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Identification and Control of a Hydraulic Forestry Crane

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Abstract: This article presents the identification and control of an electro-hydraulic crane. The crane is of the type used on forestry vehicles known as *forwarders*, which travel off-road collecting logs cut by the *harvesters*. The dynamics identified include significant frictional forces, dead zones, and structural and hydraulic vibrations. The control algorithm proposed, comprised of a linear controller and a compensator for nonlinearities, is able to accurately track a reference trajectory for the end effector, despite uncertainties in the arm mechanics and hydraulic system dynamics. A further control design is presented which uses an inner loop to compensate for vibrations in the hydraulic system, and its performance is experimentally verified.

Keywords: Robotics in Agriculture and Forestry, Hydraulic Manipulator, System Identification.

1. INTRODUCTION

The Swedish forest industry has a long-term goal of developing autonomous and semi-autonomous forestry vehicles [1], [2].

There are mainly two types of off-road vehicle used in the forestry industry: the *harvester*, which fells and delimbs the trees, and cuts the trunk into logs of a predetermined size, and the *forwarder*, which collects the logs in a tray, and carries them to the nearest road for collection. A forwarder is shown in Fig. 1.

These two types of vehicle have similar on-board hydraulic manipulators (cranes). An important stage towards automation of forestry vehicles is establishing accurate dynamical models of these manipulators, and designing high-performance low-level control systems. These can then be used in concert with high-level motion planners and teleoperation systems, e.g. [3].

Related recent publications include [4, 5, 6, 8]. Of particular interest is the work of Münzer [7], which considers many of the same issues on a crane with similar configuration to the one in our lab.

In this paper recent experimental results on a real forwarder crane working towards this goal will be presented. Each joint controller uses an angular position sensor (encoder) for feedback control of the joint position. An inner-loop control based on direct control of cylinders pressure/force has been implemented for the first link of the crane, and shows impressive results on damping of oscillations. However, it is not yet implemented on the other links.

Fig. 1. A forwarder: the Komatsu 860.1.

The structure of the paper is as follows: in Section 2, a detailed description of the experimental setup is described; in Section 3 identification of nonlinearities in the system is presented; in Section 4 the method of friction compensation used is explained; experimental results for tracking a reference trajectory are given in Section 5; in Section 6 some early results on direct control of hydraulic force using pressure sensors is given; some brief conclusions and discussions of future directions are given in Section 7.

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Fig. 2. Crane installed at the Department of Applied Physics and Electronics, Umeå University.

2. EXPERIMENTAL SETUP

Experimentations and tests are carried out at the Smart Crane Lab located at Umeå University. The Laboratory is equipped with an electro-hydraulic actuated crane CRANAB model 370RCR (see Fig. 2), a forwarder crane which is somewhat smaller than most on cranes on production forwarders, but similar in configuration and dynamics.

The hydraulic hardware in the Smart Crane Lab consists of:

- hydraulic cylinders manufactured by Valmet,
- a unit containing six servo-valves from Sauer-Danfoss model L90LS,
- the power supply for this system consists of an electrical motor driving a hydraulic pump (type H4-010214-132) set to provide a constant supply pressure of 180 bars for the whole system operation.

In addition, the associated sensing equipment includes:

- encoders of 4000 pulses/turn, present to measure the various links angular positions,
- pressure transducers (HD 3403-10-C3.39) capable of sensing in a range of [0, 200] bar.

The crane can be directly manipulated by a chair, same as the ones mounted in the cabin of real *forwarders*. This chair contains buttons and joysticks that allow the drivers to have full control over the whole machine operation and the crane. Signals from this chair are handled by the processing unit.

