resemblance as well as a very good graphical presentation.

Field of view

To suit several forest machine applications such as a harvester, a harwarder and a forwarder, the field of view in the simulator is about 200 degrees horizontal, distributed over three displays. Such a big field of view increases the conflicts between vision and sense of balance. A field of view of 110 degrees is normally used due to the fact that almost all work takes place in front of the operator. The only time the wider field of view is used in a forwarder application is when the operator is looking for new piles or logs on the ground that need to be picked up. In this case, the operator is turning his/her head about 90 degrees to the left or right of the piles and logs, meaning that the risk for conflict between vision and sense of balance is small.
Flicker of the displays
The displays/projector has a refresh rate of 60 Hz (16.7 ms). The critical frequency is between 40 and 60 Hz; since the simulator is in the upper frequency limit and the simulator room is painted black, the risk of display flicker is small.

Resolution of displays
In this case the simulator has three displays 2.0 x 1.5 metres in size, with a projector for each screen and each display showing a two-dimensional presentation. The resolution of each display is 1,400 x 1,050 pixels, corresponding to an effective resolution of 3.3 arc minutes per pixel. Many simulators have a resolution of 3 to 5 arc minutes per pixel.

Frame rate of the displays
The display frame rates depend very much on the computers. In this simulator there is one computer for each display, and the frame rate of each computer is about 40 Hz (25 ms). The access time for the eye to process a new picture is about 20 ms (50 Hz). This presents a small risk of conflict between sense of balance and vision.

Frame rate of the computer
The simulator consists of four computers, one dedicated for calculation and three for the displays (one computer for each display). The calculation computer has a frame rate of 80 Hz and the display computers have a frame rate of 40 Hz. As such, two calculations are made for each picture.

Motion platform
The simulator has a fixed platform. While a motion platform would probably provide a more realistic feeling, the machine is normally standing still when the operator is working with the boom. For a harvester, boom operation time is about 90% of the total working time, and on a forwarder boom operation time is about 50% of total machine operation time.

Calibration
The simulator has been evaluated by professional operators several times over many years, indicating that the forest machines in the simulator act much like the real machines. Each operator also has the opportunity to adjust the response so he/she feels that the machine acts closely to the behaviour of his/her own real machine.

Scenarios in the simulator
The forwarder that has been analyzed is a Valmet 860.4 with the knuckle boom Cranab CRF 11 (see Figure 4.12). We also used the harvester Valmet 911.4.
Conclusion

The literature describes several factors that can influence the operator in a forest machine simulator. While it is impossible to reduce all factors that can cause simulator sickness, some can be reduced. When it comes to technical factors, the Oryx simulator meets all the requirements of a research tool. The only problem is the large field of view, which can cause simulator sickness. On the other hand it is not possible to totally avoid simulator sickness since everyone is, in different ways, predisposed to some simulator sickness.

During the fidelity test (Paper B) and the evaluation of boom tip control (Paper C), none of the operators or students experienced any kind of simulator sickness.
Chapter 5
Outline of Appended Papers

This chapter provides an overview of the contents of the four papers. A brief discussion on the main result of each paper completes each presentation.

5.1 Summary of Paper A

The objective in Paper A was to develop a kinematic control strategy for the maximum lifting capacity algorithm, which is suited for computer-controlled redundant knuckle booms. The algorithm was analyzed with respect to time consumption when the TCP moves along a pre-determined path. The analysis shows the necessary joint speed requirements and the time consumptions for certain motion cycles, including the case when the joints reach their maximum velocity limits.

Main results

A set of equations for use in simplified boom tip control (BTC) of redundant knuckle booms, to be implemented in a forest machine simulator.

5.2 Summary of Paper B

Normally, existing forest machine simulators are used to train future forest machine operators. In this paper, the simulator was used and evaluated as a research tool. To investigate the usefulness of results from simulator work, a fidelity test was performed. A time study was conducted, during which a harvester operator cut down approximately 500 trees. Data from the stand, including tree diameter, height, position, height to first live branch and tree type were registered, as was the terrain. The same stand and terrain data was then fed into the simulator and the same operator performed the same work as before, but in the simulator.

Main results

The results demonstrated that there is good fidelity between a real forest machine and the simulator. The difference in time consumption between the reality and the simulator was about ± 5%, with a significance of > 95% for different work operations. Qualitatively, the results were on par.

5.3 Summary of Paper C

Paper A described several advantages with BTC. To evaluate the possibilities with BTC, the equations from Paper A were fed into the dynamic forest machine simulator. Time studies were carried out with inexperienced operators, using both conventional control and BTC. The participants, from a vocational school for natural resources, were volunteers. The study shows that BTC is an easier control system to learn compared to conventional control. This can lead to savings in production due to shorter learning times and operators reaching full production sooner. BTC will also lead to lower levels
of errors, as well as fewer serious errors. BTC does not provide more efficient control with respect to time, but the improvement comes faster. BTC requires less mental load than conventional control, which in the long run will reduce the mental load on the operator of forest machines such as a forwarder.

**Main results**

The use of BTC in the dynamic forest machine simulator shows several benefits. The results from the studies show that the assumed advantages described in Paper A are fulfilled except that for the lifespan of the boom the study did not include an evaluation.

### 5.4 Summary of Paper D

In this paper the maximum lifting capacity algorithm was further extended to ensure TCP path-tracking possibilities. The path-tracking takes into account both joint velocity and acceleration limits. The proposed algorithm allows straightforward implementation on any forest knuckle boom, and we conclude that the path tracking controller works properly given the simulator model of the crane. When we compare simulations with and without load in the TCP, remarkable crane stiffness is noticed. Since the simulator models are proprietary it has not yet been possible to analyze the reasons behind this remarkable stiffness which intuitively seems too high.

**Main results**

The main results are the kinematic algorithms and equations for implementing the maximum lifting capacity with path-tracking capabilities, and an evaluation of these kinematic solutions in the dynamic forest machine simulator.
Chapter 6
Comparison between reality and the simulator

In Papers A to D, simplified boom control and path planning were introduced and a
dynamic forestry machine simulator was used to verify the algorithms. The final stage of
this thesis work has been comparing the simulator with a real forest machine. To be able
to do so, the hydraulic cylinder pressures, joint angles and positions were measured on
the forest machine and then compared with results from the simulator.

6.1 Tests in the reality

The tested knuckle boom, CRF 11, was mounted on a Valmet 860.4 forwarder. An
operator was used, since it is not possible to control the boom on the forwarder
automatically with a computer. The boom was controlled by an experienced forest
machine instructor who attempted to follow the same paths as described in Paper D —
normal paths taken during the loading of logs into the load space of the forwarder. The
tests were performed both with and without the load, which had a weight of 920 kg. The
operator attempted to follow the same paths for every test, as well as for both the loaded
and unloaded boom.

The sequence was the following:

- The operator started with the grapple on the frame, between the bunks, in the
  load space (see Figure 6.1).
- The operator then moved the grapple between the bunks (see Figure 6.1).
- Then the operator placed the grapple on a spot about 7 meters from the boom
  pillar (see Figure 6.2).
- Then the operator moved the grapple back over the bunks (see Figure 6.2).
- Finally, the operator placed the grapple back on the frame between the bunks
  (see Figure 6.3).
Figure 6.1: Left: The operator started with the grapple between the bunks. Right: The operator moved the grapple between the bunks.

Figure 6.2: Left: The operator moved the grapple to a certain distance from the pillar. Right: The operator moved the grapple back over the bunks.

Figure 6.3: The operator moved the grapple back down on the frame between the bunks.

6.1.1 The measuring system
In this thesis, the tests were carried out on three different real forest machines. In Part 2 (Paper B), the Valmet 921 harvester was used to verify the fidelity of the simulator, in a
test carried out in Uppland, Sweden. In Part 4 (Paper D), a forwarder Valmet 830 was used to identify how the operator worked, with the tests carried out in a forest outside Umeå. In Part 5, a Valmet 860.4 forwarder was used to measure, with the help from Umeå University, the boom motions (revolute joint angles and prismatic joint position) and the corresponding hydraulic cylinder pressures needed to move the boom tip. This chapter describes these latter measurements (Valmet 860.4) in greater detail and compares the results to those of the dynamic forest machine simulator. This is done to further understand the validity of the results from tests using the simulator. The following measuring devices and sensors were used.

The processing unit applied in all measurement cases was the dSPACE MicroAutoBox (MABX), which directly controls the available I/O features, such as the Electronic Control Unit, the AD and DA converter units, the digital I/O and CAN subsystems. MATLAB/Simulink was used to implement executable code for the processing unit. There was also an additional Signal Conditioning Unit (SCU) to handle incoming measurement signals and adapt them to the voltage levels needed for the MABX. Finally, a Dell PC was used to monitor measurements and serve as an online user interface through the use of the Control Desk.

Sensors

The following sensors were used:

- Encoders with 4000 pulses/turn were used to measure angular positions and prismatic joint position. See the positioning of the sensors in Figure 6.4.

- Pressure transducers capable of sensing in a range of 0 to 600 bar. See the positioning of the sensors in Figures 6.5, 6.6 and 6.7.

![Figure 6.4: Drawing of the boom showing placement of pressure and position sensors.](image)
COMPARISON BETWEEN REALITY AND THE SIMULATOR

Figure 6.5: Picture of angle and pressure sensors mounted on the boom, left at $\theta_0$, right at $\theta_1$. A green arrow points at a position sensor and a red arrow at a pressure sensor.

Figure 6.6: Picture of angle and pressure sensors mounted on the boom, left at $\theta_2$, right shows the connection between the sensor and moving part. A green arrow points at a position sensor.

Figure 6.7: Left: picture of measuring device. Right: picture of the length sensor $d_3$ mounted on the boom. The turquoise arrow points at the data collecting system and the green arrow at a length position sensor.
Data collection
The sampling rate, when collecting data, was 1 000 Hz.

Calibration
The angle sensors were calibrated before and after the tests by putting the TCP of the boom in a specific position. The pressure sensors were calibrated before and after the tests with a special hydraulic calibration unit.

6.2 Tests in the simulator
The same paths as used in the field were tested by using the recorded data from the field test. The recorded data from the joint positions of the boom were saved in a file, and were used by the simulator as input to the simulator software. For the path-following controller in paper D and for this test case, the simulator provider extended the simulator with a PID controller to replace the operator of the simulator. The implemented PID controller takes the measured joint angles/positions as motion references (velocities) and based on those, the simulator derives the torques/force acting on the boom joints.

In the field test, the pressure on both sides of the cylinders was measured and then used for calculating the forces acting on the real boom. The main method for evaluating the simulator’s capability of mimicking the real machine was to compare the forces in the machine with the forces in the simulator for the very same motion profiles. To be able to compare the forces between the real machine and the simulator, a transformation of the relationship between the cylinder extensions and joints \( \theta_1 \) and \( \theta_2 \) was needed, since the simulator software uses a hinge joint (revolute joint) for \( \theta_1 \) and \( \theta_2 \). For the \( d_1 \) function, the simulator uses a prismatic (translational) joint and therefore a transformation is not needed since this function is similar to the real boom.

6.2.1 Transformation of \( \theta_1 \)
A transformation of the hinge joint motion \( \theta_1 \) to the corresponding cylinder motion is derived below. The notation is given by Figure 6.8.
COMPARISON BETWEEN REALITY AND THE SIMULATOR

Figure 6.8: Left: drawing of the joint $\theta_1$ acting between the pillar and the first boom. Right: a simplified sketch.

The cosines theorem gives:

$$L_1^2 = L_2^2 + L_3^2 - 2L_2L_3 \cos \beta$$  \hspace{1cm} (6.1)

where

$$\beta = \theta_1 + 90 - \alpha$$  \hspace{1cm} (6.2)

$$\alpha = \arctan \frac{L_4}{L_3}$$  \hspace{1cm} (6.3)

$$L_5 = \sqrt{L_3^2 + L_4^2}$$  \hspace{1cm} (6.4)

and

$$L_1 = L_c + L_r$$  \hspace{1cm} (6.5)

where $L_c$ is the length of the hydraulic cylinder house and $L_r$ is the actual length of the piston rod of the cylinder.

$$\theta_1 = \arccos \left( \frac{L_2^2 + L_3^2 - L_1^2}{2L_2L_3} \right) - 90 + \arctan \frac{L_4}{L_3}$$  \hspace{1cm} (6.6)

6.2.2 Transformation of $\theta_2$

A transfer of the hinge joint motion $\theta_2$ to the corresponding cylinder motion is derived below. The notation is given by Figures 6.9 and 6.10.
The angle $\lambda$ is given by:

$$d^2 = F^2 + G^2 = q_1^2 - 2q_1b \cos \lambda + b^2 \cos^2 \lambda + q_2^2 + 2q_2b \sin \lambda + b^2 \sin^2 \lambda$$  \hspace{1cm} (6.9)$$

which can be written:

$$A \cos \lambda + B \sin \lambda = C$$  \hspace{1cm} (6.10)$$

where
COMPARISON BETWEEN REALITY AND THE SIMULATOR

\[ A = -2q_1 b \] (6.11)

\[ B = 2q_2 b \] (6.12)

\[ C = d^2 - q_1^2 - q_2^2 - b^2 \] (6.13)

Tedious calculations give

\[ \cos \lambda = \frac{AC \pm B\sqrt{A^2 + B^2 - C^2}}{A^2 + B^2} \] (6.14)

\[ \sin \lambda = \frac{BC \pm A\sqrt{A^2 + B^2 - C^2}}{A^2 + B^2} \] (6.15)

Introduce the coordinates \((H, I)\) for the upper end of \(L\):

\[ H = q_1 - (b + c \cos \mu) \cos \lambda - c \sin \mu \sin \lambda \] (6.16)

\[ I = q_2 + (b + c \cos \mu) \sin \lambda - c \sin \mu \cos \lambda \] (6.17)

giving

\[ L = \sqrt{(H + g)^2 + (I - e)^2} \] (6.18)

The transformation equations above were implemented and used to compare the forces/torques acting on the joints.

