TIP CONTROL OF A LOG CRANE WITH MOBILE HYDRAULIC PROPORTIONAL VALVES

Lappeenranta–Lahti University of Technology LUT

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ABSTRACT

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The main objective of this work is to develop a practical open-type control system for a log crane with mobile hydraulic proportional valves. A feature of the control system is the receipt of appropriate control signals for the hydraulic system to achieve a commanded velocity of a boom tip in Cartesian coordinates. Another feature of the developed control system is that it does not require any additional adjustments to the system than the pistons' position sensors. Two variants of the system were considered: with and without counterbalance valves. A mathematical model of the hydraulic system was developed in conjunction with a multibody model to test the designed control system. The limits of application of the implemented control method were formulated based on the analysis of the obtained results. Also, the determination of the directions for improving the system was performed.
SYMBOLS AND ABBREVIATIONS

Roman characters

\( a_1 \ldots a_5 \) coefficients of the counterbalance valve model [-]

\( A \) area \([m^2]\)

\( B_e \) effective bulk modulus \([Pa]\)

\( C_V \) valve flow constant \([m^3/(s\cdot V\cdot \sqrt{Pa})]\)

\( l_i \) kinematic length of a link \([m]\)

\( L_1 \ldots L_{13} \) length \([m]\)

\( p \) pressure \([Pa]\)

\( p_{\text{ref}} \) set pressure \([Pa]\)

\( Q \) volume flow rate \([m^3/s]\)

\( U \) voltage \([V]\)

\( V \) volume \([m^3]\)

\( v \) vector of Cartesian velocities \([m/s]\)

Greek characters

\( \varepsilon_1 \ldots \varepsilon_{22} \) log crane geometry angles \([\text{rad}]\)

\( \theta \) vector of boom’s joint angles \([\text{rad}]\)

Abbreviations

DCV Direction Control Valve

MBM Multi-body Model
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1 Introduction

The operation of many different types of machinery is based on the use of hydraulic booms. Traditionally, an operator controls each hydraulic cylinder of the boom separately. In the case of the considered crane, an operator has to simultaneously and manually control at least two valves.

In coordinate steering, also known as trajectory steering, the driver steers the tip of the boom directly instead of individual joints. For example, one direction of movement of the guide rod is the horizontal movement of the boom tip and the other direction of movement - is the vertical movement of the boom tip.

Mobile cranes have been on the market for harvesters in the forestry machinery sector for a long time. Coordinate control is implemented mechanically utilizing the crane’s articulated geometry. This approach increases the number and weight of the crane’s joints and their complexity.

In industry, the company John Deere presented (John Deere, 2021) a boom control for harvesters and forwarders in 2013 and called it Intelligent Boom Control (IBC). It points out several advantages of an advanced boom tip control:

1. Simplicity for an operator
2. Accuracy and steadiness of the boom movement throughout the entire operating range
3. Improvement of damping of the cylinders’ movement
4. Increase work process productivity with automatic movement of the harvesting head to the cutting height, and after that to the processing height
5. Automatic adjustment for working on a slope
1.1 Motivation of the research, objectives, and scopes

The purpose of this work is to create a control algorithm based on coordinate steering that could be implemented directly in a hydraulic log crane. The ground for studying this subject is the need to simplify the handling of a considered object for an operator with avoiding complicating the system.

The objectives of the master thesis are presented as follows:

- Creation of a mathematical model of the considered crane. The virtual prototype comprises a hydraulic system and a mechanical multibody system.
- Development of the control algorithm and its implementation in a program code. Performing control system tests in a virtual environment.
- Development of ways to improve the implemented hydraulic system for refinement of the system performance.

The implementation of the above-mentioned mathematical model of the considered object makes it possible to simulate a wide range of possible crane configurations, such as hydraulic system parts, types of loading, and control system parameters, including those that are impossible to perform in the laboratory due to the obstacles and limitations. It also allows performing tests beyond the limits conditions to check the system behavior in emergency situations.

A number of simplifications and limitations of the performed research were taken into consideration:

- Only lift and bending cylinders are taken into consideration, neglecting the basement rotation and an extension boom.
- Detailed modelling of each component of direction control valves with pressure compensation was narrowed down to simplified flow equations.
The effect of wave phenomena in the fluid, as well as hydraulic resistance of hoses and local restrictors, was not taken into account. It was possible due to the relatively low velocities of fluid flow in the considered hydraulic system;

- Parts of the system are rigid and changes caused by plastic deformation are not taken into account.

- Signals from the sensors assumed ideal, without possible disturbances and interferences. The relevant filters should be implemented in on-board applications, but in this work, it was not considered.

1.2 Known conducted research

Significant research in the field of coordinated control was done previously by scientists, and several works were performed at our university. Hence, I find it necessary to consider them before the project development.