The processing unit applied in this particular case is 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 is used to implement executable code for the processing unit. In order to provide a sufficient range of current to drive the servo valves a RapidPro unit is installed. The RapidPro contains a Power Unit (PU) which transforms the low voltages generated by the MABX to appropriate currents for the valve solenoids. The current in each circuit can be measured. Furthermore, there is a Signal Conditioning Unit (SCU) which can handle the incoming measurement signals to voltage levels needed to be fed to the MABX. Finally, a Dell PC is present to monitor and serve as an on-line user interface through the use of Control Desk.

3. IDENTIFICATION OF NONLINEARITIES

Forestry machines are constructed in a robust way to handle different terrains and climates. For control purposes this type of system becomes a highly nonlinear system with uncertain parameters that need to be identified experimentally.

For instance, the valve has a significant dead-zone due to overlap on the spool in order to avoid leakages. Saturation to its maximum actuation level is also present. Moreover, in such heavy duty machines, friction occurs in the mechanical structure as well as in the hydraulic components and actuators, effects that have to be taken into account.

In order to have a reliable compensation for nonlinearities many tests were performed in order to extract all the possible information.

3.1 Identification of friction

A lumped model representing the total friction, present in the mechanical construction, hydraulic components and actuators, was identified. The friction is modeled by a static map as a function of current levels with respect to link velocity [9]. The expression "pseudo-friction" is used, since it represents the valve current required to overcome friction, rather than actual forces or torques.

The experiments were conducted as follows: a step current is generated in the valve which eventually results in a change of position of the corresponding actuator provided that the signal level lies above the stiction values. After some transient effect the velocity reaches a constant steady state and it can be matched to the applied valve current. The magnitude of the steps in the valve current are increased stepwise in order to produce a range of constant velocities beginning from zero.

For example, in Fig. 3 the resulting static friction maps for the first link is shown. The static frictional effects are mainly caused by Coulomb and viscous friction, but can also have some contribution from dead zones in the hydraulic valves. The identified maps can be used for compensation purposes in control design in a straightforward way since they are in terms of the input current to the four-way valve.

4. FRICTION COMPENSATION

In order to have high accuracy in the position control of the crane links, some undesired effects mainly caused by friction need to be removed. The classical approach is to use a feed-forward term added to the control signal using an estimate of the friction derived from an estimate of the link velocity [9]. In Fig. 4 such a compensation scheme is illustrated as block diagram.

However, if there is very large component of Coulomb friction, as in our case, then noisy estimates of link velocity around zero can lead to a "chattering" effect [9]. A further problem is that, if the link is stationary but a small

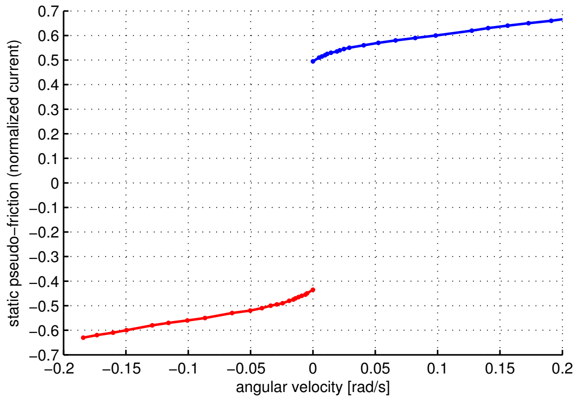

Fig. 3. Static pseudo-friction map for the first link.

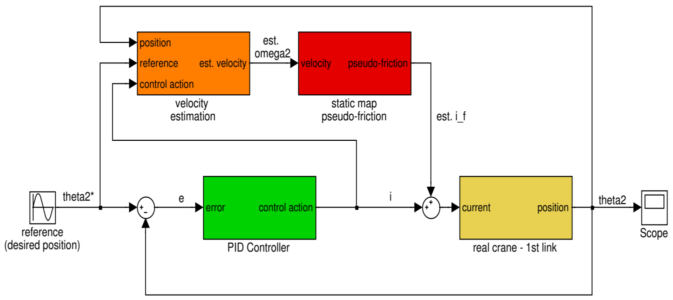

Fig. 4. Friction Compensation block diagram as example for first link.

control input has been applied, the friction compensator will randomly apply any compensation signal, since the link velocity is zero.