6.3 Test results

The test results presented are without load in the grapple. The differences between TCP position, joint speeds and forces are presented and discussed below.
Position of the TCP

![Graph of position of the TCP in polar coordinates, boom out motion.](image)

Figure 6.11: The position of the boom tip in polar coordinates, *boom out* motion.

Joint forces and speeds

![Graph of cylinder force of the first section of the boom, boom out motion.](image)

Figure 6.12: The cylinder force of the first section of the boom, *boom out* motion.

![Graph of joint speed of $\theta_1$, boom out motion.](image)

Figure 6.13: The joint speed of $\theta_1$, *boom out* motion.
Figure 6.14: The position of the boom tip in polar coordinates, \textit{boom in motion}.

Figure 6.15: The cylinder force of the first section of the boom, \textit{boom in motion}.

Figure 6.16: The joint speed of $\theta_1$, \textit{boom in motion}.
Due to very small differences, less than a couple of millimeters, it is impossible to see a difference in the TCP position, therefore only the curves in real experiments are shown, (Figures 6.11 and 6.14).

While there is a difference between the measured force and the simulated force, the general behavior is similar in both cases. The differences at the beginning and end of the test are explained as follows. In the real case the grapple is in the beginning resting on the load space of the forwarder and at the end of the motion the grapple is resting (at least partly) on ground and hence the measured pressures indicate a low force. The resting on a supporting surface is not captured by the simulator. The reason for this discrepancy is that the defined motion paths are hard to synchronize exactly with the geometry of the ground and forwarder chassis models in the simulator. It is hard to see a correspondence between the motion profiles and the force profiles, this is explained by that the gravitational forces on the boom is dominating largely over the dynamic forces to set the boom in motion. The noise in the simulated force curve is due to noise in the differentiated joint angle measurement.

Due to very small differences it is hard to see the difference in joint speed for $\theta_1$ between the real machine and the simulator.

Below the simulated and measured velocities for the other two DOF’s ($\theta_2$ and $d_z$) are shown.

![Figure 6.17: The joint speed of $\theta_2$, boom out motion.](image)
COMPARISON BETWEEN REALITY AND THE SIMULATOR

Figure 6.18: The joint speed of $\theta_2$, boom in motion.

Figure 6.19: The joint speed of $d_3$, boom out motion.

Figure 6.20: The joint speed of $d_3$, boom in motion.
Due to very small differences it is hard to see a difference (Figure 6.17 to 6.20) in joint speed for $\theta_2$ and $d_3$ between the real machine and the simulator. It was impossible to verify the forces acting on the joints $\theta_2$ and $d_3$ due to still unknown problem in the simulator or in the measurements.

6.4 Conclusions

There are almost no deviations when, the TCP, in the simulator follows the same references which the operator performed on the forest machine. This also holds for the joint speeds of the boom. When comparing the forces of the cylinders in each joint there was a very good relation, between simulator and the forest machine, for the cylinder of first section of the boom. The problem mentioned above with verifying the forces is a subject of further studies.
Chapter 7
Discussion and conclusion

One of the aims of the thesis has been to develop an algorithm for simplified boom control, boom tip control, as well as automatic functions of knuckle booms for forest machines. The maximum lifting capacity algorithm, which is based on inverse kinematics of kinematically redundant knuckle booms, has been introduced. Simulations have been used throughout the thesis work to verify the suggested algorithms.

First, the maximum lifting capacity was evaluated in a kinematic simulation software. The results (Paper A) show the necessary speed requirements for all joints when performing straight paths in the manipulator work area. The simulations also show time consumptions. These simulations demonstrate that the algorithm worked well and was suitable for implementation in a forest machine simulator.

Before the implementation of boom tip control, a study was performed to evaluate the fidelity of the dynamic forest machine simulator. This was done to verify if the simulator was good enough to be used for studies of boom tip control and semi-automatic functions of the boom. The results from the study (Paper B) demonstrated that there is good fidelity between the forest machine simulator and a real forest machine, and that the results that would be generated by different forthcoming studies on the simulator would be reliable.

Boom tip control was implemented in the simulator and then an evaluation of the advantages was performed (Paper C). The results show that boom tip control is an easier system to learn compared to conventional control. This can lead to savings in production due to shorter learning times and the operators being able to reach full production sooner. Boom tip control does not provide more efficient control with respect to time, but the actual improvement comes faster. Boom tip control creates less mental strain than conventional control, which in the long run will reduce mental stress on the operators of forest machines such as a forwarder.

The maximum lifting capacity algorithm was then developed to ensure TCP path-tracking of a forest knuckle boom, while taking into account both velocity and acceleration joint limits (Paper D). The path-tracking function was then implemented into the simulator. When comparing simulations performed in the kinematic simulation software with simulations performed in the dynamic forest machine simulator, the results show that they are very similar and have only very small deviations, independent of load.

The final investigations performed were done to compare the simulator results with results from a real forest machine. This study did not involve the boom tip control but was instead based on boom motion profiles typical for real forwarder operations. A PID controller was implemented in the simulator to facilitate that the real motion profiles was followed more or less exactly during simulation. The results show that the simulator is capable of reproducing the real motion profiles with good accuracy. For the inner boom joint the investigations show good agreement in force profiles between the real machine
and the simulator. However, for the two outer joints it was not possible to draw the same conclusion. This might be due to measurement problems or a combination of measurement problems and effects not captured by the simulator. Unfortunately, time and resource limitations did not allow deeper analysis and renewed measurements for completely concluding the evaluation of the simulator. Still, the results from paper B, paper C and the investigation in chapter 6 makes us confident that the simulator is good enough for the research work done during this thesis work.

This was the first time that this dynamic forest machine simulator was used for investigation of automation of a forest machine knuckle boom. It was also the first time a comparison with real measured data was made between the simulator and a real forest machine. Although these types of boom control have never been performed before, the conformity between reality and the simulator was very good. The simulator also proved to be a useful research tool for studies performed in the context of this thesis work.

The thesis had two overall long term objectives. The first was to provide the industry and forestry sector with usable and verified ideas and results in the area of automation. This was accomplished in simulations and the results are being implemented in a joint venture between a hydraulic manufacturer, a forest machine manufacturer and a forest enterprise. The second objective was to strengthen the research and development links between the forestry sector and technical university research. This was accomplished through a number of courses, projects and Masters Theses over the last three years. About 150 students in total have been studying forest machine technology.

7.1 Future work

Recommendations for future work can be summarized as follows:

More simulator studies. In this thesis work, only a few paths have been analyzed. Studies need to be carried out with several operators to identify how other paths can be of interest for automation.

Include obstacles. Depending on where the grapple is placed in the load space, obstacle avoidance — avoiding the bunks and the frame of the load space in particular — needs most of all to be included.

Implementation on forest machines. The results from this thesis have reached a level where the natural next step is to implement the findings on a real forwarder. This is a step that is initiated already.

Evaluation. To be certain of the positive results that have been reached in this thesis, the results need to be verified with several machines and operators. Here, normal time studies of forwarders and harvesters need to be conducted.

Human-machine interaction (HMI). How should the simplified control and automation work in relation to the operator? More studies need to be carried out to analyze how the best HMI in a forest machine should be designed. This is best developed and analyzed using the DFMS.
Further control development. The thesis has only presented kinematic solutions. Control development based on a more complete model of the dynamics of the machine including the hydraulics is a suitable topic for further research.
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Paper A

KINEMATIC CONTROL OF REDUNDANT KNUCKLE BOOMS

ABSTRACT

The Swedish forestry industry competes in the global marketplace, and since its raw material is more expensive than in other parts of the world, the supply chain from stump to industry must be very efficient. One part of this supply chain is cutting down and transporting trees out of the forest to the landing area for further transportation via truck to the paper mill or sawmill. Forestry machines equipped with booms are used to cut down, handle and transport the trees. If we can reduce boom handling time and thereby increase productivity by 10%, the Swedish forestry industry could conceivably increase their earnings by up to SEK250 million annually.

One way of decreasing boom handling time is to introduce automatization. This paper describes how to solve the kinematic control of the knuckle booms used on forestry machines if automatization is introduced. The objective is to develop a kinematic control strategy for maximum lifting capacity, which is suited for computer-controlled knuckle booms that are redundant. The strategy is analyzed with respect to time consumption when the manipulator tip moves along a pre-determined path. The analysis is made on a knuckle boom used on a forwarder in a forestry application. The knuckle boom has one redundant degree of freedom. The analysis shows the necessary joint speed requirements and the time consumptions for certain motion cycles, as well as maintained path following despite actuators reaching their velocity saturation level.

KEYWORDS

Hydraulic manipulator, redundant, kinematic control, local optimization, knuckle boom, forest machine, forwarder, boom tip control, joystick control, simulations.

INTRODUCTION

Powerful and radical mechanization trends underway in the Swedish forest industry since the middle of the 1960s have rendered the industry almost 100% mechanized, and counts as one of the main reasons why Swedish forestry companies have remained competitive internationally.

The forestry machines of today are high-tech units with advanced control engineering. Technological developments have resulted in a radical increase in performance. For the machine operator, this has meant increased workload and fewer natural stops during ordinary work. At the same time, quality control has been playing an increasing role in the operators’ work. It is no longer sufficient to only control the machine and its functions; the operator also has responsibility for considering environmental concerns, and for the planning and follow-up of his/her work. While the working environment has improved considerably over the years vis-à-vis the physical stress placed on the
operator, the increased workload, in combination with the many decisions that operators must now make, has increased mental stress. As such, the operator could become a bottleneck in the context of new initiatives aimed at increasing productivity. Introducing more automated functions and letting the machine do the more repetitive parts of the work, would allow the operator to devote his/her time to making decisions about tree selection, wood quality and environmental concerns.

The control of the manipulator’s movement consumes most of the operators’ working time on both the forwarder and harvester machines. There is potential for simplifying the manipulator control in order to reduce the mental workload, allow greater focus on more important tasks, and increase production. Moreover, the risk of repetitive stress injuries can also be reduced. The manipulator control can be simplified through the introduction of manipulator tip control and automated control of certain manipulator movements.

Controlling a forestry machine involves almost continuous precision work with the hands; a high rate of production thus requires intensive precision work. This kind of repetitive work, coupled with high intensity, will cause tense muscles and muscle fibers. In a harvester application, the operator uses the crane about 80% of the total work time. He/she cuts one tree every 47 seconds, and makes 12 decisions per tree; this results in an average of 24 functions per tree and about 1000 trees cut per day. In a forwarder application, the operator uses the crane about 50% of the total work time. He/she loads and unloads on average of a grapple with logs every 30 seconds and handles about 2000 trees per day.

Actions that increase the amount of blood running through the muscles are of crucial importance for minimizing the risk of repetitive stress injuries. Controlling the manipulator should, therefore, be as dynamic as possible, which calls for many short pauses. Decreased demands on the precision of the joystick work will also generate less muscle tension, thereby reducing stress. As such, technical solutions that allow for short pauses during intensive periods of joystick work would be a positive development [7, 8].

Operators of forestry machines are exposed to great mental stress during their work. They must receive and process a large amount of information and make decisions under considerable time pressures. At the same time, the precision work with the hands also creates a relatively large amount of stress on brain activity [8].

A natural conclusion of this is that a simplified manipulator control could reduce mental stress on the operator.

The forestry machines of today are controlled by two joysticks of different design. With the movement of one joystick in one direction, the operator controls a specific hydraulic actuator (cylinder) on the boom. This means that the operator has to combine different joystick movements to move the tip of the boom in the desired direction (see Figure 1).
Figure 1: Conventional control.

The concept *boom tip control* means that the tip of the boom is controlled with only one joystick. Up/down on the joystick corresponds to up/down on the tip of the boom, out/in corresponds to out/in on the tip of the boom, and left/right corresponds to left/right on the tip of the boom (see Figure 2).

Figure 2: Boom tip control.

With the extension, and the linear degree of freedom in the outer boom in combination with the two rotational degrees of freedom, it is possible for the TCP (Tool Center Point) to reach the next point on a desired path in a number of ways. In [16], computer analysis shows how the extension degree of freedom affects speed and lifting force. The results show that the extension gives shorter time cycles and that one has better lifting force capacity closer to the ground. This means that, to the extent possible, the extension should be included in the kinematic control algorithm for the boom.

**Joystick control**

As mentioned above, the TCP is controlled with only one joystick. To have a solid analogy between the joystick and the manipulator, the joystick functions are placed in the following order (see Figure 3).
Figure 3: Joystick functions with boom tip control.

The swing function is placed in the left and right directions of the joystick.

**Potential advantages of boom tip control**

A simplified manipulator control would provide the following advantages compared to the conventional manipulator control.

- Less physical strain on the operator;
- Decreased learning time; and
- Longer lifespan of the manipulator.

After several years controlling the manipulator, forestry machine operators often have neck, shoulder and back injuries [7]. Indeed, more than half of all operators have had problems in these parts of the body. These problems can be attributed to the manipulator control.

It has been shown that operators unconsciously tense themselves before manipulator control work, resulting in increased stress [7]. A simplified manipulator control would most likely influence the operators to control the manipulator in a more relaxed way.

Work with conventional control on knuckle booms is very complicated since one can reach every point within the knuckle boom’s workspace in many different ways (because the knuckle booms are redundant). A simplified control would most likely make it easier to control the knuckle manipulator since the tip would be controlled directly and since the redundancy would be resolved automatically.

Consequently, a simplified control would also most likely mean that learning time would decrease substantially [26].

There is a significant difference between a skilled operator and a non-skilled operator in terms of controlling the manipulator [25]. A non-skilled operator will control the manipulator in a jerky manner, which affects the lifespan of the manipulator itself. A
simplified control, then, would eliminate many jerky movements and thereby increase the manipulator’s lifespan.

Targeted machines and research objectives

Most of the manipulators used in forestry are of the knuckle boom variety. They are hydraulically powered and controlled by hydraulic servo valves. The same types of hydraulic manipulators are also commonly used on trucks and in stationary applications. Knuckle booms are often constructed with extra degrees of freedom (DOFs) and are thereby redundant. A redundant manipulator, designed for positioning only, has more than 3 DOF. In this paper, a redundant manipulator with three revolute joints and one prismatic joint is studied.