Dmitrii Shevchuk (2020) created a mathematical model of a hydraulic crane comprising a load sensing pump, proportional control valves, and hydraulic cylinders. He implemented the developed model in MATLAB Simulink and performed verification of the hydraulic model through experiments on the test setup.

Chang and Lee (2000) studied closed-loop time-delayed coordinate control for a straight-line motion of a 13-ton hydraulic excavator. As a result, the effective end-effector velocity of 0.5 m/s was achieved with an accuracy of 3 cm. The feedback is carried out through the measurement of hydraulic pressure and the boom articulation angles.

Činkelj et al. (2010) investigated the application of coordinate guidance to a six-degree-of-freedom hydraulic telescopic handler with a payload of 2000 kg. The steering was implemented based on a closed-loop PI control algorithm. The feedback measurements were achieved utilizing a wire draw sensor for telescope lengths and resolvers for the joints'
rotation angle. The developed system resulted in the maximum error of 63 mm at 2.4 m in the straight-line tracking of the tool point.

Jussi Peltola (2014) in his work developed a practical coordinated control to be installed together with a traditional control system in a construction machine. The use of coordinated direction control valves with zero overlap and feedback-controlled spools in conjunction with inclinometers, unrealistic roller potentiometers, and pressure transmitters. As result, on a trajectory length along the x-coordinate of 2.3 m, the y-coordinate is kept within 30 mm.

Hamid Roozbahani (2020) developed and implemented an advanced control system for a material handling hydraulic manipulator. Two types of control are described in the article: open-loop and close-loop. In the former case, the simulation results with step-shaped input signals are an error of 2.08 m in positive x-coordinate with 1.75 m shift in y-coordinate, and an error of -4.15 m in negative x-coordinate with translocation -1.23 m in y-coordinate.

Jussi Manner (2017) studied the influence of boom tip control on the slopes of learning curves for beginner-level forwarder operators. According to his results, the implementation of boom tip control does not affect significantly the slopes of learning curves but improves crane work productivity and simplifies control for an operator.

Julia Malysheva (2021) studied the question of a faster than real-time simulation using the example of the PATU-655. She developed two simulation models of the system’s mechanical structure: one based on a computationally efficient dynamic topological formulation, and another one utilizing the software MATLAB Simulink with the package Simscape. The single model of the crane’s hydraulic system is used for both models and it is based on the lumped fluid power theory.
2 Log crane modelling

A mathematical model of the considered hydraulic log crane was developed following the hydraulic diagram and mechanical assembly. The photograph and the schematic are presented in figures 1 and 2.

Figure 1. The photograph of the log crane PATU 655
After that, the developed mathematical model was implemented in the software MATLAB Simulink utilizing the Simscape Multibody package.

2.1 Hydraulic system of the crane

The crane actuating system is an open-loop hydraulic system. The hydraulic scheme of the considered crane is presented in the figure 3.
The main components of the system are hydraulic cylinders 1 and 2 that are used to move crane booms, and proportional direction control valves 3 and 4 to control them. The power source of the system is a hydraulic pump station 5. Hydraulic hoses 6 and 7 are used to connect the hydraulic components to maintain system mobility.

2.1.1 Hydraulic system modelling

To describe the dynamic behavior of a hydraulic system a dynamic continuity equation is used. It can be described as follows:

$$\frac{dp}{dt} = \frac{B_e}{V} \cdot \left( \sum_{i=1}^{n} Q_{in_i} - \sum_{j=1}^{m} Q_{out_j} - \dot{V} \right)$$

(1)
where $\sum_{i=1}^{n} Q_{in_i}$ and $\sum_{j=1}^{m} Q_{out_j}$ are sums of all inbound and outbound flow [m$^3$/s], and $\dot{V}$ is a first-time derivative of a geometrical volume [m$^3$/s]. This equation was implemented for every volume between different components. In our case, changes in geometrical dimensions were neglected.

Effective bulk modulus was used to achieve a more realistic behaviour of the system. It takes into account non-soluble air mixed in oil, as well as the compressibility of the working fluid.

$$\frac{1}{B_e} = \frac{1}{B_c} + \frac{V_g}{V_t} \frac{1}{B_g}$$  \hspace{1cm} (2)

where $B_c$ is a bulk modulus of fluid [Pa], $V_g$ is a volume of gas [m$^3$] and $V_t$ is total volume [m$^3$].

2.1.2 The mathematical model of a proportional direction control valve

To control the hydraulic crane movement utilization of proportional direction control valves is necessary and widely spread in the industry. The pressure-compensated valves Danfoss PVG 32 are used in the considered hydraulic crane. Their detailed hydraulic configurations are presented in figures 4 and 5.
Figure 4. Hydraulic scheme of the inlet module PVP (Danfoss 2019, p. 25)

Figure 5. Hydraulic scheme of the compensated valve PVB (Danfoss 2019, p. 51)
Because of the valve configuration, it can behave in two main patterns. If the pressure difference between the pump and controlled ports is bigger than a set pressure, the flow is proportional to voltage and independent of the actual pressure difference. On the opposite side, if the pressure difference becomes smaller than a set pressure, the flow through the valve can be characterized as flow through an orifice.

