Modeling and Control of Hydraulic Rotary Actuators

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***Abstract*— The steps for modeling and control of a hydraulic rotary actuator are discussed. Our aim is to present experimental results working with a particular sensing device for angular position as a complement to pressure sensing devices. We provide the steps in experimental system identification used for modeling the system dynamics. The cascade controller designed contains an inner loop for an accurate tracking of torque while stabilizing position reference trajectories. The performance of this design is experimentally verified.**

***Index Terms*— Robotics in Agriculture and Forestry, Hydraulic Manipulator, System Identification, Control Design.**

I. Introduction

The Swedish forest industry has a long-term goal of developing autonomous and semi-autonomous forestry vehicles [3], [1]. These vehicles are equipped with cranes mainly for harvesting or collection of logs. To this end two type of end effectors are applied: the *harvester head*, which fells and delimbs the trees, and cuts the trunk into logs of a predetermined size, and the *forwarder gripper*, which holds the logs while being carried to the tray for transportation (see Fig. 1). Hydraulic rotary actuators are mostly used as actuators for these devices.

An important stage towards automation of this vehicles is the control of the end effector to perform predefined tasks. For this aim, it is important establishing accurate dynamical models of the rotary actuators, and designing high-performance low-level control systems [5], [8], [6], [9]. However, the difficulties are imposed by the lack of sensing devices suitable for measuring the rotational displacement. The manufacturing of this particular hydraulic devices does not allow the direct installation of sensors, such as encoders, resistive elements, etc. On the other hand, these fragile electronic devices are not suitable for the rough working environment of these machines. Hitting the ground, trees, etc, would easily make them to malfunction.

These observations motivated the design of our own partial solution for a robust sensing of the angular position of these devices. The goal of this article is to presents results in modeling and control design applying this particular solution.

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Fig. 1. A forwarder: The Valmet 860.1 manufactured by Komatsu Forest AB.

To this end we provide results of the steps followed along the evolution of the project. The main contribution is to demonstrate that accurate motion control can be achieved by the combination of two sensing devices: the hydraulic pressure transducers and a home-made magnetic encoder.

The structure of the paper is as follows: in Section II, the experimental setup at Umeå University is described briefly together with the solution of sensing adopted; in Section III, some preliminaries about system modeling are defined; experimental identification of parameters of the mechanical setup as well as hydraulic system dynamics are investigated in Sections IV-A and IV-B; exemplified control design techniques for position feedback, torque feedback and a combination as cascade control are presented in Sections V-A, V-B and VI-C; brief conclusions and discussions of future directions are given in Section VII.

II. Experimental Setup

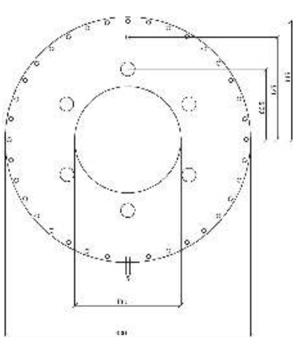
The experimental study is carried out at the Smart Crane Lab at Umeå University. The Laboratory is equipped with an electro-hydraulically actuated crane of the type 370RCR (see Fig. 2) manufactured by Cranab AB. It is somewhat smaller than most cranes on production forwarders, but similar in configuration and dynamics.

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

The hydraulic hardware in the Smart Crane Lab is designed to supply a constant pressure of 180 bars for the whole machine operation. In addition, the associated sensing equipment includes encoders of 1000 pulses/turn to measure the various links angular positions, and pressure transducers capable of sensing in a range of [0, 200] bar.

The crane can be directly manipulated by an operator station, same as the ones mounted in the cabin of real *forwarders*. This station contains buttons and joysticks that allow the operator to have full control over the vehicle and the crane. For the control of the crane as well as the implementation of algorithms a dSPACE Prototyping Hardware is used.

The angular position of the end effector is measured using an aluminium disc (see Fig. 3(b)) with a total of thirty-six magnets with a resolution of ten degrees as seen in Fig. 3(a). A magnetic sensor manufactured by IFM electronics is used to sense these magnets. The signals coming from the magnetic sensors are read and converted to incremental position values by the processing unit.



(a)

(b)

Fig. 3. (a) CAD model of the disk designed for the rotator's angular position sensing; (b) Frontal view of the installation of such a device.