Why is a redundant DOF introduced? The extra cylinder, sensor and other mechanics, as well as the more complicated controller translate into extra weight, complexity and cost. Some objectives of the redundant DOF are:

- singularity avoidance;
- obstacle avoidance;
- robot dexterity;
- energy minimization;
- manipulator precision;
- lifting capacity; and
- velocity of operation.

This study was initiated by the introduction of computer control in a manually operated 4 DOF manipulator with a configuration as shown in Figures 1 and 2.

The objective is to develop a kinematic control strategy to achieve the maximum lifting capacity of a redundant knuckle boom. The strategy is analyzed with respect to time consumption when the manipulator tip moves along a pre-determined path. The analysis is made on a knuckle boom used on a forwarder in a forestry application. The same type of manipulator geometry is used on other forestry machines as well as on trucks.

STATE OF THE ART

After a review of the research conducted up to this date one can see that there has been increased interest in redundant manipulators since the beginning of the 1980s, and that there remains great academic interest in problems regarding kinematic redundancy.

In industry, it was necessary to introduce one or more redundant degrees of freedom to solve complicated applications. The redundancy, however, involves considerably more complicated control algorithms than in non-redundant manipulators.
Most of the methods are based on local optimization, and use the popular pseudo inverse solution. The kinematic equation that describes the relationship between manipulator end effectors speed and corresponding joint speeds is defined as follows:

\[ \dot{x} = J \dot{\theta} \]  

(1)

where \( \dot{x} \) is an \( m \times 1 \) velocity vector in Cartesian coordinates for the manipulator end effectors, \( \dot{\theta} \) is the \( n \times 1 \) joint velocity vector for the joints, \( n > m \) and \( J \) is the \( [m \times n] \) Jacobian.

We solve the equation with respect to \( \dot{\theta} \)

\[ \dot{\theta} = J^* \dot{x} \]  

(2)

where

\[ J^* = J^T (JJ^T)^{-1} \]  

(3)

\( J^* \) is the pseudo inverse of the Jacobian matrix according to the generalized Moore-Penrose inverse [22].

In [12] the problem with joint drift, when one uses only the pseudo inverse control, is analyzed when a cyclical task is performed. Despite the well-developed theory in [13], making the control conservative remains problematic—when repeating a work cycle several times, the joint configurations will not be repeatable and the manipulator can run into unfavorable configurations.

To overcome this drawback, a more general solution, the addition of a term, is given by:

\[ \dot{\theta} = J^* \dot{x} + (I - J^* J) \dot{\phi} \]  

(4)

where \( \dot{\phi} \) is an arbitrary joint velocity vector and \( (I - J^* J) \) is the null-space projection matrix of \( J \). This corresponds to a self-motion of the manipulator that has no effect on the velocity of the end effectors. The attractiveness of this is twofold. The first term, \( J^* \dot{x} \), minimizes \( \dot{\theta}^T \dot{\theta} \), which presumably means that all the joints will be prevented from moving too fast. The second term, \( (I - J^* J) \dot{\phi} \), can improve the manipulator’s configuration by assigning different optimization or performance criteria by means of a proper selection of \( \dot{\phi} \), to achieve, for example, singularity avoidance [14].

Other secondary criteria are: obstacle avoidance [11, 23], joint torque optimization [9, 19, 20], joint velocity constraints [1, 5, 15, 17], energy minimization [1, 2, 10], manipulator precision [10], speed of operation [2], joint limit avoidance [3], maximization of various end effector dexterity measures [11], multiple performance criteria [4, 6, 18], global optimization and global versus local optimization [26].
Those seeking more information about redundant manipulators can find a review in [21] and a tutorial in [24].

**BASIC EQUATIONS**

The studied manipulator is velocity-controlled by means of a 3-DOF joystick, operating in a cylindrical \((r, \theta, z)\) or Cartesian \((x, y, z)\) coordinate system. If we are working in the cylindrical coordinate system, the control of the \(\theta_0\)-joint is separated from the control of the other joints.

![Manipulator geometry](image)

**Figure 4: Manipulator geometry.**

If we are working in the Cartesian coordinate system, with commanded \(\dot{x}, \dot{y}\) and \(\dot{z}\), we find the commanded \(\dot{\theta}_0\) by differentiating \(\theta_0 = \arctan(y/x)\) and using

\[
\begin{align*}
    r &= \sqrt{x^2 + y^2}, \\
    \dot{\theta}_0 &= (\theta_0 \dot{x} + c_0 \dot{y})/r
\end{align*}
\]

where \(s_0 = \sin \theta_0\) and \(c_0 = \cos \theta_0\).

The commanded \(\dot{r}\) is found by differentiating \(r = \sqrt{x^2 + y^2}\)

\[
\dot{r} = c_0 \dot{x} + s_0 \dot{y}.
\]

We have now expressed the derivatives of \(x\) and \(y\), \(\dot{x}\) and \(\dot{y}\), as the derivatives \(\dot{\theta}_0\) and \(\dot{r}\), and can concentrate our study on the kinematic control in the \(r\)-\(z\) plane, where the redundant DOF is used.

A vector \(\widetilde{\theta}\) is defined:
\[ \theta = [\theta_1, \theta_2, d_3]^T \]  

(8)

The Tool Center Point coordinates \( r \) and \( z \) defines the vector \( \mathbf{x} \):

\[ \mathbf{x} = [r, z]^T \]  

(9)

Figure 4 provides the following relationships:

\[ r = d_1 s_1 + (d_2 + d_3)s_{12} \]  

(10)

\[ z = d_0 + d_1 c_1 + (d_2 + d_3)c_{12} \]  

(11)

where \( s_{12} = \sin(\theta_1 + \theta_2) \), \( c_{12} = \cos(\theta_1 + \theta_2) \).

\[ \dot{r} = d_1 c_1 \dot{\theta}_1 + (d_2 + d_3)c_{12}(\dot{\theta}_1 + \dot{\theta}_2) + s_{12} \dot{d}_3, \]  

(12)

\[ \dot{z} = -d_1 s_1 \dot{\theta}_1 - (d_2 + d_3)s_{12}(\dot{\theta}_1 + \dot{\theta}_2) + c_{12} \dot{d}_3, \]  

(13)

or in matrix form:

\[ \mathbf{\dot{x}} = J \mathbf{\dot{\theta}}, \]  

(14)

where the Jacobian \( J \) (2x3 matrix) has elements \( j_{11}, j_{12}, ..., j_{23} \):

\[ J = \begin{bmatrix}
    d_1 c_1 + (d_2 + d_3)c_{12} & (d_2 + d_3)c_{12} & s_{12} \\
    -d_1 s_1 - (d_2 + d_3)s_{12} & -(d_2 + d_3)s_{12} & c_{12}
\end{bmatrix} \]  

(15)

Since the Jacobian \( J \) is not square, the matrix cannot be directly inverted. This problem can be solved by introducing a constraint. For this purpose a control strategy called maximum lifting capacity will be introduced in the following section.

**MAXIMUM LIFTING CAPACITY**

In some applications velocity is of major importance. However, it is not possible to have maximum velocity as a constraint since the operator will not use maximum velocity at all times, but only when he/she feels it is necessary. In other applications, velocity is of minor importance. For this application, the static (low velocity) lifting capacity is essential. The kinematic control in this work is based on an optimization study of the lifting capacity [16], as a function of \( \theta_1 \), \( \theta_2 \), and \( d_3 \), based on the force or torque characteristics and the geometrical arrangements of the cylinders. From the studies it is possible to analyze how the lifting capacity is dependent on the prismatic function \( d_3 \).
Figure 5: Selection of $d_3$ for maximum lifting capacity.

Figure 5 shows how $d_3$ should be chosen for maximum lifting capacity for a specific manipulator. Except for the lower left and upper middle part of the work area, the three curves ($d_3 = 0$, 50 and 100%) can be approximated by circles. In other cases, where the $d_3$-curves are more complicated, a look-up table for $d_3 = d_3(r, z)$ plus interpolation can be used.

In order to avoid unnecessarily large accelerations in $d_3$, some “smoothing” of the optimal $d_3 (r, z)$ function may be introduced, especially in areas in the r-z plane where the optimum is flat (i.e., where the effect of $d_3$ on the lifting capacity is small). The proposed kinematic control of $d_3$ is shown in Figure 6.
Figure 6 gives:

$$\rho = \sqrt{(r-r_e)^2 + (z-z_e)^2}$$  \hspace{1cm} (16)

where \(r_e\) and \(z_e\) are the coordinates for the centre point of the circles in the proposed kinematic control function.

The approach to describe the transition between \(\rho_{\min}\) and \(\rho_{\max}\) should be expressed as a “smooth” function. The choice of the coordinates for \(r_e\) and \(z_e\) can be done according to Figure 5 and Figure 6 and the geometrical data from a specific boom.

In the zone where \(d_3\) is active it should vary smoothly without large accelerations. We have chosen a function given in equation (17) according to Figure 7. This function has \(\frac{d d_3}{d \rho} = 0\) for \(\rho = \rho_{\min}\) and for \(\rho = \rho_{\max}\) thus avoiding jumps in \(d_3\)'s velocity.

. 

Figure 6: \(d_3\) as a function of TCP’s position.
Figure 7: $d_3$ as a function of $\rho_i$

The smoothing function depicted in Figure 7 is given by equations (17) to (19).

\[ d_3 = \frac{d_{3\text{max}}}{2} \left[ 1 + \frac{3(\rho - p)}{2q} - \frac{(\rho - p)^3}{2q^3} \right] \]  \hspace{1cm} (17)

with

\[ p = \frac{\rho_{\text{max}} + \rho_{\text{min}}}{2} \]  \hspace{1cm} (18)

\[ q = \frac{\rho_{\text{max}} - \rho_{\text{min}}}{2} \]  \hspace{1cm} (19)

Differentiating equation (17) gives

\[ \dot{d}_3 = f(r, \rho)\dot{r} + g(z, \rho)\dot{z} \]  \hspace{1cm} (20)

with

\[ f(r, \rho) = \frac{c(r - r_c)}{\rho} \quad \text{for} \quad \rho_{\text{min}} \leq \rho \leq \rho_{\text{max}} \]

\[ g(z, \rho) = \frac{c(z - z_c)}{\rho} \quad \text{for} \quad \rho_{\text{min}} \leq \rho \leq \rho_{\text{max}} \]  \hspace{1cm} (21)

\[ f(r, \rho) = g(z, \rho) = 0 \quad \rho < \rho_{\text{min}} ; \rho > \rho_{\text{max}} \]

where
Independent of what characteristics we use for $d_3$ we can simplify our notations: $f(r, \rho) = f, g(z, \rho) = g$.

Using $\dot{d}_3$ from equation (20) in equation (16) gives:

$$
\begin{bmatrix}
\dot{r} \\
\dot{z}
\end{bmatrix}
= 
\begin{bmatrix}
j_{11} & j_{12} & j_{13} \\
j_{21} & j_{22} & j_{23}
\end{bmatrix}
\begin{bmatrix}
\dot{\theta}_1 \\
\dot{\theta}_2
\end{bmatrix}
+ 
\begin{bmatrix}
\dot{f} \\
g
\end{bmatrix}
$$

(23)

Rewriting equation (23) gives

$$
\begin{bmatrix}
j_{11} & j_{12} \\
j_{21} & j_{22}
\end{bmatrix}
\begin{bmatrix}
\dot{\theta}_1 \\
\dot{\theta}_2
\end{bmatrix}
= 
\begin{bmatrix}
1-j_{13}f & -j_{13}g \\
-j_{23}f & 1-j_{23}g
\end{bmatrix}
\begin{bmatrix}
\dot{r} \\
\dot{z}
\end{bmatrix}
$$

(24)

or

$$
\begin{bmatrix}
\dot{\theta}_1 \\
\dot{\theta}_2
\end{bmatrix}
= 
\begin{bmatrix}
j_{11} & j_{12} \\
j_{21} & j_{22}
\end{bmatrix}^{-1}
\begin{bmatrix}
1-j_{13}f & -j_{13}g \\
-j_{23}f & 1-j_{23}g
\end{bmatrix}
\begin{bmatrix}
\dot{r} \\
\dot{z}
\end{bmatrix}
$$

(25)

Equations (20), (25) and (15) give, after some calculations, the kinematic control law

$$
\dot{\theta} = P\dot{x}
$$

(26)

where the $(3 \times 2)$ $P$ matrix has the following elements:

$$
\begin{align*}
p_{11} &= D(s_{12} - f)/(d_is_{2}) \\
p_{12} &= D(c_{12} - g)/(d_is_{2}) \\
p_{21} &= D[d_if(s_{12} - f)/(d_is_{2}) - (d_2 + d_3) - s_{12}]/(d_is_{2}) \\
p_{22} &= D[d_ig(s_{12} - c_1)/(d_2 + d_3) - c_{12}]/(d_is_{2}) \\
p_{31} &= f \\
p_{32} &= g
\end{align*}
$$

(27)

where

$$
D = \frac{1}{d_i(d_{23} - ac_2)}
$$

(28)
Velocity limitations

The joints provide maximum velocities $\dot{\theta}_{1\max}$, $\dot{\theta}_{2\max}$ and $\dot{d}_{3\max}$, respectively. If a joint (for example, joint 1) receives a command signal $\dot{\theta}_{c1}$, with $|\dot{\theta}_{c1}| > \dot{\theta}_{1\max}$, the velocity limitations will cause a position error. This problem is solved in the following way:

Introduce:

$$
\begin{align*}
\alpha_i &= \left| \frac{\dot{\theta}_i}{\dot{\theta}_{i\max}} \right|, \quad i = 1, 2, 3 \\
\alpha_3 &= \left| \frac{\dot{d}_3}{\dot{d}_{3\max}} \right|
\end{align*}
$$

(29)

In a practical case, the geometrical arrangement of the $\theta_1$ - and $\theta_2$ - joints (these revolute joints may be driven by hydraulic cylinders) may cause $\dot{\theta}_{1\max}$ and $\dot{\theta}_{2\max}$ to be functions of $\theta_1$ and $\theta_2$, respectively, and directions. In this case, joint speeds are set as equal in both directions. This is due to the fact that for each machine application the hydraulic pressure and flow are different for each machine application. Therefore the joint speeds are set to suit one typical pressure and flow.