To overcome these difficulties, a modification in the classical friction compensation approach is proposed, which in our particular case gives satisfactory results in the practical sense. The velocity-estimate used in the compensation signal is calculated like so:

$$\hat{\dot{\theta}} = (1 - \alpha(\dot{r}))\dot{\theta} + \dot{r}\alpha(\dot{r}) + \mu\,\mathrm{sign}(i). \tag{1}$$

Here, $\dot{\theta}$ is the velocity estimated from a differentiator and low-pass filter arrangement, and $\dot{r}$ is the derivative of the link's position reference signal, which is known exactly. The function $\alpha(\dot{r})$ is a bell-shaped function which is equal to 1 at $\dot{r} = 0$, and smoothly drops to zero outside the range $[-0.1, 0.1]$ rad/s. The motivation is that, around zero, assuming reasonably good tracking, the reference velocity is smoother and more reliable measurement of velocity than a noisy estimate of the link velocity.

The term $\mu\,\mathrm{sign}(i)$ is added to overcome the second problem described above. If the link is stationary, then the direction of the valve current i is used to decide to which direction the friction compensation force should be applied. The constant μ is chosen to be a small value, such as 0.001.

5. REFERENCE TRAJECTORY TRACKING EXPERIMENTS

In this section, results of a trajectory tracking experiment is shown. The control strategy is based on PID position feedback control and a feed-forward term for friction

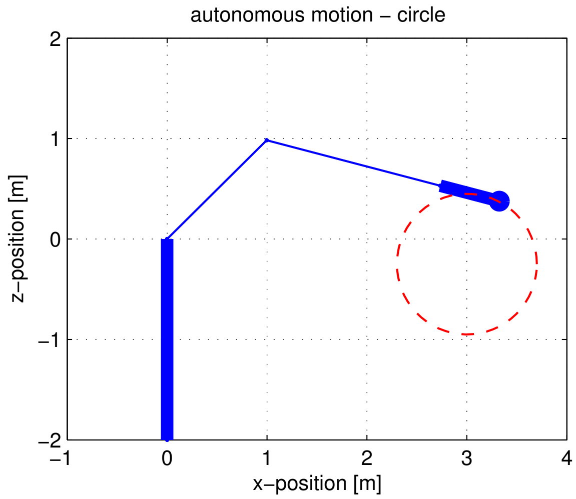

Fig. 5. Desired reference trajectory for the boom-tip in two-dimensional space. Period of rotation: 25 seconds

compensation (see Fig. 4). The intention is not to show a sophisticated motion planner, but rather evaluate the performance of the reference tracking utilizing position feedback and friction compensation.

In Fig. 5 the desired motion is depicted. Here, the boom-tip has to follow a circular reference counter-clockwise in a two-dimensional space. Although not a trajectory that would be common in practice, it has many regions close to zero link velocity, and thus provides a useful test of the friction compensation presented above.

By inverse kinematics it is possible to compute individual trajectories for each joint if a particular trajectory for the telescopic arm is defined. Fig. 6 shows the generated trajectory and the final crane motion super-imposed. As it can be seen, the reference tracking is quite accurate, however there exist some oscillations during the lifting phase.

The error for such a trajectory is less than 15 mm in average. During the cycle the end effector is moving at approximately 0.2m/s. Oscillations show up particularly during the crane's lifting phase, observable for the first quadrant of the phase angle. This can be explained by the fact that this is a hydraulic system being controlled. Due to large energy losses in hydraulic actuators, having three hydraulic actuators working at once is most likely not the most efficient solution. Another problem is that the solution does not take into account the geometry of the manipulator, i.e. the trajectories of individual links were chosen somewhat arbitrarily. For example, having the first link rotating at high velocity requires high energy, whereas extending the boom section at high velocity takes less energy due to the difference in the inertias. An interesting task to consider is optimizing the trajectories in terms of time, energy loss, oil consumption, or other costs.