A set of equations was used to describe different regimes of flow through the valve:

If \( U \leq U_l \)

\[
Q_A = \begin{cases} 
C_{v\text{ptoA}} \cdot (U_l - U) \cdot \sqrt{p_{\text{ref}}} & \text{if } p_p - p_A \geq p_{\text{ref}} \\
C_{v\text{ptoA}} \cdot (U_l - U) \cdot \sqrt{|p_p - p_A|} \cdot \text{sign}(p_p - p_A) & \text{if } p_p - p_A < p_{\text{ref}} 
\end{cases} 
\]

\[
Q_B = C_{v\text{ABtoT}} \cdot (U_l - U) \cdot \sqrt{|p_B - p_T|} 
\]

\[
Q_P = Q_A 
\]

If \( U \geq U_h \)

\[
Q_B = \begin{cases} 
C_{v\text{ptoB}} \cdot (U_l - U_h) \cdot \sqrt{p_{\text{ref}}} & \text{if } p_p - p_B \geq p_{\text{ref}} \\
C_{v\text{ptoB}} \cdot (U_l - U_h) \cdot \sqrt{|p_P - p_B|} \cdot \text{sign}(p_P - p_B) & \text{if } p_P - p_B < p_{\text{ref}} 
\end{cases} 
\]

\[
Q_A = C_{v\text{ABtoT}} \cdot (U_l - U_h) \cdot \sqrt{|p_A - p_T|} 
\]

\[
Q_P = Q_B 
\]

where indexes A, B, P, and T a represent flow parameters in load, pump, and tank lines respectively, \( U_l \) and \( U_h \) are valve opening voltage thresholds \([V]\).

2.1.3 The mathematical model of a hydraulic cylinder

Hydraulic cylinders are widely used in manipulators and cranes. To describe their movement and to compute the resultant forces following equations are used:
\[ F_R = p_A \cdot A_A - p_B \cdot A_B - F_f \]  \hspace{1cm} (5)

where \( F_R \) is the resultant force applied to the cylinder [N], indexes A and B describe system parameters related to piston and stroke sides of the piston, and \( F_f \) is friction force [N].

The calculated values of resultant cylinders’ forces are direct to the Simscape MBM, where the acceleration, velocity, and position of parts of the system are calculated. A position and velocity feedback are used for the performance of the cylinders’ mathematical models.

The friction force in the hydraulic cylinder is taken into account as an external influence. This aspect of the system can be described in various ways and usually, it describes only static dependence between velocity and resultant force and takes into account the Coulomb and Viscous friction components. Nevertheless, some of the observed phenomena cannot be described with such an approach, such as stick-slip motion, pre-sliding displacement, and friction lag. Therefore, an analytic model of friction forces proposed by LuGre was implemented:

\[
F_f = \sigma_0 \cdot z + \sigma_1 \cdot \frac{dz}{dt} + k_v \cdot \dot{h} \\
\frac{dz}{dt} = \dot{h} - \frac{\vert \dot{h} \vert}{g(\dot{h})} \cdot z \\
g(\dot{h}) = \frac{1}{\sigma_0} \left[ F_C + (F_S - F_C) \cdot e^{-\frac{\dot{h}}{\nu_S}} \right] \hspace{1cm} (6)
\]

where \( z \) is an internal state, \( g(\dot{h}) \) describes the steady-state phenomena of friction for motion with constant velocity, \( \nu_S \) is the Stribeck velocity [m/s], \( F_S \) is the static friction [N], \( F_C \) is the Coulomb friction [N], \( k_v \) is a coefficient of viscous friction [N/(m/s)], and \( \sigma_0 \) and \( \sigma_1 \) are represent stiffness and damping coefficients [de Wit, 1995].
2.2 Model parameters

The grounds for the system parameters are the PATU 655 log crane manufactured by Kesla Corporation. It represents a medium-size loader for agricultural machines with a maximum load capacity on a max length of 320 kg. The crane is installed in the Laboratory of Intelligent Machines of LUT University and powered by a hydraulic power station with constant pressure control. It allows modelling the pump line as a constant pressure source. The general parameters of the hydraulic system are listed in table 1.