We find the largest $\alpha_i$ - value, $\alpha_{\max}$:

$$
\alpha_{\max} = \max \{ \alpha_i \} \\
i = 1, 2, 3
$$

(30)

If $\alpha_{\max} \leq 1$, there are no velocity limitations, and $\dot{\theta}$ is determined by equation (14).

Assume that the joystick is working in the cylindrical coordinate system, and that the $\theta_0$ - joint is separately controlled.

Assume $\alpha_1 = \alpha_{\max} > 1$

Choose $\dot{\theta}_1 = \dot{\theta}_{1\max}$ if $\dot{\theta}_1 > 0$ and $\dot{\theta}_1 = -\dot{\theta}_{1\max}$ if $\dot{\theta}_1 < 0$. Since we have lost one DOF (i.e., the system is non-redundant), the calculation of $\dot{\theta}_2$ and $\dot{d}_3$ is straightforward ($r_c$ and $z_c$ are commanded velocities).

$$
\begin{bmatrix}
\dot{r}_c \\
\dot{z}_c
\end{bmatrix} = J
\begin{bmatrix}
\pm \dot{\theta}_{1\max} \\
\dot{\theta}_{2c} \\
\dot{d}_{3c}
\end{bmatrix}
$$

(31)

giving
\[
\begin{bmatrix}
\dot{\theta}_{2c} \\
\dot{d}_{3c}
\end{bmatrix} = \frac{1}{D_1} \begin{bmatrix}
j_{23} & -j_{13} \\
-j_{22} & j_{12}
\end{bmatrix} \begin{bmatrix}
\dot{r}_c & \mp j_{11} \cdot \dot{\theta}_{1\text{max}} \\
\dot{z}_c & \mp j_{21} \cdot \dot{\theta}_{1\text{max}}
\end{bmatrix}
\]

(32)

with

\[D_1 = j_{12}j_{23} - j_{13}j_{22}\]

To find out if the new values for \(\dot{\theta}_{2c}\) and \(d_{3c}\) given by (32) will exceed their maximum velocities, \(\alpha_2\) and \(\alpha_3\) must be calculated again, giving \(\alpha'_2\), \(\alpha'_3\) and \(\alpha'_{\text{max}}\). Control law (32) is used if \(\alpha'_{\text{max}} \leq 1\). If \(\alpha'_{\text{max}} > 1\), two joints, \(\theta_1\) and \(\theta_2\) or \(d_3\) must be working at their maximum velocities, and hence we now have only one DOF, and two DOFs are necessary to follow a commanded path in the \(r-z\) plane.

Assume that \(\alpha'_{\text{max}} = \alpha'_2\), i.e., \(|\dot{\theta}_2| > \dot{\theta}_{2\text{max}}\). Choose \(\dot{\theta}_2 = \dot{\theta}_{2\text{max}}\) if \(\dot{\theta}_2 > 0\) or \(\dot{\theta}_2 = -\dot{\theta}_{2\text{max}}\) if \(\dot{\theta}_2 < 0\). The commanded velocity must be scaled by a factor \(\beta < 1\), since we cannot follow the commanded path with two joints at their maximum speeds due to the fact that we have lost two DOFs:

\[
\begin{bmatrix}
\beta \dot{r}_c \\
\beta \dot{z}_c
\end{bmatrix} = \begin{bmatrix}
j_{11} & j_{12} & j_{13} \\
j_{21} & j_{22} & j_{23}
\end{bmatrix} \begin{bmatrix}
\pm \dot{\theta}_{1\text{max}} \\
\pm \dot{\theta}_{2\text{max}}
\end{bmatrix}
\]

(34)

Equation (34) has two unknowns, \(\beta\) and \(d_3\).

We find:

\[
\beta = \frac{j_{23}k_1 - j_{13}k_2}{j_{23}\dot{r}_c - j_{13}\dot{z}_c}
\]

(35)

and

\[
\dot{d}_3 = \frac{\beta \dot{r}_c - k_1}{j_{13}}
\]

(36)

where

\[
k_1 = \pm j_{11}\dot{\theta}_{1\text{max}} \pm j_{12}\dot{\theta}_{2\text{max}}
\]

(37)

\[
k_2 = \pm j_{21}\dot{\theta}_{1\text{max}} \pm j_{22}\dot{\theta}_{2\text{max}}
\]

(38)

If instead \(\alpha'_{\text{max}} = \alpha'_3\), similar calculations will apply.
Similar calculations will also apply if we have speed limitations in $\theta_2$ or $d_3$, i.e.,

$$\alpha_2 = \alpha_{\text{max}} \quad \text{or} \quad \alpha_3 = \alpha_{\text{max}}.$$  

**Mechanical limitations**

If one of the $\theta_1$, $\theta_2$ - or $d_3$ - joints reaches a mechanical limit (actually a software limit occurs before the joint has reached a mechanical limit), we are losing our redundant degree of freedom, but can still follow a desired path (until a second joint reaches a mechanical limit). To solve this, $\dot{\theta}$ is first calculated by means of equation (26). If the $\theta_1$ – joint is at a mechanical limit and equation (26) shows that the joint should surpass the mechanical limit, we have to perform a second calculation exactly as done in the section Velocity Limitations, but now with $\dot{\theta}_{1_{\text{max}}}$ replaced by 0. Similar calculations are made for $\theta_2$ and $d_3$ when they reach their mechanical limits. In order to avoid large transients when approaching a mechanical limit, the maximum velocity used for calculations is decreased for that joint.

**SIMULATIONS**

The maximum lifting capacity control algorithm has been tested through simulations in a program developed especially for this work. We have limited the simulations to one typical task. The task describes a normal working cycle, which occurs when a forwarder is loading and unloading logs from the ground and off/on the carrier. The task consists of three linear segments in the workspace, as shown (ABC) in Figure 8. The paths are as follows: A–B, B–C and C–A. The position and velocity of the joints are analyzed for each path. The coordinates for each point in the workspace are:

A (1.5, 1.0)  
B (5.5, 1.0)  
C (5.5, -3.0)
Figure 8: Workspace and simulation task.
For the simulations we have used following data. The workspace of the boom Cranab 850 is seen in Figure 8. The joint limits are:

\[
\begin{align*}
\theta_{1\text{min}} &= 8\text{deg} \\
\theta_{1\text{max}} &= 123\text{deg} \\
\theta_{2\text{min}} &= 2\text{deg} \\
\theta_{2\text{max}} &= 178\text{deg} \\
d_{3\text{min}} &= 0m \\
d_{3\text{max}} &= 1.4m \\
r_c &= 3.5m \\
z_c &= 0.65m \\
\rho_{\text{min}} &= 3.5m \\
\rho_{\text{max}} &= 4.9m \\
\end{align*}
\]

The maximum velocities of the joints that are used for all simulations are:

\[
\begin{align*}
\dot{\theta}_{1\text{max}} &= 0.88 \text{ rad/s} \\
\dot{\theta}_{2\text{max}} &= 1.23 \text{ rad/s} \\
\dot{d}_{3\text{max}} &= 0.42 \text{ m/s} \\
\end{align*}
\]
The joint speeds are set as equal in both directions.

The following figures from the simulations show the time records of $\theta_1, \theta_2, d_3, \dot{\theta}_1, \dot{\theta}_2$ and $\ddot{d}_3$ when the Tool Centre Point moves from point A to B, B to C and C to A in the workspace at a velocity of 1.0 m/s. The theoretically achievable positioning times at these speeds are as follows:

A to B and B to C: 4 s

C to A: 5.66 s

If these times are not met, it is an indication that one or more of the DOFs are saturated. The joints are limited to maximum allowable velocities. To be able to display $d_3$ on the same plot as $\theta_1$ and $\theta_2$, $d_3$ has been multiplied with a constant set to 100.
TCP moves from point A to B.

Figure 9: TCP moves from A to B with a speed of 1.0 m/s. Variations in $\theta_1$ and $\theta_2$. $d_3$ is not moving.

Figure 10: TCP moves from A to B with a speed of 1.0 m/s. Variations in $\theta_1$ and $\theta_2$. $d_3$ is not moving.

The algorithm needs 4.0 seconds to go from point A to B, which is the same time as the theoretically achievable time. The velocities of the joints change smoothly in both cases. Only the joints $\dot{\theta}_1$ and $\dot{\theta}_2$ are used, and none of the joint velocities reach their speed limits.
TCP moves from point B to C.

![Figure 11: TCP moves from B to C with a speed of 1.0 m/s. Variations $\theta_1$, $\theta_2$ and $d_3$.](image1)

The algorithm needs 4.0 seconds to go from point B to C. $d_3$ reaches its maximum velocity limit at approximately at 3.9 seconds and a corresponding change in velocity profile for $\dot{\theta}_1$ and $\dot{\theta}_2$ can be seen.

![Figure 12: TCP moves from B to C with a speed of 1.0 m/s. Variations in $\dot{\theta}_1$, $\dot{\theta}_2$ and $\dot{d}_3$.](image2)
TCP moves from point A to C.

Figure 13: TCP moves from A to C with a speed of 1.0 m/s. Variations in $\theta_1$, $\theta_2$ and $d_3$.

Figure 14: TCP moves from A to C with a speed of 1.0 m/s. Variations in $\dot{\theta}_1$, $\dot{\theta}_2$ and $\dot{d}_3$.

The algorithm needs 5.77 seconds to go from point A to C, which is 0.11 seconds longer than the theoretically achievable time. $\theta_2$ and $d_3$ reach their maximum velocity limits. A scaling of the commanded signal according to (30) is necessary, which increases the time consumption. During the scaling, the path is followed, but with a velocity lower than 1.0 m/s.

CONCLUSIONS

An algorithm for computation of the inverse kinematics of kinematically redundant hydraulic manipulators has been investigated. The hydraulic manipulator used in the simulation study consists of a 3 DOF hydraulic forestry machine manipulator. The simulations show the necessary speed requirements for all joints when performing straight paths in the manipulator work area. The simulations also show the time consumptions.
Although this work was an attempt to propose complete algorithms that could be directly applicable to a real-life system, it is still only a simulation study and lacks the realism that would come from implementation on a real hydraulic manipulator.

In future work, the maximum lifting capacity method will be tested in a real-time forestry simulator. The method will be evaluated by forestry machine operators to see if the boom tip control could in fact increase productivity. Semi-automated functions will also be tested.

REFERENCES


Paper B

The Fidelity of a Real-Time Forest Machine Simulator

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Keywords:

Fidelity, forestry machine simulator, real-time simulator.
ABSTRACT: It is essential to reduce the mental and physical stress on forest machine operators. The operator in a harvester cuts down one tree every 47 seconds, makes 12 decisions per tree and performs an average of 24 functions per tree. In Sweden, the forestry industry uses the Cut To Length (CTL) method, which means that the tree is cut into pieces on site in the stand. This is done 1000 times a day. A solution that would help harvester operators is the use of full or semi-automation, as well as other ways of improving Human-Machine Interaction (HMI). It is not practical or cost-effective to initially develop automation or HMI ideas on real machines; a better solution is to use simulators. Normally, existing forest machine simulators are used in teaching future forest machine operators. In our case, we used the simulator as a research tool. To rely on the results coming from tests with the simulator, we performed a fidelity test. We conducted a time study in which a harvester operator cut down approximately 500 trees, and also measured data from the stand, including tree diameter, height, position, height to first live branch, and tree type. We also measured the terrain. The same stand and terrain data was fed into the simulator and the same operator performed the same work again. The results demonstrated that there is good fidelity between a real forest machine and the simulator. The time difference between reality and the simulator is just ± 5% for different boom operations. Qualitatively, the results were on par. Several aspects of simulator fidelity will be discussed in this paper.

1. Introduction

The use of forest machine simulators has increased in the context of training new forest machine operators, and the question has been raised as to whether it is possible to use the simulators as a research tool as well. The subsequent question is whether the simulators are reproducing reality in a way that is good enough to achieve reliable results.

More than a decade ago, the Simulation Interoperability Standards Origination (SISO) was established to facilitate information exchange and the standardization of simulator use among different companies and research organizations working with simulations. Within SISO, a group called Fidelity Implementation Study Group (ISG) attempted to standardize the nomenclature used in simulation. Among other things, they defined what is meant by fidelity and validity [4]. The following definitions have been suggested:

Fidelity: The degree to which a model or simulation reproduces the state and behavior of a real world object or the perception of a real world object, feature, condition, or chosen standard in a measurable or perceivable manner; a measure of the realism of a model or simulation; faithfulness.

Validity: The quality of being inferred deduced or calculated correctly enough to suit a specific application, with particular application to a model or simulations representational capability or logical truth of a derivation or statement, based on a given set of propositions.

The question about what fidelity and validity means in conjunction with simulation has been frequently discussed during the last 15 to 20 years; an exact, established definition does not yet exist because there are many parameters that influence a simulation.
1.1 Fidelity

The concept of fidelity relates to what extent the simulator’s characteristics meet the features of a real system. There are no simple descriptions of a simulator’s fidelity. Over twenty different definitions of fidelity exist in the literature. Some examples are equipment fidelity, fidelity of the surrounding environment, psychological fidelity, fidelity of the task, and functional fidelity [4]. Each type of fidelity can be applicable to a specific situation, which means that one specific type can, in general, be used for all types of simulations. However, together they will demonstrate, in at least two dimensions, a difference between simulators and a real vehicle. These two dimensions divide the simulators into two groups, depending on what kinds of cues the simulators provide:

- The equipment cues refer to the experienced compatibility between special features in the simulator and the real system.
- The surrounding cues refer to how well the simulator mimics the surrounding environment and movements therein.

Fidelity captures how well the equipment and the surrounding cues are describing a real situation. There is a distinction between real cues, measured objectivity, and the cues that the operator experiences, which leads to the following definitions of two types of fidelity:

- Objective fidelity; and
- Experienced fidelity.

To be able to provide a judgment of how well the forest machine simulator reproduces a real situation to a harvester operator, we need to know which factors can influence the obtained results. There are several factors affecting a human being’s interpretation of the surrounding world and their perception in a virtual environment, as well as how this is affected by the physical characteristics of the simulator itself.