6. DIRECT HYDRAULIC CONTROL

By measuring the pressures in the hydraulic cylinder chambers it is possible to counteract oscillations much more directly than via measurements of link positions. In this section, results on damping of oscillations in the first

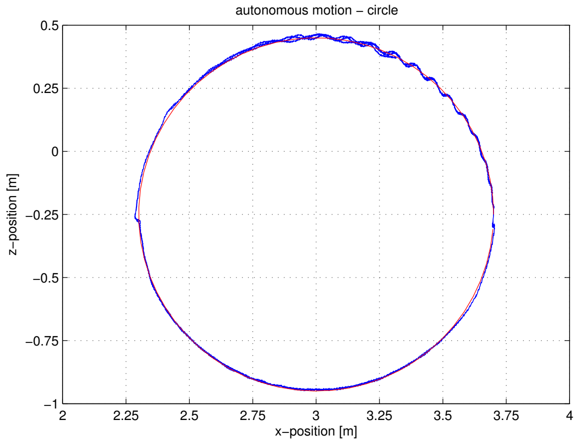

Fig. 6. Desired reference trajectory vs. performed boom-tip trajectory.

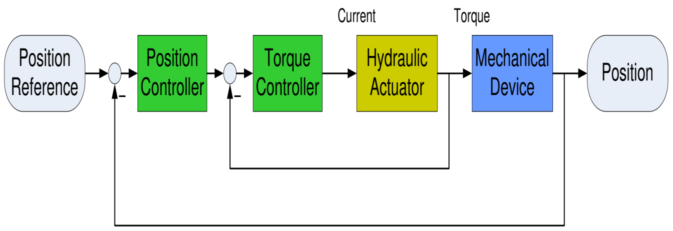

Fig. 7. Two stages cascade control.

link of the crane are shown. The analysis is concentrated on this link since it showed to be the main cause of such an oscillatory behavior. Besides, the absence of pressure sensor devices does not allow us to reproduce the reference tracking system described above. However, such systems are currently under preparation.

The idea is to have a cascade control scheme to control the hydraulic force and angular position simultaneously, which will result in reduced oscillations while asymptotically stabilize the desired force and motion.

Fig. 7 shows a block diagram describing the scheme of the cascade control. As it can be seen, the control problem is split into two stages. The outer loop controller calculates the reference piston forces F_{ref} needed to drive the manipulator along the predefined joint space θ_{ref}. The inner loop controller takes this force reference F_{ref} and computes the servo-valve input current u needed to make the true piston forces F asymptotically track F_{ref}. Since F asymptotically tracks F_{ref}, which is itself an asymptotically stabilizing control for the manipulator motion around θ_{ref}, the overall cascade system is asymptotically stable around the trajectory reference θ_{ref}.

6.1 Hydraulic force model

Since modeling the hydraulic force from first principles is far from trivial, due to nonlinearities and the very large number of parameters, in this work it is shown how to use system identification methods to find a transfer function relating the input current (to the four-way valve) with the hydraulic force produced by the cylinders. However, the model structure and order are chosen based on first-principles analysis, which are described below.

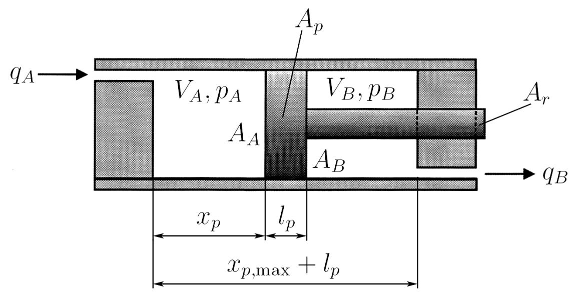

Fig. 8. Double acting piston-type actuating cylinder.