Table 1. General parameters of the hydraulic system

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Physical dimension</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump pressure</td>
<td>bar</td>
<td>140</td>
</tr>
<tr>
<td>Tank pressure</td>
<td>bar</td>
<td>1</td>
</tr>
<tr>
<td>Oil bulk modulus</td>
<td>MPa</td>
<td>1500</td>
</tr>
</tbody>
</table>

The pressure-compensated DCVs are Danfoss PVG 32. They comprise a closed centre PVP 157B5111 inlet module which includes a pilot pressure reduction valve. Three pressure compensating modules are connected to the inlet module. The C-spool is used for manipulation of the lifting cylinder and the A-spool is used for the bending cylinder. The relevant parameters are listed in tables 2 and 3.
Two hydraulic cylinders are used to manipulate the log crane. Their parameters are shown in tables 4 and 5.
### Table 4. Parameters of the lifting cylinder

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Physical dimension</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston diameter</td>
<td>mm</td>
<td>100</td>
</tr>
<tr>
<td>Rod diameter</td>
<td>mm</td>
<td>56</td>
</tr>
<tr>
<td>Range</td>
<td>mm</td>
<td>0...535</td>
</tr>
<tr>
<td>Non-soluble air volume</td>
<td>cm³</td>
<td>25</td>
</tr>
<tr>
<td>The inner diameter of the hoses</td>
<td>mm</td>
<td>12.7</td>
</tr>
<tr>
<td>Hose length</td>
<td>m</td>
<td>3</td>
</tr>
</tbody>
</table>

### Table 5. Parameters of the bending cylinder

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Physical dimension</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston diameter</td>
<td>mm</td>
<td>90</td>
</tr>
<tr>
<td>Rod diameter</td>
<td>mm</td>
<td>56</td>
</tr>
<tr>
<td>Range</td>
<td>mm</td>
<td>0...780</td>
</tr>
<tr>
<td>Non-soluble air volume</td>
<td>cm³</td>
<td>10</td>
</tr>
<tr>
<td>The inner diameter of the hoses</td>
<td>mm</td>
<td>12.7</td>
</tr>
<tr>
<td>Hose length</td>
<td>m</td>
<td>2</td>
</tr>
</tbody>
</table>
3 The control algorithm

In general, a control system for an object can be developed in two ways, known as open-loop control and closed-loop control. In a former case, the control system does not take external disturbances into account and a user expects the system to perform according to the normal behavior. In the latter case, the system configuration collects information related to the system state and the control system accordingly changes the control command. It is also characterized by instability and raises demand for higher bandwidth valves with zero overlaps to achieve the desired performance. Therefore, the open-loop control system with mobile hydraulic valves is selected.

3.1 Coordinate control algorithm

So far, control systems of industrial machines with hydraulic booms utilize an open-loop control system with feedback through the operator’s senses. Also, control signals into the object are provided through a proportional connection between the operator’s joysticks and direction control valves. Therefore, the accuracy of control significantly depends on the operator’s skill and experience.

A coordinate control algorithm was implemented to improve the control system accuracy and to make it more intuitive for operators. The structural diagram of the algorithm is presented in figure 6.
In general, the purpose of the algorithm is to change the input control signals into more native and intuitive ones. This goal is achieved through producing control voltage for the DCVs in the way the boom tip will move strictly in a determined direction with the required velocity.

3.1.1 Boom velocity kinematics

Boom kinematics is utilized to transform the input boom tip velocity commands into the required angle velocities of the boom joints. The Denavit-Hartenberg method was implemented to formulate dependencies between the angular velocities and the linear velocities of the end-effector. [R.N. Jazar, 2010]
To create a relationship between the joint velocities of the boom joints and the boom tip Cartesian velocity the following equation is used:

\[ v_0 = J_0(\theta) \cdot \dot{\theta} \]  \hspace{1cm} (7)

The observed hydraulic log crane represents a two-link arm, so a 2 x 2 Jacobian matrix can describe the transformation from joint rates to the end-effector velocities in terms of the base frame:

\[
J_0(\theta) = \begin{bmatrix}
-l_1 \cdot \sin(\theta_1) - l_2 \cdot \sin(\theta_1 + \theta_2) & -l_2 \cdot \sin(\theta_1 + \theta_2) \\
-l_1 \cdot \cos(\theta_1) + l_2 \cdot \cos(\theta_1 + \theta_2) & l_2 \cdot \cos(\theta_1 + \theta_2)
\end{bmatrix}
\]  \hspace{1cm} (8)

For this work, the calculation of the joint rates has the most importance. Hence, the use of the inverted Jacobian matrix is necessary. To obtain the relevant equation, the Jacobian matrix in equation 7 should be inverted:

\[ \dot{\theta} = J^{-1}(\theta) \cdot v \]  \hspace{1cm} (9)