1.2 Validity

A simulator must emulate the real world to enable, for example, evaluation of human-machine interaction (HMI) and new futuristic solutions. There are two levels of validity for a simulator. The first indicates how well the simulator emulates the behavior of the operator; this is called behavior validity. The second level is about the physical data, layout and dynamic characteristics of the simulator; this is called physical validity and is, in general, labeling the fidelity of the simulator.

Behavior validity

Behavior validity is very important to take into consideration regardless of whether the research takes place in a simulator or in a real-life setting. If a simulator is used for tests, it is very important that the operator decisions made in the simulator agree with the
operator decisions made in a real-life setting. For a forest machine, it is important to reproduce the operator responses that occur in the natural forest environment.

There are several methods for evaluating the validity of a simulator. The best method is to compare a simulator and a machine as they perform the same task under the same conditions. If the numerical values are identical on the two systems, you have absolute validity. Another method, called relative validity, is to compare the performance between the two systems. It is, of course, possible to combine these two methods. If the simulator should be useful for research, relative validity is a must, but absolute validity is not necessary [5]. This is due to the fact that research questions normally relate to the effect of independent variables rather than to numerical values.

Physical validity

The more a simulator reproduces reality in terms of its use, how the response from input is presented on the screen, and how accurately the simulator reacts to physical input, the higher the grade of fidelity it has. For this reason, a simulator that has a moving platform is said to have a higher grade of fidelity. This does not mean, however, that a more sophisticated simulator automatically has higher fidelity than a less sophisticated one when it comes to behavior validity.

This paper presents a study of a forest harvester simulator compared to a real forest harvester.

2. The Forest Machine Simulator

The simulator is a real-time forest machine simulator. All movements are based on input from an operator and not on any pre-implemented sequences such as in a game. All movements on the forest machine are based on physical data such as weight, geometry, friction, moment, and other dynamic qualities.

The simulator has three screens, each screen being 2 x 1.5 meters in size; with 1400 x 1050 pixels on each screen (see Figure 1). The frame rate is 40 frames per second.

![Figure 1: The forest machine simulator.](image-url)
3. Field tests

To be able to compare the results from a final felling operation with the simulator, we needed to measure and cut down a specific stand. Besides measuring the stand itself, the terrain and obstacles such as stones were also measured. To achieve a better understanding of how the operator used different functions, such as harvester pushbuttons and joysticks, all machine functions were also registered. In addition, a conventional time study was performed.

3.1 Data Collection

In total, 483 trees were measured. Each tree received a specific number, one on the diameter at breast height (used for the time study), and one at the buttress (used for positioning of the tree).

The following parameters were measured for each tree:

- type of tree (spruce, pine or birch);
- the diameter at breast height;
- tree height;
- height to first live branch; and
- irregularities on the stem form.

To be able to recreate the stand in the simulator, the position of each tree was measured with differential GPS to an accuracy of ± 2 centimeters in x-, y- and z-directions. The positions of the trees were measured after the final cutting. We also measured the position, number of trees, assortment, diameter and length of each log in each pile. The GPS data was put into GIS software, and a coordinate was created so that the data could be transmitted to the simulator.

The CAN (Controller Area Network) bus was used to collect information about all of the functions used by the operator (see Figure 2). A special program was developed to listen in on the CAN bus, which connects all of the distributed computers on the machine. The information on the CAN bus was registered 10 times per second.
3.2 Time studies

The operator was instructed to cut down the trees according to normal routines. The operator choose to use single-side felling technique, which means:

- the machine was driving along the outer side of the stand;
- the trees were put down diagonally forward or into the stand;
- the logs were put into piles on the outer side of the machine; and
- the trees were cut down in a 13- to 14-meter-wide band.

3.3 Machine and operator

The harvester machine was a three-year-old Valmet 921. The operator had 15 years of experience managing single grip harvesters.

4. Results

In this part of the paper the results from the data collection and time studies are presented.

4.1 Positioning

From the GPS data we could see where the machine had been driving, and also the position of each tree (see Figure 3 and Figure 4).
We could also obtain information on where the machine had been standing when the trees were cut. The size of the round dot represents the diameter of the tree, and the color represents what kind of tree it was.

4.2 Button and joystick functions

The pushbuttons and joysticks were registered in the field and in the simulator. Two trials were performed—one in the morning with 81480 registrations and one in the afternoon with 97952 registrations. A more detailed analysis of the registrations is presented in [2].

Due to problems listening in on the CAN bus in the simulator, an interval of 15 minutes was selected, representing 21 trees (21 working cycles). The same trees in the field and in the simulator were used to make comparisons. Each working cycle (handling of one tree) starts when the harvester head grasps the tree just before starting to cut it.
There are some significant differences between the simulator and reality when working with the harvester head. One is the use of reverse of the feeding rollers reverse on harvester head, which was used three times in the simulator and 17 times in the field. In the simulator, limbing and feeding occurs perfectly, free of problems with slipping or with big branches. The top of the trees had to be cut manually in the simulator, thereby creating a large difference when compared to the field studies.

Due to variations of deflection and time, it is harder to directly compare the use of the buttons and joysticks between the field and the simulator. To receive as good fit as possible between the machine and the simulator, the operator sets all of the adjustable joystick functions (ramps, dead band, etc.) in the simulator to mimic the behavior of the machine as closely as possible. Even though the functions of the joysticks were adjusted, there remained differences between the machine and the simulator. The hydraulic system in the machine is a dynamic system, which significantly influences crane behavior; in the simulator it is not completely modeled. Examples of one joystick function from the simulator and reality are depicted in Figures 5 and 6.

![Figure 5: Joystick function for the first section of boom (sample rate 10 Hz) in the simulator.](image1)

![Figure 6: Joystick function for first section of boom (sample rate 10 Hz) in the machine.](image2)
4.3 Time studies

The same observer was used in both time studies, shown in Table 1 and divided into different actions.

<table>
<thead>
<tr>
<th>Field study</th>
<th>Simulator</th>
<th>Difference in %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boom out</td>
<td>11.71</td>
<td>10.86</td>
</tr>
<tr>
<td>Felling-intake</td>
<td>15.22</td>
<td>16.01</td>
</tr>
<tr>
<td>Delimbing-cutting</td>
<td>42.16</td>
<td>43.88</td>
</tr>
<tr>
<td>Moving the machine</td>
<td>8.64</td>
<td>5.32</td>
</tr>
<tr>
<td>Other</td>
<td>5.59</td>
<td>2.34</td>
</tr>
<tr>
<td>Total</td>
<td>84.12</td>
<td>78.41</td>
</tr>
</tbody>
</table>

**Boom out**

Boom out was faster in the simulator, probably due to a lack of low vegetation or obstacles such as stones or stumps. The operator was also more careful in the field than in the simulator in order to avoid damaging the sword or the harvester head.

**Felling-intake**

Although normally two separate actions, the felling and intake of the harvester head has been combined due to the fact that it is difficult to determine when the action of intake actually starts. In the simulator, the time study observer can see when the operator moves the joysticks; this is the time that is registered. In the field, the time study observer stands outside the machine, making it easier to separate the two actions. The action took longer in the simulator. A probable cause is that the operator made longer intakes of the trees, in time, in the simulator. The operator is less sensitive regarding the placement of the machine because there are now motions in the simulator. The boom was also stronger in the simulator due to greater force, which means that the operator had been standing still for a longer time and in the same place. The same phenomena had been noticed in comparing a small and large harvester, where the bigger harvester has a stronger boom. This also affects the time consumption in moving the machine.

**Delimbing-cutting**

There is a strong agreement between the simulator study and the field study.

**Moving the machine**

The time difference is strongly connected to the intake. Since the operator is standing still more frequently, there are fewer movements of the machine. The number of movements is 13% less in the simulator. The big difference in time is connected to the fact that it will take longer to move the machine in the field than in the simulator, since the terrain has much more influence when driving in the field. A simulator with a motion platform would most likely improve the correspondence in this regard.
Other

The label other in Table 1 refers to all other time that is needed to complete the work, such as problems when delimbing, when dropping a tree, when handling forked stems or when putting stems in order. The large difference in time consumption is due to the fact that there were no problems with delimbing, and no forked or crooked stems in the simulator.

4.4 Stem profiles

When cutting the stand, the mean stem volume (according to the bucking computer in the machine) was 507 m$^3$ (solid under bark). In the simulator (again, according to the bucking computer) this was 710 m$^3$. This substantial difference in volume (40%) led to a deeper analysis as to why the difference occurred.

The measuring accuracy of the machine

To analyze the measuring accuracy of the machine, data from 46 trees in another stand was collected. The mean values from the stems are presented in Table 2.

Table 2: Arithmetic mean value for test trees.

<table>
<thead>
<tr>
<th></th>
<th>Pine</th>
<th>Spruce</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of stems</td>
<td>22</td>
<td>24</td>
</tr>
<tr>
<td>Height (m)</td>
<td>24.1</td>
<td>21.3</td>
</tr>
<tr>
<td>Callipered breast height diameter</td>
<td>32.6</td>
<td>23.7</td>
</tr>
<tr>
<td>Machine measured breast height diameter</td>
<td>31.3</td>
<td>23.5</td>
</tr>
</tbody>
</table>

Manual measuring of diameters and length were conducted using a caliper and measuring tape. The results from these measurements are shown in Table 3.

Table 3: Difference between manual and machine

<table>
<thead>
<tr>
<th></th>
<th>Mean difference</th>
<th>Standard deviation</th>
<th>Number of observations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter, mm</td>
<td>1.86</td>
<td>9.15</td>
<td>74</td>
</tr>
<tr>
<td>Length, cm</td>
<td>-0.09</td>
<td>1.5</td>
<td>34</td>
</tr>
<tr>
<td>Volume, %</td>
<td>-0.35</td>
<td>2.66</td>
<td>8</td>
</tr>
</tbody>
</table>

The conclusion is that, on average, the machine measured the diameter, length and volume with satisfactory precision.

Analysis of simulator measurements

To compare the calculations of the stem volume with different methods, 50 new trees were created with the actual stem volume method in the simulator. These trees were cut down and the volume was measured in the simulator. The results with regard to the mean stem volume are presented in Table 4, Figure 7 and Figure 8 in respect to Brandel volume function, [1], and Edgren-Nylinder volume function, [3].

Table 4: Mean stem volume. In the parentheses is the difference according to the simulator.

<table>
<thead>
<tr>
<th></th>
<th>Harvester measured</th>
<th>Brandel volume function</th>
<th>Edgren-Nylinder volume function</th>
<th>Simulator</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pine</td>
<td>823 (56%)</td>
<td>829 (56%)</td>
<td>851 (58%)</td>
<td>1475</td>
</tr>
<tr>
<td>Spruce</td>
<td>429 (59%)</td>
<td>453 (62%)</td>
<td>441 (61%)</td>
<td>728</td>
</tr>
</tbody>
</table>
A more detailed analysis of the diameter vector (a diameter measure per dm module in length) for each stem (along each stem) revealed a systematic difference. This is illustrated in Figures 9 and 10, for two of the stems. These trees have been subjectively chosen to demonstrate the systematic error. There are large individual differences. From Figures 9 and 10 it is also clear that all the stem profiles in the simulator have a minimum diameter of 75 mm.
Figure 9: Comparison of the taper for stem 54, Scots pine, (diameter vector) measured in the simulator and machine and calculated according the Edgren-Nylinder function.

Figure 10: Comparison of the taper for stem 56, Norway spruce, (diameter vector) measured in the simulator and machine and calculated according the Edgren-Nylinder function.

The results from the volume calculation analysis show that the volume functions in the simulator gave wrong results.

5. Discussion

5.1 Time studies

From the time studies we can see that there is good fidelity between the simulator and the machine with regard to performance and time consumption of boom operations. We have a difference of about ± 5% when using the boom and the harvester head. The largest time difference occurs when the harvester is being moved between two workspaces in the stand. This is due to the fact that the crane is stronger in the simulator than on the machine (which means fewer machine movements), as well as faster driving of the machine in the simulator environment than in a real stand.
5.2 Buttons and joystick functions

When it comes to buttons we have larger differences, mostly when it comes to the delimbing of the tree. In the simulator we have ideal conditions—we have no problems with branches and there is enough feeding force.

The joystick movements in the simulator and on a real machine are quite similar. The difference that does arise is that joystick movements are larger in the machine. For the telescope and rotator functions, the differences are smaller, and for the swing and lifting functions they are larger. Other differences are that the joystick movements are jerkier in the machine, due to the machine’s vibrations. In the simulator there are no vibrations since there is no moving platform.

5.3 Stem profiles

The analysis of the stem profiles showed a relatively large difference between measured values, and a large variance due to large individual variations. In order to estimate the differences, the Edgren-Nylinder function was implemented.

6. Conclusion

The results show that there is good fidelity between the forest machine simulator and a real forest machine, especially regarding boom functions, and that the results—to be generated by different forthcoming studies on the simulator—will be reliable.

7. References


Author Biographies

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Paper C
BOOM TIP CONTROL – An evaluation of boom tip control for forest machines in a real-time forest machine simulator.

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ABSTRACT

At a time when forest operations have become fully mechanized, we are entering a new era in which we will probably see increasing automation of the industry. This process will involve the automation of entire operations, including knuckle boom manipulation and other parts of the process such as the collection, transmission and reporting of information.

A key incentive for bringing in automation is to improve working conditions for the machine operator. While much has already been done over the years to reduce the physical stress to which operators are subjected, the higher pace of operations today, combined with the many qualified decisions that the operators must make under pressure, is imposing greater mental stress on them. Automating the boom operation would reduce mental pressure on the operator. Detailed time studies have been carried out on a case where simplified boom control has been introduced, and tests carried out in a simulator clearly demonstrate that simplified boom control is a feasible way of increasing productivity, shortening training time, and improving operators’ working conditions.

INTRODUCTION

As a result of an intense effort to mechanize the Swedish large-scale forestry industry since the middle of the 1960s, the industry today is almost 100% mechanized. This is one of the main explanations as to why Swedish forestry has been able to maintain its international competitiveness; to sustain this leading position into the future, the industry’s productivity levels must continue to increase.