III. Describing Dynamics of the plant

A. Modeling the mechanics

The equation of motion for a mechanical system with one degree of freedom is,

$$J \cdot \ddot{\theta} = \tau \,, \tag{1}$$

where J is the inertia of the mechanical link, θ is the angular position and τ is the sum of external torques. In our case, τ can be roughly represented as

$$\tau = \tau_{hyd.dynamics} - \tau_{friction} - \tau_{disturbance} \,, \tag{2}$$

where $\tau_{hyd.dynamics}$ is the hydraulic input torque that causes a change in rotational motion, $\tau_{friction}$ corresponds to friction forces, and $\tau_{disturbance}$ describes all external disturbances. Combining (2) and (1), the dynamics for a particular link can be modeled by

$$J \cdot \ddot{\theta} = \tau_{hydraulics} - \tau_{friction} - \tau_{disturbance} \,. \tag{3}$$

In order to model the valve cracking point together with friction forces, the friction model composed by Coulomb and viscous friction [10] can be applied. Thus, the friction torque can be modeled as

$$\tau_{friction} = b \cdot \dot{\theta} + \tau_{Coulomb} \cdot \text{sign}(\dot{\theta}) \,, \tag{4}$$

where b is the viscous friction and $\tau_{Coulomb}$ represents the Coulomb part of the friction. Assuming that the system is free of disturbances, by substituting (4) into (3) we get

$$J \cdot \ddot{\theta} = \tau_{hydraulics} - b \cdot \dot{\theta} - \tau_{Coulomb} \cdot \text{sign}(\dot{\theta}) \,. \tag{5}$$

Moreover, if we make the assumption that the hydraulic torque $\tau_{hydraulics}$ generated by the cylinder has a direct relation to the control input $u(t)$, then the next pseudo-model could be considered instead of (5):

$$\hat{J} \cdot \ddot{\theta} = u - \hat{b} \cdot \dot{\theta} - \hat{\tau}_{Coulomb} \cdot \text{sign}(\dot{\theta}) \,, \tag{6}$$

where $\hat{J}$, $\hat{b}$ and $\hat{\tau}_{Coulomb}$ will have a proportional relation to the real parameters J, b and $\tau_{Coulomb}$.

B. Modeling the hydraulic dynamics

The control input u is related to a current applied to the valve, which proportionally translates into change of pressure given a laminar flow [7]. In practice, however, the hydraulic actuation has complex nonlinear dynamics, such that $\tau_{hydraulics}$ becomes rather a dynamical response to the control input u [2]. A linear approximation of the hydraulic actuation as response to an electrical input is derived next.

Disregarding internal disturbances, nonlinearities caused by friction and the efficiency of the components, the applied torque produced by a rotary actuator (see Fig. 4) is given by [7]

$$\tau_{rotor} = D_a(p_1 - p_2), \tag{7}$$

where D_a is the volumetric displacement, p_1 and p_2 are the pressures measured at each chamber. The relations governing the dynamics of the pressures are [7]:

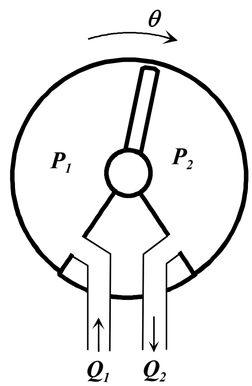


Fig. 4. Hydraulic rotary actuator (taken from [2]).

$$\begin{aligned} \dot{p}_1 &= \tfrac{\beta}{V_1}[-C_{em}p_1 - D_a\dot{\theta} + q_1], \\ \dot{p}_2 &= \tfrac{\beta}{V_2}[-C_{em}p_2 + D_a\dot{\theta} - q_2], \end{aligned} \tag{8}$$

where V_1 and V_2 are the total volume in the actuator chambers, C_{em} is the actuator internal leakage, θ is the rotor displacement, q_1 and q_2 are the input and out flow to and from the rotor chambers. The linearized version of the hydraulic flow for q_1 and q_2 is [7], [2]:

$$\begin{aligned} q_1 &= 2K_q x_s - 2K_c(p_1 - p_s/2) \\ q_2 &= 2K_q x_s + 2K_c(p_2 - p_s/2) \end{aligned} \tag{9}$$

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 [2]

$$x_s(s) = \frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2}u(s) \tag{10}$$

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

By combining equations (7) to (10) 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,

$$\tau_{hydraulics}(s) = \frac{b_3s^3 + b_2s^2 + b_1s + b_0}{a_4s^4 + a_3s^3 + a_2s^2 + a_1s + a_0}u(s)\,. \tag{11}$$

This fourth order transfer function suggests a linear plant model of the process under certain conditions.

IV. Identification of model parameters

A. Parameter Estimation of the Mechanical Model

A friction compensation scheme [4] can be applied to compensate the nonlinear effect produced by the Coulomb friction,

$$u = v + \hat{\tau}_{Coulomb} \cdot \mathrm{sign}(\dot{\theta}),$$

where v is the nominal input signal. In this way the model (6) is linearized as

$$\hat{J} \cdot \ddot{\theta} = v - \hat{b} \cdot \dot{\theta} + e(t), \tag{12}$$

where the parameters $\hat{J}$, $\hat{b}$ are to be identified, and $e(t)$ can be interpreted as the modeling error.