3.1.2 Log crane kinematics

As can be seen from equation 8, the current value of the joint angles is required for the algorithm to perform. At the same time, the system comprises only pistons’ position sensors. Therefore, it is necessary to formulate the relation between the pistons’ positions and the joint angles. The relevant equations are obtained from the crane geometry [Peltola, 2014].
The relationship between the lift cylinder’s piston position $h_1$ and the first joint’s angle $\theta_1$ can be written as follows:

$$\varepsilon_3 = \cos\left(\frac{L_1^2 + L_2^2 - h_1^2}{2 \cdot L_1 \cdot L_2}\right)$$

$$\theta_1 = \varepsilon_3 - \varepsilon_2 + \varepsilon_1$$

(10)
The relationship between the bending cylinder’s piston position $h_2$ and the second joint’s angle $\theta_2$ is formulated as follows:

$$\epsilon_{16} = \arccos \left( \frac{L_4^2 + L_5^2 - h_2^2}{2 \cdot L_4 \cdot L_5} \right)$$

$$\epsilon_{20} = \pi - \epsilon_7 - \epsilon_8 - \epsilon_{16}$$

$$L_{10} = \sqrt{L_5^2 + L_8^2 - 2 \cdot L_5 \cdot L_8 \cdot \cos (\epsilon_{20})}$$

$$\epsilon_{21} = \arccos \left( \frac{L_8^2 + L_{10}^2 - L_5^2}{2 \cdot L_8 \cdot L_{10}} \right)$$

$$\epsilon_{22} = \arccos \left( \frac{L_7^2 + L_{10}^2 - L_6^2}{2 \cdot L_7 \cdot L_{10}} \right)$$

$$\theta_2 = \pi - \epsilon_6 + \epsilon_7 - \epsilon_{21} - \epsilon_{22}$$
3.1.3 Calculation of cylinder velocities

The complicity of the crane geometry requires proper formulation of relationships between the cylinders’ velocities and the joints’ angular velocities. An analysis of the crane geometry was promoted to obtain the relevant equations. [Peltola, 2014].

Taking into account equation 10 and figure 7, the relation between the lifting cylinder velocity \( \dot{h}_1 \) and the first joint angular velocity \( \dot{\theta}_1 \) can be described as follows:

\[
\begin{align*}
\varepsilon_4 &= \sin \left( \frac{L_2 \cdot \sin (\varepsilon_3)}{h_1} \right) \\
\varepsilon_5 &= \pi - \varepsilon_4 - \varepsilon_3 \\
L_3 &= L_2 \cdot \sin (\varepsilon_5) \\
\dot{h}_1 &= L_3 \cdot \dot{\theta}_1
\end{align*}
\]

A set of equations is used to connect the bending cylinder velocity \( \dot{h}_2 \) and the second joint angular velocity \( \dot{\theta}_2 \).
Figure 9. The geometry of the transitional boom [Peltola, 2014]

Figure 10. Geometry of the joint between transitional and final booms [Peltola, 2014]

\[ \varepsilon_9 = \pi + \theta_2 \] (13)
\[ \varepsilon_{10} = 2 \cdot \pi - \varepsilon_9 - \varepsilon_6 + \varepsilon_7 \]

\[ L_0 = \sqrt{L_7^2 + L_8^2 - 2 \cdot L_7 \cdot L_8 \cdot \cos(\varepsilon_{10})} \]

\[ \varepsilon_{11} = \text{asin} \left( \frac{L_7 \cdot \sin(\varepsilon_{10})}{L_0} \right) \]

\[ \varepsilon_{12} = \text{asin} \left( \frac{L_8 \cdot \sin(\varepsilon_{10})}{L_0} \right) \]

\[ \varepsilon_{13} = \text{acos} \left( \frac{L_9^2 + L_6^2 - L_0^2}{2 \cdot L_6 \cdot L_6} \right) \]

\[ \varepsilon_{14} = \text{asin} \left( \frac{L_6 \cdot \sin(\varepsilon_{13})}{L_0} \right) \]

\[ \varepsilon_{15} = \text{asin} \left( \frac{L_5 \cdot \sin(\varepsilon_{14})}{L_6} \right) \]

\[ \varepsilon_{17} = \text{asin} \left( \frac{L_5 \cdot \sin(\varepsilon_{16})}{h_2} \right) \]

\[ \varepsilon_{18} = \varepsilon_{12} + \varepsilon_{15} \]

\[ L_{10} = \sqrt{L_6^2 + L_7^2 - 2 \cdot L_6 \cdot L_7 \cdot \cos(\varepsilon_{10})} \]

\[ \varepsilon_{19} = \text{asin} \left( \frac{L_7 \cdot \sin(\varepsilon_{18})}{L_{10}} \right) \]

\[ L_{11} = L_{10} \cdot \sin(\varepsilon_{19}) \]

\[ L_{12} = L_9 \cdot \sin(\varepsilon_{15}) \]

\[ L_{13} = L_4 \cdot \sin(\varepsilon_{17}) \]

\[ v_6 = \dot{\theta}_2 \cdot L_{11} \]

\[ \omega_{13} = \frac{v_6}{L_{12}} \]

\[ \dot{h}_2 = \omega_{13} \cdot L_{13} \]

### 3.1.4 Estimation of control voltage

The final step of the control algorithm is the estimation of control voltage which is required to achieve the desired piston velocity. Because of the use of asymmetric cylinders, the direction of the movement has to be taken into account.