The forestry machines of today are high-tech units with advanced control engineering. Technological developments have resulted in a radical increase in performance. For the operator, this has meant increased workload and fewer planned and unplanned stops during typical work. At the same time, quality control requirements in the drivers’ workflow have also changed. It is no longer sufficient to simply control the machine and its functions—the operators are also responsible for meeting environmental concerns, and for the planning and follow-up of their work (Löfgren, 2004). While the working environment with regard to physical stress on the operator has improved considerably over the years, the increased workload, in combination with the many decisions that the driver must make, mean that forest machine operators are exposed to great mental and physical stress (Attebrant et al., 1998). Operating a forest machine places high demands on the coordination abilities of the operators, since the machine is controlled with two joysticks (one in each hand) and foot pedals (Attebrant et al., 1996; Gellerstedt, 1993). Over time, these functions will induce substantial physical stress on the operator due to the necessarily quick, exact and repetitive movements of the hands, arms and head (Hägg, 2001). Moreover, the work has no natural breaks or pauses, which are very important for giving muscles a chance to recover from the workload (Ericsson & Odenrick, 1997; Löfgren, 2004). As a result, the operator’s work needs to be simplified in order to provide him/her with the opportunity to focus on other tasks and to mitigate the risk of mistakes and hasty decisions (Attebrant et al., 1994). After all, the operator is controlling the boom up to 55% of his/her working time on a forwarder and up to 95%
of the time on a harvester. Hence, our hypothesis is that simplified control of the boom would reduce physical and mental stress and simultaneously decrease training time and increase productivity. Forest machine operators experience a higher frequency of repetitive strain injuries than they did 50 years ago, despite increased mechanization and more easily maneuverable functions (Eklund & Cederqvist, 1998; Attebrant et al., 1998).

Whether the machine is a forwarder, harvester or another type of machine with a joystick control, the process of learning how to control a forest machine is a similar one. A concept that Hägg (2001) uses to describe the learning process is “motor learning”. This type of learning process can be divided into three phases. In the first phase, the novice tries to understand the task to be performed by analyzing the situation. In phase two, the associative phase, the novice tries to solve the task by identifying previous motor experiences that could be combined with each other to address the task at hand. During this phase, concentration and mental stress are very high. The third, final and longest phase completes the mental program—it is performed by learning the required motion patterns through repetition, and is done subconsciously. This also means that, consequently, the mental stress decreases successively (Alm & Ohlsson, 2003). At the same time, negative succession on the muscular system will appear as the motions are performed subconsciously, in that the flexibility of the involved muscles will decrease. This depends on changed motion patterns and new energy-saving muscle activations (Hägg, 2001).

The process of learning how to drive a car is a useful comparison (Alm & Ohlsson, 2003). In the first phase, the driver must learn where the pedals and controls are. In phase two, driving begins—concentration is very high and the driver is focused on how to drive (for example, the driver has difficulties maintaining fluent conversations while driving). In the final phase, many of the motions have been fully learned and have become automatic, and there are no longer any difficulties talking and listening to the radio while driving. It is, however, difficult to change practiced motion patterns. The process of learning boom control is similar to the process of learning how to drive a car.

Stress can be defined as a physical and mental state in which a person feels powerlessness. The level of stress can depend on many things, including the amount of information one must deal with, time restrictions on solving a task, or a requirement to perform a role beyond the capability of the individual (Alm & Ohlsson, 2003). The physiological measure of stress is the amount of stress hormones (i.e., catecholamines such as adrenaline, noradrenalin and cortisol) in the blood (Allwood, 1997). In general, the ability to assimilate and use information in an optimal way will decrease with extreme values of stress, time restrictions, the degree of difficulty, and tiredness. The lack of control on the working process can lead to decreased motivation to work and feelings of meaninglessness, which will increase mental stress even further.

Selye (1956) suggests a more biological definition of stress called the “General Adaption Syndrome”, or GAS. While stress is reflected by the sum of non-specific changes as they develop over time during continued exposure to a stressor, GAS encompasses all non-specific changes as they occur during continued exposure to a stressor.
Thus, GAS may be defined as the manifestation of stress in the whole body, as it
develops over time. A fully-developed GAS consists of three stages: the alarm reaction,
the stage of resistance, and the stage of exhaustion. Yet it is not necessary for all three
stages to develop before GAS exists. Only the most severe stress leads to exhaustion.
Most of the physical or mental exertions, infections or other stressors that are introduced
during a limited period produce changes corresponding only to the first and second
stages; at first, they may upset and alarm us, but then we adapt to them.

In the long run, stress will lead to physical troubles. Mental stress increases muscle
tension and can delay healing processes for work-related injuries (Lundberg, 2003).
Examples of stress-related diseases include high blood pressure, strokes, gastric ulcers,
migraines and asthma (Alm & Ohlsson, 2003), as well as dizziness and insomnia
(Attebrant et al., 1998).

The Yerkes-Dodson law describes the relationship between stress and performance. In
this case, stress is called “activation level”. The relationship between stress and
performance can be described as an inverted u-curve, which shows that too much or too
low activation has a negative effect on performance. Increased activation, physical or
mental, leads to faster problem-solving but also an increased number of faults (Yerkes &
Dodson, 1988; Allwood, 1997). The appropriate level of activation depends on the task
performed. Activities involving high mental complexity and decision-making benefit
from low activation compared to tasks that have been automated (Allwood, 1997). The
harvester operator’s work, which requires a level of decision-making over and above
normal boom control, benefits from lower activation levels; the forwarder operator
benefits from higher activation levels.

Even if the physical stress is low in forestry machines, the long work days and lack of
variation in the work itself will, over the long run, lead to stress-related injuries
(Attebrant et al., 1998). The control of joysticks demands exact, repetitive movements at
a high operational tempo, along with high precision. This puts significant demands on
the coordination of fingers, hands and arms. The demands on precision, coordination
and concentration can generate repetitive muscle activity in the shoulders, which is
exacerbated by whole-body vibrations in the cab while working (Attebrant et al., 1998;

In many cases, work in forestry machines leads to muscle injuries in the shoulders, neck,
back, wrist and forearms. Upward (flexion), downward (extension) and sideways
(deviation) movements, combined with a fixed working position, can enhance the risk of
injuries of long duration of function movements (Eklund & Cederqvist, 1998). In
addition to the joystick control work, the operator must consider a lot of other
information, putting great demands on perception and cognitive capacity. This adds to
the muscle tension of the operator (Attebrant et al., 1998).

Fatigue can both be central and muscular. Central or psychological fatigue may depend
on factors such as duration of work, demands on concentration, monotony, noise and
other environmental or health aspects. Muscular or physical fatigue arises when all
energy reserves are exhausted or, in some cases when there is accumulation of lactic
acid in the muscles. Recovery only occurs when the stress is decreased or stops. The
time required for effective recovery of the muscles depends on the nature of the pause. Only total relaxation provides all involved muscles an opportunity to recover (Ericsson & Odenrick, 1997). Normally, fatigue in the muscles is explained by the fact that the blood circulation in the muscles gradually decreases as muscle tension increases, since blood pressure is approximately equal in the surrounding muscles.

Physical fatigue can be both general and local. General physical fatigue arises during prolonged and heavy physical activity involving big muscle groups (such as manual forest work). Local physical activity causing muscle fatigue is a combination of pain and tiredness, which arises when a few small muscle groups are strained. Local physical activity is also affected by psychological factors such as motivation (Ericsson & Odenrick, 1997). Tiredness and pain will eventually lead to chronic trouble and diseases. As such, operator work should be designed to be varied, involving both light and heavy stresses in combination with suitably long and frequent pauses, as well as total relaxation.

Automation can improve performance as well as standardize work in situations where the operator is in control (Brander, Eriksson & Löfgren, 2004). This will happen if the control system handles the more complex and repetitive tasks, relieving operator stress (Bossé & Breton, 2002; Alm & Ohlsson, 2003). Above all, the advantages of automation would be improved efficiency and standardization of the operator’s work. Automation would provide a certain degree of stability when performing work tasks (Bossé & Breton, 2003). With the automation of functions for efficient and effective operator work, mental stress would become less strenuous and the risk of “human error” would be diminished (Bossé & Breton, 2003). The mental and physical energy freed up by automation could then be used for other tasks—such as planning the next stage of the work—without leading to stress or fatigue.

For tasks that cannot take advantage of full automation, semi-automation is of interest. Semi-automation means that certain sequences are made automatic, but that the control system still needs operator supervision. Boom tip control is an example of semi-automation, where by means of the joystick the operator performs sequential movements of the boom instead of having one joystick movement corresponding to just one movement of the boom. This simplified way of controlling the boom gives the operator several opportunities for short pauses (Löfgren & Nordén, 2003a).

Principles of boom control

The forestry machines of today are controlled by two joysticks of different design. By moving one joystick in one direction, the operator controls a specific hydraulic cylinder on the boom (see Figure 1). This means that the operator has to combine different joystick movements to move the tip of the boom in the desired direction. In other words, the operator solves the inverse kinematics problem.
Figure 1: Conventional control of forest knuckle boom.

The concept *boom tip control* (Löfgren, 2004, Löfgren & Wikander, 2008) means that the tip of the boom is controlled with only one joystick. Up/down on the joystick corresponds to up/down on the tip of the boom, out/in on the joystick corresponds to out/in on the tip of the boom, and left/right on the joystick corresponds to left/right on the tip of the boom (see Figure 2).

Figure 2: Boom tip control.

EXPERIMENT

Method

This study was conducted in a real forest machine simulator at Skogforsk, Uppsala, Sweden. The purpose of the study was to investigate how two types of boom control, conventional control and boom tip control, affect the learning capabilities of inexperienced operators.

Participants

The participants in the study were from a vocational school for natural resource use and were between 15 and 19 years of age. Altogether, sixteen students (fourteen male and two female) voluntarily participated in the study. None of the students had previously participated in any simulator or real forest machine studies.
The participants were randomly divided into two groups, the boom tip group and the conventional group. There was one female in each group.

Apparatus

The study was conducted in a real-time forest machine simulator (Löfgren, Ohlsson, Wikander, 2007). The simulator was a fixed-base simulator and consisted of a driver seat, and featured all of the buttons and joysticks found in the Valmet 860, a real forwarder. Three multi-synchronized projectors created a field of view of ± 120°. All of the simulated movements of the forest machine in the simulator were based on physical characteristics such as friction, moment of inertia and other dynamic properties of a real machine. This means that the characteristics of the simulator were very much consistent with those of a real forest machine (Löfgren, Ohlsson & Wikander, 2007). The visual feedback of the simulator was presented on three screens, each of which were 2.0 x 1.5 meters, and with a resolution of 1400 x 1050 pixels (see Figure 3).

Figure 3: The forest machine real time simulator.

Experimental design

The study included general exercises and specific evaluating tests. The focus of the exercises was on the smooth and precise handling of the boom, as well as learning all of the joystick functions. The exercises primarily emulated real forwarder operations and were completely different from the evaluating tests. The evaluating test represented a specifically designed test motion path (see Figure 4) on a gravel court with cones, targets, wooden pallets and a wall of concrete. The objects had deliberately been chosen and dimensioned to look like real objects that would exist in a forest. The evaluating test was designed to measure the time consumption and precision of the boom handling.
Logs are handled in real forwarder work but to study only the boom control, the specific aspects of log handling were avoided in the simulator test. Therefore, the ordinary grapple at the tip of the boom was replaced by a rectangular parallelepiped, which was designed to have a certain volume and weight suitable for the test. When the operator placed the rectangular parallelepiped at the respective targets, he/she pressed a confirm button to trigger a measurement of precision.

The joysticks were adjusted and checked to ensure that they provided the same actuator speed in all directions, as would be the case in a real forwarder. All the participants used the same adjustments. The grapple function was not used, since the scenario did not involve tree handling.

**Measured variables**

Four measures were defined to describe potential benefits, and one measure was defined to describe the effect on the operator by comparing boom tip control to conventional control. The potential benefits were measured by the specially-defined measures of performance, time consumption, weighted points and points.

- The *time consumption* is the total time used without penalty for errors.

- The *performance* is the total time plus so called *weighted points* that represent weighted time penalties for errors (collisions and deviations over certain thresholds from nominal paths) and inaccuracies with respect to positioning at the five targets defined in Figure 4.
The mental load was evaluated using the NASA Task Load Index (NASA-TLX), which provides a measure of the experienced mental load through the estimation and weighting of six classes of data: mental load, physical load, time-dependent demand, own performance, effort, and frustration.

**Procedure**

The participants were briefed that they would take part in a total of 14 hours worth of practices and tests over a period of nine weeks. In total, each participant took part in seven practices and tests. During the practice sessions, the boom was to be controlled as it would be during normal work with a forwarder boom. The practices were focused on softness and exactness of the boom control, such that the participants were able to learn the functions and functionality of the boom.

After each practice, a test was carried out and repeated five times. After each test the participants were given the chance to take a short, two-minute break. At each test the participant was told that precision and caution were very important. All participants were told that each test was equally important.

At any time, the participants could be informed of their own results, but not those of the other participants. Each test was carried out with a test leader.

**RESULTS**

First, the *Performance* is evaluated. The boom tip group used on average $4.0 \pm 0.28$ minutes; the conventional group used $4.8 \pm 0.34$ minutes. Thus, the boom tip group
performed anywhere between 3.6% and 29% faster than the conventional group, and 16% faster on average.

![Performance graph](image)

**Figure 5: Mean value of the performance, with regression lines, for each group.**

The derivative of the regression line demonstrates superior performance by the boom tip group in terms of time spent on the task—indicating a steeper learning curve for this group.

The *Time consumption* shows the total time used without penalties. The boom tip group used on average $3.1 \pm 0.29$ minutes; the conventional group used $3.3 \pm 0.28$ minutes. Thus, the boom tip group performed between 10% slower and up to 29% faster than the conventional group, and 5% faster on average.
The derivative of the regression line demonstrates superior time efficiency by the boom tip group—again indicating a steeper learning curve for this group.