Disregarding internal disturbances, nonlinearities caused by friction, etc, the applied force produced by the linear hydraulic actuator shown in Fig. 8 is given by [11]

$$F = A_A p_A - A_B p_B \tag{2}$$

where A_A and A_B are the areas at each respective chamber, p_A and p_B are the pressures measured at each chamber. The relations governing the dynamics of the pressures are [11]:

$$\begin{aligned} \dot{p}_A &= \tfrac{\beta}{V_A}[-C_{em}p_A - A_A\dot{x}_p + q_A], \\ \dot{p}_B &= \tfrac{\beta}{V_B}[-C_{em}p_B + A_B\dot{x}_p - q_B], \end{aligned} \tag{3}$$

where V_A and V_B are the chambers volumes, C_{em} is the cylinder internal leakage, x_p is the pistons displacement, q_A and q_B are the input and out flow to and from the cylinder chambers. The linearized version of the hydraulic flow for q_A and q_B is [11], [12]:

$$\begin{aligned} q_A &= 2K_q x_s - 2K_c(p_A - p_s/2) \\ q_B &= 2K_q x_s + 2K_c(p_B - p_s/2) \end{aligned} \tag{4}$$

where K_q and K_c are known as the valves coefficients, p_s is the supplied pump pressure and x_s denotes the motion of the four-way valve spool displacement.

The relation between the four-way valve spool position x_s and the input current u can be written as [12]

$$x_s(s) = \frac{\omega_n^2}{s^2 + 2\xi\omega_n + \omega_n^2} u(s) \tag{5}$$

where ξ and ω_n represent the damping ratio and natural frequency characteristics of the servo valve.

By combining equations (2) to (5) and by considering that the system parameters represent some numerical values, a linear model is obtained, in which parameters have been collected and substituted to simplify notation,

$$\begin{aligned} F_{cyl}(s) = &\tfrac{A_A \cdot c\omega_n^2(s+f) + A_B \cdot h\omega_n^2(s+a)}{(s+a)(s+f)(s^2+2\xi\omega_n+\omega_n^2)} \cdot u(s) \\ &+ \tfrac{s(-A_A \cdot b(s+f) - A_B \cdot g(s+a))}{(s+a)(s+f)} x_p(s) \\ &+ \tfrac{A_A e(s+f) - A_B m(s+a)}{(s+a)(s+f)}. \end{aligned} \tag{6}$$

Based on this analysis, it is shown that the hydraulic torque can be modeled by a fourth-order linear time invariant system.

6.2 Hydraulic system identification

Closed-loop identification was used to find the hydraulic dynamics. Different trajectories were designed and tracked

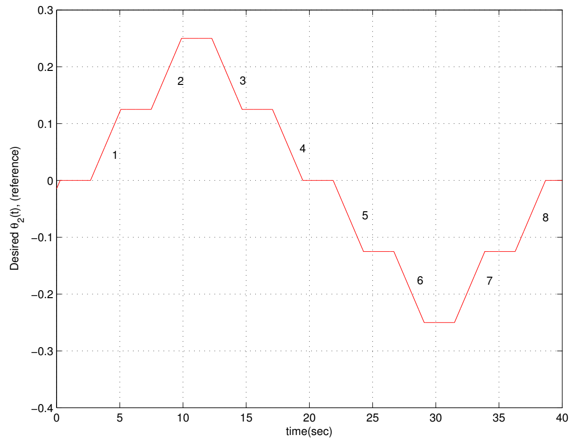

Fig. 9. Desired trajectory for the first link position.