If \( \dot{h} > 0 \)
\[ Q = \dot{h} \cdot A_A \]
\[ U = U_l - \frac{Q}{C_{v_{ptoA}} \cdot \sqrt{p_{ref}}} \]  
(14)

If \( \dot{h} < 0 \)

\[ Q = \dot{h} \cdot A_B \]
\[ U = U_h - \frac{Q}{C_{v_{ptoB}} \cdot \sqrt{p_{ref}}} \]  
(15)

Else

\[ U = U_n \]  
(16)

where \( U_n \) is a voltage of the neutral spool position [V].

3.2 Input signal limitation

Because of limitations in the flow through the DCVs, the situation when one of the system cylinders reaches its velocity saturation is possible. In the case of such events, maintenance of the stability in the movement direction of the boom tip is important. It can be achieved through relevant limitations of the input velocity command. But, because of the complicity of the crane kinematics, such saturations have to be adaptive and unique for each configuration of the crane joint angles. Permanent estimation of velocity limits for current crane joints position is implemented to meet the described requirement.

3.2.1 Estimation of maximum angles’ velocity

The first step is estimation of the maximum piston velocity in both directions, retraction and extraction. This parameter depends on the maximum flow through DCVs and geometrical characteristics of cylinders.

\[ Q_{A_{max}} = C_{v_{ptoA}} \cdot (U_l - U_{l_{max}}) \cdot \sqrt{p_{ref}} \]  
(17)
\[ h_{max_{ext}} = \frac{Q_{\lambda_{max}}}{A_A} \]

where \( U_{t_{max}} \) is control voltage for maximum valve opening [V], \( Q_{\lambda_{max}} \) is flow through valve when opening is maximum [m\(^3\)/s], and \( h_{max_{ext}} \) is maximum stroke extraction velocity [m/s].

\[ Q_{B_{max}} = C_{vptoB} \cdot (U_{h_{max}} - U_n) \cdot \sqrt{p_{ref}} \]
\[ \dot{h}_{max_{ret}} = \frac{Q_{B_{max}}}{A_B} \]  (18)

where \( h_{max_{ext}} \) is maximum stroke retraction velocity [m/s].

As can be seen from equations 17 and 18, the limits of the stroke velocities are constant and depend only on the configuration of the hydraulic system.

Estimation of the maximum angular rate of the first joint can be done with moderate changes in equation 12.

\[ \dot{\theta}_{1_{max}} = \frac{\dot{h}_{1_{max}}}{L_3} \]  (19)

where \( \dot{h}_{1_{max}} \) is a 2 x 1 vector of maximum velocities of the lifting cylinder [m/s], and \( \dot{\theta}_{1_{max}} \) is a 2 x 1 vector for maximum angular rates of the first joints.

Estimation of the maximum angular rate of the first joint can be done with moderate changes in equation 13.
\[
\begin{align*}
\omega_{13}^{\text{max}} &= \frac{\dot{h}_{2}^{\text{max}}}{L_{12}} \\
v_{6}^{\text{max}} &= \omega_{13}^{\text{max}} \cdot L_{12} \\
\dot{\theta}_{2}^{\text{max}} &= \frac{v_{6}^{\text{max}}}{L_{11}}
\end{align*}
\]  

(20)

where \(\dot{h}_{2}^{\text{max}}\) is a 2 x 1 vector of maximum velocities of the bending cylinder [m/s], and \(\dot{\theta}_{2}^{\text{max}}\) is a 2 x 1 vector for maximum angular rates of the second joints.

3.2.2 Calculation of maximum velocity

The obtained values of the maximum angular rates are used in further calculations. The selection of the correct direction is required for the right velocity limits estimation. Therefore, internal feedback is implemented in the control algorithm. The correct value of the maximum joints’ angular rate from equations 19 and 20 is chosen based on the sign of the required joints’ angle velocity from equation 9 for estimation of the velocity limits.

After obtaining the correct angular rates limits, the Cartesian velocities limits are calculated utilizing the Denavit-Hartenberg method and equation 7 and implemented in the algorithm as parameters for a dynamic saturation block. [R.N. Jazar, 2010]

\[
^{0}v_{\text{max}} = ^{0}f(\theta) \cdot \dot{\theta}_{\text{max}}
\]  

(21)

3.3 Modelling results

The described equations and dependences were implemented and transferred into a programming environment for modeling Simulink.
A set of consequent linear commands was used to test the performance of the developed control algorithm. The relevant graphs are presented in figures 11 and 12.