Also the Weighted points measure alone shows better results for the boom tip group. The boom tip group received on average 0.8 ± 0.17 minutes; the conventional group received 1.5 ± 0.22 minutes. Thus, the boom tip group performed between 19% and 60% better than the conventional group, and 40% better on average. While the boom tip group was consistently lower and more stable during the tests, the conventional group was to some extent closing the gap by the end of the tests.

The points measure provides information on the participant’s precision work. The boom tip received on average 6.6 ± 0.87 points; the conventional group received 10.3 ± 1.49 points. The boom tip group performed between 15% and 52% better than the conventional group, and 33% better on average at avoiding errors.
The derivate of the regression lines is steeper for the conventional group—indicating that this group reduced the number of points better than the boom tip group.

The X-2 test on time consumption between the boom tip group and the conventional group revealed a statistically significant effect. The regression analysis showed a more rapidly declining trend for the boom tip group. The X-2 test on weighted points shows statistically significant results. The boom tip group was much better at avoiding errors compared to the conventional group. The X-2 test on performance is statistically significant, indicating that the boom tip group outperformed the conventional group.

*NASA-TLX*

Figure 9 shows each group’s rated, unweighted mean value, with regression lines, of the mental stress (according tom NASA-TLX) as experienced during each test. Notice the covariance between the two curves despite the fact that the evaluations were made independent of each person and at different occasions. This is due to experienced performance in the simulator. The regression lines show that mental stress decreased during the trials. The decrease in mental stress was larger for the boom tip group than for the conventional group.
DISCUSSION

The objective of this study was to investigate the difference between a simplified control and conventional control of a knuckle boom on a forwarder. The conducted experiment addresses several aspects of this objective. The validity of the simulator is confirmed by other simulator and field studies.

The performance, time consumption and number of errors indicate that the curves for each group converge over time, while mental stress diverges over time, in favor of boom tip control. While it is likely that both control approaches would give approximately the same performance over the long run, operators learn much faster and are less affected by stress using boom tip control.

CONCLUSION

The present study shows that boom tip control is an easier system to learn compared to conventional control. This can lead to savings in production due to shorter learning times, as operators will reach full production sooner. With boom tip control operators will reach lower levels of errors sooner, including the relative number of serious errors. Boom tip control does not enable more efficient control with respect to time, but the improvements come faster. Boom tip control generates less mental workload than conventional control, which in the long run would reduce the mental workload on the operator of forest machines such as a forwarder.

REFERENCES


Paper D

KINEMATIC CONTROL OF REDUNDANT KNUCKLE BOOMS WITH PATH TRACKING UNDER GEOMETRIC, VELOCITY AND ACCELERATION CONSTRAINTS

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ABSTRACT

The Swedish forestry industry competes in the global marketplace and since its raw material is more expensive than in other parts of the world, the supply chain from stump to industry must be very effective. One part of this supply chain is cutting down and transporting trees out from the forest to the landing area for further transportation via truck to the paper mill or sawmill. Forestry machines equipped with booms are used to cut down, handle and transport the trees. If we can reduce boom handling time and thereby increase the productivity by 10%, the Swedish forestry industry can increase their earnings by up to SEK250 million annually.

In this paper a maximum lifting capacity algorithm developed in earlier work was further extended to ensure TCP path-tracking possibilities for forest machine knuckle booms. The path-tracking takes into account both joint velocity and acceleration limits. The maximum lifting capacity algorithm resolves the kinematic redundancy of the boom and the path tracking PID controller enables path following of typical candidate paths for semi-automatic boom functions. The joint references are derived based on kinematics only, and the boom behavior is analyzed and verified using a commercial, but for the purpose extended, dynamic forest machine simulator.

The proposed algorithms and controller allow straightforward implementation on any forest knuckle boom, and we conclude that the path tracking controller works properly given the simulator model of the crane. When comparing simulations with and without load in the TCP, remarkable crane stiffness is noticed. Since the simulator models are proprietary it has not yet been possible to analyze the reasons behind this remarkable stiffness which intuitively seems too high.

KEYWORDS

Hydraulic manipulator, redundancy, kinematic control, local optimization, knuckle boom, forest machine, forwarder, boom tip control, joystick control, simulations.

INTRODUCTION

At a time when forest operations have now been fully mechanized, we are entering a new era in which automation will become increasingly important. This process will involve the automation of entire operations, including knuckle boom manipulation and also other parts of the process, such as the collection, transmission and reporting of information.

The forestry machines of today are high-tech units with advanced control engineering, and technological developments have resulted in a radical increase in performance. For the operator this has meant increased workload and fewer natural stops during typical work. By using more automated functions and letting the machine itself take care of repetitive work, the operator would be able to devote his or her time to decisions regarding tree selection, wood quality and environmental concerns.
There is potential for simplifying the manipulator control, partly to reduce mental workload and achieve a positive effect on more important tasks, and partly to increase productivity. The risk of stress injuries can also be reduced. The manipulator control can be simplified through the introduction of manipulator tip control and the automated control of certain manipulator movements.

Controlling a forestry machine today involves almost continuous precision work with the hands. A high level of production requires intensive precision work. This repetitive and intense work, however, causes statically tensed muscles and muscle fibers. In a forwarder application, the operator uses the crane about 50% of the total work time, loads and unloads an average of a grapple with logs every 30 seconds, and handles about 2000 trees per day.

Actions that increase blood circulation through the muscles are of crucial importance to minimizing the risk of stress injuries. As such, the work involved in controlling the manipulator should be as dynamic as possible, which calls for many short pauses. Decreased demands for precise joystick work would also generate less muscle tension and thereby reduce stress. Technical solutions that provide short pauses during intensive joystick work are therefore positive [6, 7].

Operators of forestry machines are exposed to substantial mental stress during their work. They must receive and process a large amount of information and make decisions under significant time pressures. At the same time, the precision work with the hands creates a relatively large amount of stress on brain activity [7].

A natural conclusion, therefore, is that a simplified manipulator control is likely to reduce mental stress on the operator.

Forestry machines of today are controlled by two joysticks of different design. By moving one joystick in one direction, the operator controls a specific hydraulic actuator (cylinder) on the boom. As such, the operator must combine different joystick movements to move the tip of the boom in the desired direction.

The concept **boom tip control** [13, 14 and 15] means that the tip of the boom is controlled with only one joystick. Up/down on the joystick corresponds to up/down on the tip of the boom, out/in on the joystick corresponds to out/in on the tip of the boom, and left/right corresponds to left/right on the tip of the boom. A computer-based kinematic control strategy transforms the joystick’s output signals (in our case in a cylindrical coordinate system) into hydraulic cylinder control signals.

After a review of research conducted up to this date, one can see that there has been increased interest in redundant manipulators since the beginning of the 1980s, and that there remains considerable academic interest in problems of kinematic redundancy. In industry, it has been necessary to introduce one or more redundant degrees of freedom to solve complicated applications; this redundancy involves considerably more complicated control algorithms than those used for non-redundant manipulators.
One method used in path planning for redundant manipulators is based on local optimization and the use of the quite popular pseudo inverse solution combined with inequality constraints to avoid joint limits, singularities or collisions with obstacles that may be present in the workspace. This approach is often referred to as the “Extended Jacobian Technique” [2, 3, 4, 5, 8, 12 and 17].

Other methods used for solving path planning of redundant manipulators are the Pontryagin maximum principle [9], B-splines with limitations on joint speed, joint acceleration and joint jerk [11], the Lyapunov stability theory [10], soft motion planning [1], a time-scaling method [16], genetic algorithm [17, 19] or a time-optimal path planning [18].

In this paper, we present an online kinematic algorithm for redundant manipulators, which takes into account joint velocity and acceleration constraints while keeping the desired path and maximizing lifting capacity. The paper is organized as follows. The maximum lifting capacity algorithm [13, 14 and 15] is presented first. Next, we present measured data on joint velocities, joint accelerations and boom tip speed for a redundant knuckle boom on a forwarder. Based on measured data, we subsequently describe different representative boom tip paths. Finally, the proposed algorithm is analyzed in a dynamic forest machine simulator.

**KINEMATIC CONTROL FOR MAXIMUM LIFTING CAPACITY**

In general, we want to control the tool center point (TCP) of the redundant manipulator along a prescribed path. We also chose to control the redundant degrees of freedom to maximize lifting capacity. Later in the study (and based on studies of real forwarder operations), a representative TCP path is defined and the kinematic control solution is evaluated by simulations in a dynamic real-time forest machine simulator.

The algorithm for maximum lifting capacity is presented in [13, 14 and 15] and the kinematic control strategy is based on an optimization study of the lifting capacity [14], as a function of the three degrees of freedom \( \theta_1, \theta_2, \) and \( d_3 \), based on the force or torque characteristics, the geometrical arrangements of the cylinders, and how lifting capacity is dependent on the prismatic function \( d_3 \). See Figure 1 for notations and note that \( d_1 \) and \( d_2 \) are fixed, whereas \( d_3 \) is variable.

The studied manipulator is velocity-controlled and operating in a cylindrical \((r, \theta, z)\) coordinate system. Since we are working in this system, the control of the \( \theta_0 \)-joint—which controls rotation around the z-axis—is separated from the control of the other joints.
Figure 1: The manipulator geometry in the r-z plane.

A vector $\vec{\theta}$ is defined:

$$\vec{\theta} = [\theta_1, \theta_2, d_3]^T$$

The TCP (see Figure 1) coordinates $r$ and $z$ defines the vector $\vec{x}$:

$$\vec{x} = [r, z]^T$$

Figure 1 gives the following relations:

$$r = d_1 s_1 + (d_2 + d_3) s_{12} - ac_{12}$$

$$z = d_1 c_1 + (d_2 + d_3) c_{12} + as_{12}$$

where $s_{12} = \sin(\theta_1 + \theta_2)$, $c_{12} = \cos(\theta_1 + \theta_2)$.

Differentiating $r$ and $z$ gives:

$$\dot{r} = d_1 c_1 \dot{\theta}_1 + (d_2 + d_3) c_{12} (\dot{\theta}_1 + \dot{\theta}_2) + s_{12} \dot{d}_3 + as_{12} (\ddot{\theta}_1 + \ddot{\theta}_2)$$

$$\dot{z} = -d_1 s_1 \dot{\theta}_1 - (d_2 + d_3) s_{12} (\dot{\theta}_1 + \dot{\theta}_2) + c_{12} \dot{d}_3 + ac_{12} (\ddot{\theta}_1 + \ddot{\theta}_2)$$

Equation (2) can now be written:

$$\vec{x} = J \vec{\dot{\theta}}$$

where the Jacobian $J$ (2x3 matrix) has elements $j_{11}, j_{12}, \ldots, j_{23}$.
Since the Jacobian $J$ is non-square, the matrix cannot be inverted and hence the kinematic control $\dot{\theta} = J^{-1} \dot{x}$ cannot be directly derived. This problem can be solved by introducing a constraint.

When working with heavy loads, the extra degree of freedom can be used for maximizing lifting capacity. The kinematic control algorithm is based on computer studies of lifting capacity [14] as a function of $\theta_1$, $\theta_2$ and $d_3$, using hydraulic cylinder characteristics and the geometrical arrangements of the hydraulic cylinders. From the studies it is possible to analyze how lifting capacity is dependent on the prismatic function $d_3$. The proposed kinematic control function of $d_3$ is shown in Figure 2.

![Figure 2: $d_3$ as a function of TCP's position. The function $d_3(\rho_i)$ in between $\rho_{i\min}$ and $\rho_{i\max}$ is defined below.](image)

When $d_3 = d_{3\min}$ or when $d_3 = d_{3\max}$ could be compared to two circles with different radii and origins (see Figure 2). The working area is divided into two zones of equal
width and the points A, B, C, and D are on the same straight line at the height $z_c$. Points A to D have the following coordinates:

A: $(r_i, z_c)$

B: $(r_2, z_c)$

C: $(r_i + \rho_{i\text{min}}, z_c)$ for $i = 1, 2$

D: $(r_i + \rho_{i\text{max}}, z_c)$ for $i = 1, 2$

Figure 2 gives:

$$\rho_i = \sqrt{(r - r_i)^2 + (z - z_c)^2} \quad i = 1, 2$$

(9)

where $r_i$ and $z_c$ are the coordinates for the centre point of the circles in the proposed kinematic control function.

In order to avoid unnecessarily large accelerations in $d_3$, some “smoothing” of the $d_3(r, z)$ function may be introduced where the effect of $d_3$ on the lifting capacity is small. We therefore propose a kinematic control function of $d_3$ shown in figure 3.