In order to estimate parameters of the model (12), it is suggested

- to introduce a feedback action, and
- to use a reference signal of a particular band-pass characteristic.

The simplest choice of a stabilizing controller for (12) is the proportional feedback

$$v(t) = K_p \cdot \Big(r(t) - \theta(t)\Big)\,,$$

where $r(t)$ is the reference and K_p a proportional gain. With such a choice the system (12) becomes

$$\hat{J} \cdot \ddot{\theta} + \hat{b} \cdot \dot{\theta} + K_p \cdot \theta = K_p \cdot r(t) + e(t), \tag{13}$$

which is stable provided that $K_p > 0$. By applying the Laplace Transform to (13), i.e.

$$\frac{\theta(s)}{r(s)} = \frac{K_p/\hat{J}}{s^2 + \hat{b}/\hat{J}s + K_p/\hat{J}}, \tag{14}$$

the dynamics of the mechanical system can be described as a second order transfer function relating the input signal and the output position. A parametric model for this process is given by

$$\hat{G}(s) = \frac{b_1s + b_0}{s^2 + a_1s + a_0}, \tag{15}$$

where $b_1 \approx 0$, $b_0 \approx K_p/\hat{J}$, $a_1 \approx \hat{b}/\hat{J}$, $a_0 \approx K_p/\hat{J}$.

Various experiments varying K_p were performed and the reference signals $r(t)$ were built with summatory of sinusoids with varying amplitude and frequency within the range $[2, 6]$ rad/sec. Two examples of transfer functions obtained with different gains Kp are given below:

$$\hat{G}(s)_{K_p=0.04} = \frac{0.001262s + 2.526}{s^2 + 4.589s + 3.543}\,, \tag{16}$$

$$\hat{G}(s)_{K_p=0.16} = \frac{0.01319s + 26.42}{s^2 + 10.05s + 25.24}\,, \tag{17}$$

where it can be clearly seen that delays and noisy signals make parameter $b_1 \neq 0$, but approximately. The average prediction accuracy of the estimation is 87% and the mean values taken from this estimation are[1]:

$$\bar{\hat{J}} = 0.0085, \quad \bar{\hat{b}} = 0.0554\,.$$

B. Identification of the Hydraulic Plant Model

A number of different input signals can be designed to identify a plant model for the hydraulic actuation, where the target model is given by the fourth order transfer function (11). The nominal input signal u translates proportionally into current applied to the servo-valves, and the output torque (differential of pressure) is calculated from pressure transducers. An example test is given in Fig. 5 and Fig. 6,

[1]Note that these values are not the real physical values of (5), since the input taken into account is current and no hydraulic torque as stated by the model (12).

which depicts the input signal and output torque correspondingly. The reference signal consists of a step with additional random disturbance designed to reveal the valve dynamics. An analysis of the spectral density [4] of the recorded data can reveal the second order properties of the valve [7]. One of the identified transfer function is given by

$$\frac{20.79s^3 - 1696s^2 + 8.005e004s + 1.139e005}{s^4 + 35.68s^3 + 1428s^2 + 1.972e004s + 2.783e004}, \quad (18)$$

with a natural frequency of approximately 32 rad/sec (as seen in Fig. 7), which goes according to the technical data of the valve. The damping factor, which is not directly seen from (18), is an important parameter to be known in advance. This factor gives and idea of how the hydraulic dynamics induces oscillations in the mechanical construction, reducing the performance of trajectory tracking at high velocities.

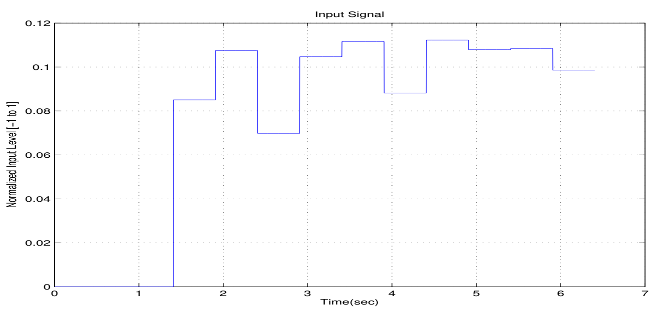


Fig. 5. Recorded input signal $u(t)$.