Figure 11. Boom tip velocity in the horizontal direction
From the obtained results can be seen a significant error between the command and the boom tip movement. From the step-by-step analysis of the model, it has been discovered that the cylinders move in certain directions uncontrolled. The acknowledgment of that can be seen in figures 13 and 14.
Figure 13. Lifting cylinder velocity

Figure 14. Bending cylinder velocity
Hence, the cylinders are not capable of lowering the load with controlled velocity, even though the flow to the controlled chamber is controlled with the DCVs. This phenomenon calls a cylinder’s run-out, and it happens because of uncontrolled flow from the pressurized chamber to the tank. Another consequence of it is cavitation in the controlled chamber because the piston moves significantly faster than expected from the inlet flow. It is shown in figures 15 and 16.

Figure 15. Lifting cylinder pressure
Figure 16. Bending cylinder pressure

Hence, the lowering velocity of a lifting cylinder depends on the load and the valve flow rate coefficient from the load port to the tank port.
4 Improvement of the system performance

Several consistent patterns were observed during the development process. They lead the ways to improve the system behavior and accuracy of the result performance.

4.1 DCV parameters change

An outstanding feature of the used pressure-compensated valves Danfoss PVG 32 is exceedance several times of the valve flow coefficients from the load ports to the tank port over the valve flow coefficients from the pump port to the load ports. The restriction of flow from the pressurized port to the tank is the solution to prevent the observed cylinders’ run-out during the load lowering. The parameters’ change is presented in table 6.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Physical dimension</th>
<th>Initial value</th>
<th>New value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lifting cylinder’s DCV A to T and B to T rated flow</td>
<td>lpm</td>
<td>130</td>
<td>45</td>
</tr>
<tr>
<td>Bending cylinder’s DCV B to T rated flow</td>
<td>lpm</td>
<td>30</td>
<td>5</td>
</tr>
</tbody>
</table>

4.1.1 Modelling results

A set of consequent linear commands was used to test the performance of the developed control algorithm with improved valve parameters. The relevant graphs are presented in figures 17 and 18.
As can be seen from the results, the decrees of the valve flow coefficients from the load ports to the tank port significantly improve the behaviour of the system and prevent lifting cylinders’ run-out.
The system with changed parameters was tested with loads of 50 kg and 320 kg. The simulation results are presented in figures from 19 to 22.

Figure 19. Boom tip velocity in the horizontal direction with a load of 50 kg

Figure 20. Boom tip velocity in the vertical direction with a load of 50 kg
Figure 21. Boom tip velocity in the horizontal direction with a load of 320 kg

Figure 22. Boom tip velocity in the vertical direction with a load of 320 kg

As can be seen from the obtained results, the implemented method significantly improves system behaviour across the entire load range. It is also notable, that after adjusting the
system parameters for a certain load, the system controllability and accuracy decrease with
the increase of the load differs from the setup point.

Therefore, the drawback of such a solution is that this parameter change has to be adjustable
to the load. It is required because the flow through the valve pressure-compensator does not
control the pressure drop over the valve in this direction. Hence, if the load increases, then
pressure in the pressurized chamber enlarges too, and so does the pressure drop over the
valve. That means, that after a certain amount of load, the cylinder's run-out resumes, as can
be seen in figure 22 between 32 and 37 seconds.

At the same time, the decrease of load requires an increase in the valve flow coefficient,
because increased hydraulic resistance leads to an increase in pressure in the pressurized
chamber. To overcome this additional resistant force, the hydraulic power station has to
maintain increased pressure which leads to increased power consumption and a decrease in
system efficiency.
4.2 Implementation of counterbalance valves

The use a counterbalance valve is a widely used solution to control a cylinder that is holding a load. During lifting the load, it works as a check valve and allows the working fluid to freely enter the pressurized cylinder chamber. On the other hand, the valve construction allows to controllably lower the load and maintain the oil pressure in the pressurized chamber.

Figure 23. The log crane with counterbalance valves hydraulic circuit
4.2.1 Modelling of a counterbalance valve

The mathematical model of the counterbalance valve is formulated in the simplified semi-empirical form and depends on pressure difference over the valve. [Handroos, 1996]

If $p_1 > p_2$

$$\dot{C}_V = \frac{p_1 - p_{ref} - a_4 \cdot p_2 + a_5 \cdot p_3 - C_V \cdot [a_1 + a_2 \cdot (p_1 - p_2)]}{a_3}$$

$$Q = C_V \cdot \sqrt{p_1 - p_2}$$

(22)

Else

$$Q = C_V \cdot \sqrt{p_2 - p_1}$$

(23)