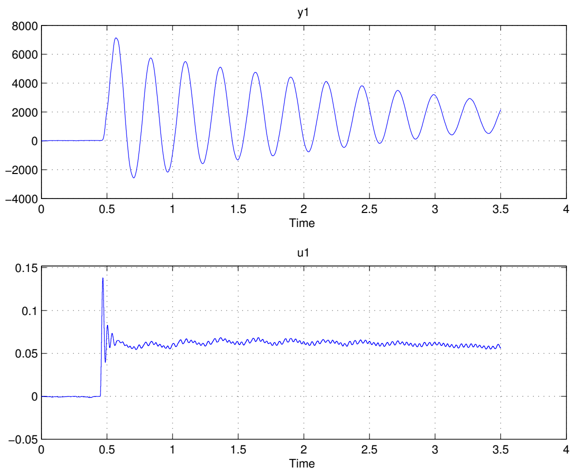

Fig. 10. Force response to the desired trajectory. The graph on top shows Force and bottom shows input current.

using the position control presented in Section 5. The resulting valve current resembles a series of step responses. In Fig. 9 one example of such trajectory is presented, and in Fig. 10 the response of the force and the input current for the first step (denoted by 1 in Fig. 9), from this trajectory, is shown.

An analysis of the Fourier transform and spectral analysis (see Fig. 11 and 12) reveals the second order characteristics of the valve dynamics (5).

Fourth order systems were identified using several methods including the prediction error method (PEM) and the output error (OE) method. The best validation results were obtained with PEM. The transfer function found was:

$$F_{pem}(s) = \frac{b_3 s^3 + b_2 s^2 + b_1 s + b_0}{a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0} \tag{7}$$

where $b_3 = 2.123 \times 10^5$, $b_2 = 4.253 \times 10^7$, $b_1 = 4.579 \times 10^8$, $b_0 = 6.197 \times 10^8$, $a_4 = 1$, $a_3 = 15.68$, $a_2 = 647.2$, $a_1 = 8636$, $a_0 = 2.353 \times 10^4$.

Properties such as DC gain, damping and natural frequency in terms of bode diagram is depicted in Fig. 13 for different methods.

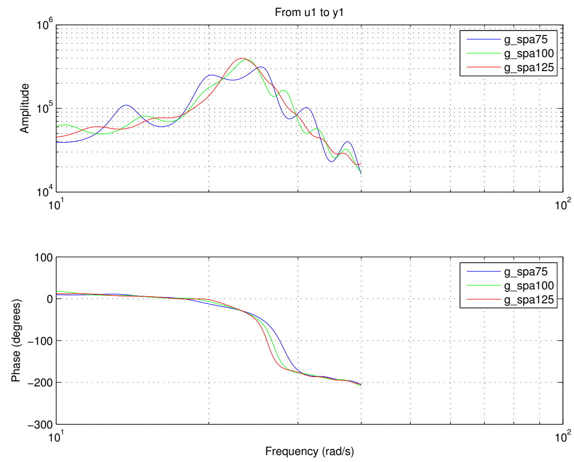

Fig. 11. Spectral density analysis of recorded data using different hamming windows.

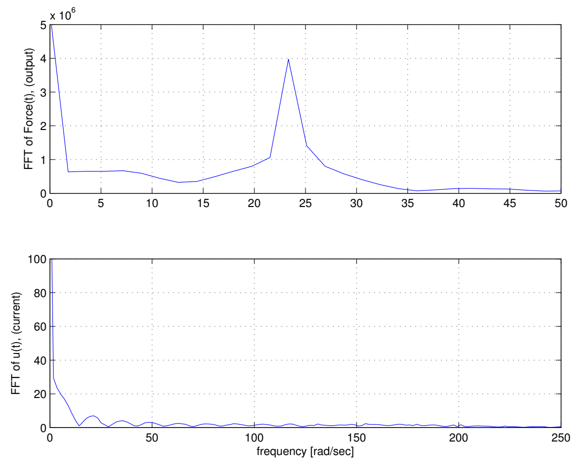

Fig. 12. Fast Fourier transform of the recorded data.