![Figure 3: “Soft motion” algorithm of function $d_3$.](image)

$$d_3 = \frac{d_{3\text{max}}}{2} \left[ 1 + \frac{3(\rho_i - p_i)_{\text{max}}}{2q_i} - \frac{(\rho_i - p_i)_{\text{max}}^3}{2q_i^3} \right]$$

(10)
with
\[ p_i = \frac{\rho_{i_{\text{max}}} + \rho_{i_{\text{min}}}}{2} \]  
(11)
\[ q_i = \frac{\rho_{i_{\text{max}}} - \rho_{i_{\text{min}}}}{2} \]  
(12)

Differentiating equation (10) gives
\[ \dot{d}_3 = f_i(r, \rho_i) + g_i(z, \rho_i) \dot{z} \]  
(13)

With
\[ f_i(r, \rho_i) = c_i (r - r_c) / \rho_i \quad \text{for} \quad \rho_{\text{min}} \leq \rho_i \leq \rho_{\text{max}} \]
\[ g_i(z, \rho_i) = c_i (z - z_c) / \rho_i \quad \rho_{\text{min}} \leq \rho_i \leq \rho_{\text{max}} \]
\[ f_i(r, \rho_i) = 0 \quad \rho_i < \rho_{\text{min}}, \rho_i > \rho_{\text{max}} \]  
(14)

where
\[ c_i = \frac{3d_{\text{max}}}{4q_i} \left( \frac{1}{2q_i^2} \right) \]  
(15)

We simplify our notations:
\[ f_i(r, \rho_i) = f, \quad g_i(z, \rho_i) = g \]  
(16)

Using \( \dot{d}_3 \) from equation (13) in equation (7) gives:
\[ \begin{bmatrix} \dot{r} \\ \dot{\theta}_1 \\ \dot{\theta}_2 \end{bmatrix} = \begin{bmatrix} j_{11} & j_{12} & j_{13} \\ j_{21} & j_{22} & j_{23} \end{bmatrix} \begin{bmatrix} \dot{\theta}_1 \\ \dot{\theta}_2 \end{bmatrix} + \begin{bmatrix} \dot{r} \\ g \dot{z} \end{bmatrix} \]  
(17)

Rewriting equation (17) gives
\[ \begin{bmatrix} \dot{\theta}_1 \\ \dot{\theta}_2 \end{bmatrix} = \begin{bmatrix} j_{11} & j_{12} \\ j_{21} & j_{22} \end{bmatrix}^{-1} \begin{bmatrix} 1 - j_{13} f & - j_{13} g \\ - j_{23} f & 1 - j_{23} g \end{bmatrix} \begin{bmatrix} \dot{r} \\ \dot{\theta}_2 \end{bmatrix} \]  
(18)

Equations (13), (18) and (3) give, after some calculations, the kinematic control law:
\[ \dot{\theta} = P \dot{\theta} \]  
(19)

where the (3x2) P matrix has the following elements:
Velocity and acceleration limitations

The joints have maximum velocities \( \dot{\theta}_{1\text{max}}, \dot{\theta}_{2\text{max}}, \dot{\theta}_{3\text{max}} \) and maximum accelerations \( \ddot{\theta}_{1\text{max}}, \ddot{\theta}_{2\text{max}}, \ddot{\theta}_{3\text{max}} \). In a practical case, the geometrical arrangement of the \( \theta_1 \) – and \( \theta_2 \) – cylinders may cause \( \dot{\theta}_{1\text{max}} \) and \( \dot{\theta}_{2\text{max}} \) to be functions of \( \theta_1 \) and \( \theta_2 \), respectively, and directions. In this case, we make a simplification such that joint speed limits are set as equal in both directions. This is motivated by the fact that for each machine application hydraulic pressure and flow are different. The joint speeds are therefore set to suit one typical pressure and flow.

We wish to either keep the TCP speed constant or to have it follow a specified speed profile, \( v_s \), which varies along the path. The kinematic control law equation (19) is calculated at equidistant points along the path.

During one sample time \( t(n) \) the TCP moves the distance \( \Delta s \)

\[
\Delta s = \sqrt{\Delta r^2 + \Delta z^2}
\]  

(22)

The time \( t(n) \) for a sample is

\[
t(n) = \frac{\Delta s}{v_s(n)}
\]  

(23)

and varies when \( v_s(n) \) varies. In order to avoid acceleration limitations, the specified velocity has a positive ramp at the beginning of the path and a negative ramp at the end of the path.

Increments in position and velocity for \( \theta_1 \) are:

\[
\Delta \theta_1(n) = \dot{\theta}_1(n)t(n) + \ddot{\theta}_1(n)t(n)^2 / 2
\]  

(24)
\[ \Delta \dot{\theta}_1(n) = \dot{\theta}_1(n)t(n) \]  

(25)

Similar equations are valid for \( \theta_2 \) and \( d_3 \).

If any of the accelerations exceed the maximum value, the sample time must be prolonged. Assume that \( \theta_1 \) has the relatively highest “acceleration ratio” \( \dot{\theta}_1 / \dot{\theta}_{1\text{max}} \), giving the new sample time \( t_a(n) \) according to equation (24) with \( \dot{\theta}_1(n) \) replaced by \( \dot{\theta}_{1\text{max}} \)

\[ t_a(n) = \frac{\dot{\theta}_1(n)}{\dot{\theta}_{1\text{max}}}[\sqrt{1 + 2\dot{\theta}_{1\text{max}}\Delta \theta_1(n)/\dot{\theta}_1^2} - 1] \]  

(26)

With this new sampling time we get new velocities and accelerations from equations (24) to (25).

Since the sampling time is increased, the accelerations are reduced.

If one of the velocities exceeds its maximum value, for example \( \dot{\theta}_1 \),

\[ \Delta \dot{\theta}_1(n) = \dot{\theta}_{1\text{max}} - \dot{\theta}_1(n) \]  

(27)

We replace \( t(n) \) by \( t_v(n) \) in equations (24) to (25):

\[ t_v = 2\Delta \theta_1(n)/(\dot{\theta}_{1\text{max}} + \dot{\theta}_1(n)) \]  

(28)

With this new sampling time we get new velocities and accelerations.

The same procedure is applied if \( \theta_2 \) or \( d_3 \) reaches velocity or acceleration limits.

**Mechanical limitations**

Although we lose the redundant degree of freedom if one of the \( \theta_1 \), \( \theta_2 \), or \( d_3 \) - cylinders reaches a mechanical limit, we can still follow a desired path (until a second joint reaches a mechanical limit). To handle these mechanical limitations in the control law \( \dot{\theta}_1 \) is first calculated by means of equation (19). If the \( \theta_1 \) - cylinder is at a mechanical limit and equation (19) shows that the cylinder should pass through the mechanical limit, then a second calculation has to be performed exactly as in the previous section but now with \( \dot{\theta}_{1\text{max}} \) replaced by 0. Similar calculations are made for \( \theta_2 \) and \( d_3 \) when they reach their mechanical limits.

In order to avoid large transients when approaching a mechanical limit, the maximum velocity used for the calculations is decreased for the corresponding joint.
DEFINING REPRESENTATIVE TCP PATHS

To find out how the operator works with the boom during ordinary work on a real forwarder with conventional control, we measured data from such a boom. The angular and linear positions of all the joints were collected. The boom was prepared with angular sensors on each of the joints and data from the sensors were collected with a computer. The sampling rate was 1000 Hz. Certain characteristics were specifically analyzed, particularly the boom tip position in the r-z plane.

The motion patterns depicted in Figures 4 and 5 were achieved.

The notion *boom out* is used to indicate when the operator moves the boom tip from the load space out to pick up logs on the ground.

The notion *boom in* is used to indicate when the operator moves the boom tip from the ground up into the load space with logs in the grapple.

![Figure 4: The position of the boom tip in polar coordinates for *boom out* motion.](image)

![Figure 5: The position of the boom tip in polar coordinates for *boom in* motion.](image)
In Figures 4 and 5 one can see how $z$ varies as a function of $r$, giving an idea of how to describe a representative path for *boom out* and for *boom in*.

**Path planning**

Based on Figures 4 and 5, as well as studies of several forwarder operators, the decision was made to use the path described in Figure 6.

![Figure 6: Representative TCP path of forwarder operation](image)

We have: $r_0, z_0, r_1, z_1, r_2, z_2, R, \alpha_0$

A→B: $\Delta r = 0, \Delta z = \Delta s$  

(29)

B→C: $\Delta r = \Delta s \sin \alpha, \Delta z = \Delta s \cos \alpha, \Delta \alpha = \frac{\Delta s}{R}, 0 \leq \alpha \leq \alpha_0$  

(30)

$$\alpha_0 = \arcsin \left[ \frac{aR + b \sqrt{a^2 + b^2 - R^2}}{a^2 + b^2} \right] + \pi / 2$$

(31)

$$a = r_2 - r_1 - R$$

(32)

$$b = z_2 - z_1$$

(33)

C→D: $\Delta r = \Delta s \cos(\alpha - \pi / 2), \Delta z = -\Delta s \sin(\alpha_0 - \pi / 2)$

(34)

**SIMULATIONS**

The maximum lifting capacity algorithm applied to path-following has been tested through simulations in a kinematic simulation software developed especially for analysis of knuckle booms, and in a dynamic forest machine simulator. The simulations in the dynamic real-time forest machine simulator constitute a dynamic simulation where the
weight of each part of the boom, the moment of inertia, and friction effects in the joints are included. This is not the case for the kinematic simulation software, which only calculates the kinematics. The paths to be followed are similar to those an operator would perform when moving the boom from the loading space out to pick up logs on the ground and back again (as defined in Figure 6), but with different values for $r_0, r_1, r_2, r_3, R$ and $\Delta s$ used in equations (29) to (34) depending on whether the motion is boom out or boom in. Figures 7 and 8 show the paths for in and out motion as extracted from the dynamic forest machine simulator.

![Figure 7: Left: Screenshot from the dynamic real-time forest machine simulator of the paths performed, in this case the boom out motion. Right: Screenshot from the dynamic real-time forest machine simulator of the paths performed, in this case the boom in motion.](image)

There is a network connection between the kinematic simulation software and the dynamic forest machine simulator. In practice, this means that the calculations of the speeds for each joint are performed in the kinematic simulation software, with the results then sent to the dynamic real-time forest machine simulator for activation of the boom (Figure 8).

![Figure 8: The connection between the kinematic simulation software and the dynamic forest machine simulator.](image)

A special Graphical User Interface (GUI) was developed for the kinematic simulation software (see Figure 9).
Figure 9: The GUI menu of the kinematic simulation software.

We were able to do the following with the GUI:

- Evaluate three different algorithms: maximum lifting capacity, maximum speed and dynamic programming;
- Define any length of the boom parts, which enabled analysis of any type of knuckle boom;
- Define any start point in the working area of the boom;
- Define the limitations of:
  - minimum and maximum joint angle;
  - maximum joint speeds and accelerations;
- Set parameters for velocity ramps close to the end position of each joint;
- Generate a graphical presentation of the results;
- Start the automatic boom in and boom out functions;
- Remember the last position outside the load space of the forwarder;
- Automatically go to the last position outside the load space of the forwarder;
- Remember the last position in the load space of the forwarder;
• Automatically go to the last position in the load space of the forwarder; and
• Put the boom into transport position.

**PID velocity controller**

The dynamic forest machine simulator is in its commercial form run by a machine operator. Hence, for testing the path following algorithms a motion controller had to be implemented in the simulator. This was done by the simulator provider specifically for the purpose of this research. The implemented PID controller takes as input the velocity references from the kinematic simulation software as depicted in Figure 9 and controls the boom joints accordingly (see Figure 10).

\[ \dot{\theta}_0, \dot{\theta}_1, \dot{\theta}_2, \dot{d}_j \]

**Figure 10: PID-regulator in the dynamic real-time forest machine simulator.**

The PID controller was tuned according to the Ziegler Nichols method.

**Results**

The figures below demonstrate the results from the simulations from both the kinematic simulation software and the simulations in the dynamic real-time forest machine simulator. The simulations were performed with three different loads in the grapple: no load, 1000 kg and 2000 kg. The last figures show the position difference of the joints when we are using different loads.
Boom out motion

Figure 11: TCP moves along the path, boom out motion, in the kinematic simulation software with no load in the grapple. Variations in $r$ and $z$.

Figure 12: Motion references for $\dot{\theta}_1$, $\dot{\theta}_2$, and $\ddot{d}_3$ as given by the kinematic simulation software.

Figure 13: Motion responses in $\dot{\theta}_1$, $\dot{\theta}_2$, and $\ddot{d}_3$ from the dynamic simulator.

The three figures above show the TCP motion path and the corresponding joint motion references and responses for a case where joint $d_3$ starts (at 3.2 seconds) and $\dot{\theta}_1$ starts to...
decrease. At 3.7 seconds $\dot{\theta}_1$ is saturated and $\dot{\theta}_2$ and $\dot{d}_3$ decreases. At 3.9 seconds $\dot{\theta}_1$ starts to decrease and $\dot{\theta}_2$ decreases and $\dot{d}_3$ increases.

**Boom in motion**

![Figure 14: TCP moves along the path, boom in motion, in the kinematic simulation software with no load in the grapple. Variations in $r$ and $z$.](image)

![Figure 15: Motion references for $\dot{\theta}_1$, $\dot{\theta}_2$ and $\dot{d}_3$ as given by the kinematic simulation software.](image)

![Figure 16: Motion responses in $\dot{\theta}_1$, $\dot{\theta}_2$ and $\dot{d}_3$ from the dynamic simulator.](image)
The three figures above show the TCP motion path and the corresponding joint motion references and responses for a case where $\dot{\theta}_2$ increases (at 4.2 seconds) since $\dot{\theta}_1$ is in saturation and $\dot{d}_3$ decreases. At 5.1 seconds the joint $d_3$ reaches its minimum position and only joint $\theta_1$ and $\theta_2$ are activated. At 6 seconds $\dot{\theta}_1$ is in saturation. At 6.4 seconds $\dot{\theta}_1$ decreases and also $\dot{\theta}_2$ decreases.

**Path tracking errors with a load in the TCP**

Loads of 1000 kg and 2000 kg were added at the TCP in order to analyze the effect crane load on path following performance. During this analysis the chassis was “locked”, meaning that only the dynamic movements of the crane were included in the simulations.

![Figure 17: Differences in the TCP, the height position $z$ in relation to length position $r$, between the motion reference and the response for loads of 1000 kg and 2000 kg for boom out motion.](image)

From the two figures above it is obvious that the crane behaves remarkably stiff. The PID controller acts upon kinematically derived velocity references and hence does not
compensate for tracking errors in position. This means that the errors in positions according to the figures above are due to both any velocity tracking errors and any mechanical flexibility in the crane structure. One conclusion of this is that the PID velocity controller works properly given the simulator model of the crane. However, for further development of a path tracking controller it is likely that more complete crane models are needed. The exact modelling of the crane in the simulator is proprietary and has not been open to this research and hence it has not been possible to analyze the reasons behind the stiff crane behaviour in detail. This is a subject of further studies of importance for future use of the simulator as a research tool.

CONCLUSIONS

In this paper, a maximum lifting capacity algorithm has been described and utilized to ensure TCP path-tracking of a forest knuckle boom while taking into account both velocity and acceleration joint limits. The proposed algorithm allows straightforward implementation on any forest knuckle boom, and we conclude that the path tracking controller works properly given the simulator model of the crane. When comparing simulations with and without load in the TCP, remarkable crane stiffness is noticed. Since the simulator models are proprietary it has not yet been possible to analyze the reasons behind this remarkable stiffness which intuitively seems too high.

REFERENCES


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