Implemented valve parameters are presented in table 7.
Table 7. Parameters of counterbalance valves

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Physical dimension</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_1$</td>
<td>-</td>
<td>$18.2 \cdot 10^{12}$</td>
</tr>
<tr>
<td>$a_2$</td>
<td>-</td>
<td>$2.9 \cdot 10^{6}$</td>
</tr>
<tr>
<td>$a_3$</td>
<td>-</td>
<td>$0.113 \cdot 10^{12}$</td>
</tr>
<tr>
<td>$a_4$</td>
<td>-</td>
<td>7.87</td>
</tr>
<tr>
<td>$a_5$</td>
<td>-</td>
<td>4</td>
</tr>
<tr>
<td>$p_{ref}$</td>
<td>bar</td>
<td>110</td>
</tr>
<tr>
<td>Check valve rated flow with a pressure drop of 6 bar</td>
<td>lpm</td>
<td>40</td>
</tr>
</tbody>
</table>

4.2.2 Modelling results

A set of consequent linear commands was used to test the performance of the developed control algorithm with improved valve parameters. The relevant graphs are presented in figures 25 and 26.

Figure 25. Boom tip velocity in the horizontal direction
From the analysis of the figures, we can conclude that implementation of the counterbalance valves fully remedies the cylinders’ run-out, but occasions the significant lack of hydraulic system stability. In general, the manipulator itself is predisposed to vibrations. These oscillations of the crane construction result in additional pressure pulsations in the hydraulic system. A counterbalance valve happens to lack stability because of its spool mechanism and dynamic parameters. Hence, the simultaneous influence of the mentioned vibration sources causes significant instability in the hydraulic system. It can be seen in figure 27 between 2 and 7 seconds.
4.2.3 Counterbalance parameters improvement

The pilot ratio is one of the key counterbalance valve parameters. The increase of this parameter leads to improvement of the valve’s static characteristic and degradation of its dynamic performance. In the case under consideration, the smallest value of the pilot ratio is tested to improve the system behavior.

Table 8. Parameters’ change

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Physical dimension</th>
<th>Initial value</th>
<th>New value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pilot ratio ($a_5$)</td>
<td>-</td>
<td>4</td>
<td>1.25</td>
</tr>
</tbody>
</table>

4.2.4 Modelling results

A set of consequent linear commands was used to test the performance of the developed control algorithm with improved valve parameters. The relevant graphs are presented in figures 28 and 29.
As can be seen from the modeling results, the implementation of the counterbalance valves removes the cylinders’ run-out. At the same time, it provides additional delays in the system,
consequently, it decreases the stability of the system. There are two sources of this delay and each of them takes place in different directions of the cylinder’s movement, and the reason for both thus events are fluid compressibility.

Firstly, during the load lifting process, the delay happens because the counterbalance valve performs as a check valve, so the flow does not pass before the pressure in the hose overreaches pressure in the pressurized cylinder chamber. This event can be observed in figure 30 between 21 and 23 seconds.

On another hand, during the lowering of the load, the delay happens because the counterbalance valve is closed, and initially pressure in the control line is too low. Therefore, the valve stays closed until the pressure in the controlled chamber surpasses the low limit of the valve. It can be seen in figure 31 between 12 and 13 seconds.

Figure 30. Pressure in lifting cylinder's volumes
Figure 31. The counterbalance valve flow coefficient
5 Conclusions

In the described work, an open-loop coordinate boom tip control algorithm was implemented in the control system for a log crane with proportional mobile valves. The input command in the control system is tip velocity in a Cartesian coordinate system. The output signal from the control system are control voltages for the DCVs, and the control algorithm requires measurements of the cylinders’ pistons positions to operate.

The results obtained with the initial hydraulic circuit showed the impossibility of the use of an open-loop control algorithm for a hydraulic manipulator because of observed cylinder run-out. To address the challenge two solutions were considered.

The first one is the decrease of valve flow coefficients from the load ports to the tank port, or adding a hydraulic restriction, such as a throttle, into the line between a DCV and a tank. The drawback of such a solution is the necessity of changeability of the hydraulic resistance because it has to depend on the crane load.

Another possible way to improve the system is the implementation of counterbalance valves. It is a commonly employed method in lifting hydraulic circuits to prevent oil from uncontrollably leaving a cylinder that is holding a load. On the minus side, the use of a counterbalance valve superinduces a delay in the hydraulic circuit which additionally decreases the system stability.

To conclude, in the case of the use of an open-loop control system, the stability and predictability of the system play a crucial role in the accuracy of the coordinate control. Future work should focus on the increase of the system handling characteristics within a full range of operational environments.
References


Manner J., Gelin O., Mörk A., Englund M. 2017. Forwarder crane’s boom tip control system and beginner-level operators. Silva Fennica vol. 51 (2) article id 1717.