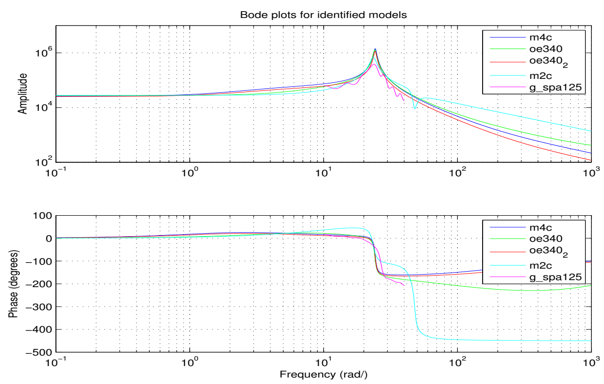

Fig. 13. Bode Diagram from different identification methods super-imposed to the spectra analysis of the initial data.

6.3 Control design for the hydraulic force

The linearized models found by identification are an approximation of the real system dynamics. Many of the parameters will vary depending on temperature, component age, and other factors, and it is important to design a controller which is robust to such uncertainties [10].

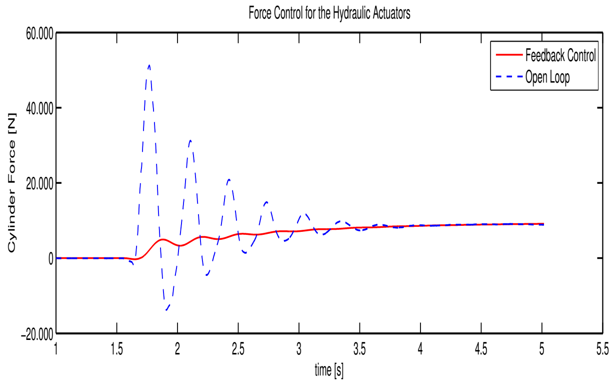

Fig. 14. Force control experimental result.

The H^∞ loop-shaping design method is an effective way to design a robust controllers to meet our requirements. The approach is based in choosing a particular designed closed-loop response, and then finding an optimal controller which minimizes the H^∞ norms of weighted performance measures [13, 14].

A simple target loop shape is a first order low pass filter of the form $G_{cl}(s) = 1/(s/W_c + 1)$, where W_c represents the cutoff frequency. The resulting controller, for a target loop with $W_c = 3$, has the transfer function:

$$C(s) = \frac{c_4 s^4 + c_3 s^3 + c_2 s^2 + c_1 s + c_0}{l_5 s^5 + l_4 s^4 + l_3 s^3 + l_2 s^2 + l_1 s + l_0} \quad (8)$$

where $c_4 = 0.02579$, $c_3 = 0.4043$, $c_2 = 16.69$, $c_1 = 222.7$, $c_0 = 606.7$, $l_5 = 1$, $l_4 = 4308$, $l_3 = 8.373\times10^5$, $l_2 = 1.133\times10^7$, $l_1 = 3.854\times10^7$, $l_0 = 3.594\times10^7$.

In Fig. 14 some experimental results are shown. In this specific result the force of the first link was recorded while performing some motion (dotted line) under only position control, while the second plot shows the pressure under a cascade control configuration. It can be said that very large oscillations were damped effectively using this method.

7. CONCLUSIONS AND FUTURE WORK

The primary target of this project is to increase production efficiency of forestry machines by means of intelligent control techniques. Essential to any autonomous or semi-autonomous forestry crane is good low-level system identification and control. In this paper we have presented some promising preliminary results along these lines. The main problems overcome were related to friction identification and compensation, and damping of oscillations within the hydraulic system.

There is much that can be done to continue this project. At present the cascade-control structure, which showed very promising results in damping of oscillations, has only been implemented on the first link of the crane. Extending this to all links is a natural evolution of the current work, and it is expected that this will allow much smoother trajectory tracking to be achieved.

Other related work includes using the proposed low-level in concert with higher-level control schemes, such as semi-autonomous teleoperation systems (see, e.g., [3]), and fully-autonomous systems based on optimized motion planning.

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