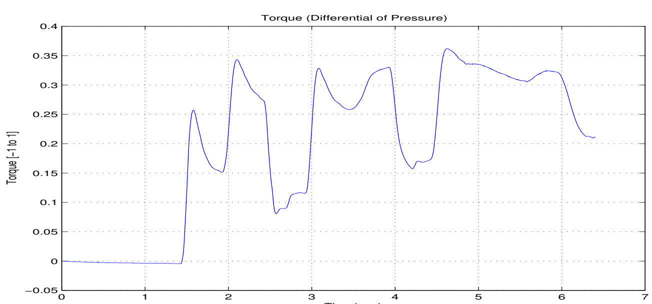


Fig. 6. Normalized differential of pressure response (torque).

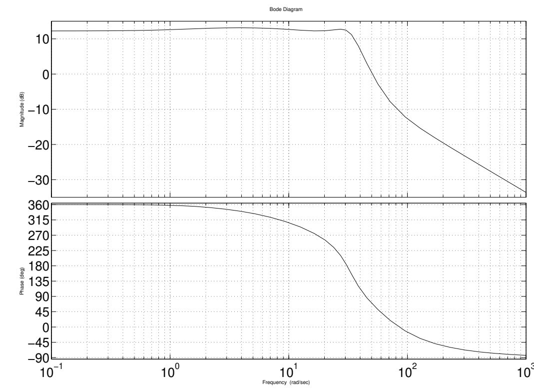


Fig. 7. Bode Diagram of the hydraulic plant identified.

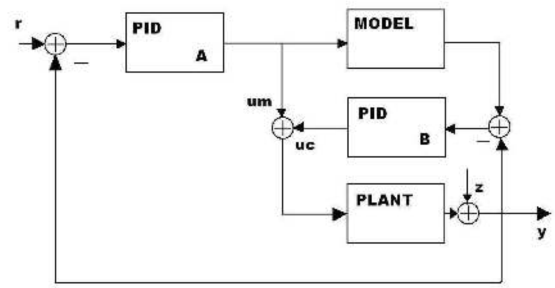


Fig. 8. Block Diagram of the model following control (MFC) structure.

V. Control Design

Two control design methods are chosen for comparison. The first case is a *model following controller* (MFC) designed to track reference trajectories by position as feedback. The second case is a *cascade loop structure* aiming at accurately control the pressure dynamics while tracking reference trajectories. The second, which was proposed in [4], gives better results for stabilizing faster motions and uses both pressure and position measurements.

A. Angular Position Control

Robots may be regarded as multi-body systems, whose outputs are nonlinearly coupled. It is alluring to perform a decoupling of such systems, which theoretically is possible. Commonly, the approaches are based on the inverse dynamic model of the manipulator. As model inaccuracies and other disturbances exist the practical usage is constricted. In the case of hydraulic actuated machines, this coupling becomes more involved due to the delivery of pressure done by a single pump.

To compute current joint torques, the state variables position, velocity, and acceleration are required. The two latter quantities cannot be measured directly. Obviously, these quantities can be derived numerically, but it leads to significant disturbances, which makes the usage rather difficult. A new control structure for position control was presented in [11] and the references therein. The concept is based on the forward model and uses the advantage that the forward model has integral properties. This means that the state variables can be evaluated much more accurately and without any loss of stability. In this method, the current torques are pre-simulated in an independent loop. That results in a two-loop control structure, which can be seen in Fig. 8. In [11] this controller was proved to be robust to parameter variations and disturbances.

The gain of the PID controllers (see Fig. 8) can be tuned by different methods, e.g Ziegler Nichols, optimization, etc. The PID named as A is driven by the process error. This particular solution enables the controllers A and B to interact among each other for effects produced by the system disturbance z. The model applied for such a controller was experimentally found as explained in section IV-A. The friction is compensated by a feed-forward action [4].

B. Pressure Control

Having the SISO linear system (18) opens the possibility to apply different ideas of linear control to accurately track a reference torque. The aim of the controller is to damp down the oscillatory response of the hydraulic actuator while tracking position reference trajectories. A rather simple choice of controller is to apply the classical PID with friction compensation. In order to tune the PID gains an optimization procedure using the error-integral criteria as cost function can be applied:

$$IAE = J(K_p, K_i, K_d) = \int_0^{\infty} |e(t; K_p, K_i, K_d)|\, dt. \quad (19)$$

By minimizing this cost function, the resulting gains for the process (18) identified are:

$$K_{p,hyd} = 0.2, \quad K_{i,hyd} = 5.6, \quad K_{d,hyd} = 0.02.$$

C. Cascade Control

It is expected that by an appropriate control of the hydraulic force/torque, it is possible to counteract oscillations more directly than via measurements of angular positions. In order to combine the design techniques presented in Sections V-A and V-B a cascade control as shown in Fig. 9 was implemented. The aim of the cascade control scheme, is to use pressure measurements to compute the actual acting force/torque generated by the actuator, and reduce oscillations while stabilizing the desired torque along the motion.

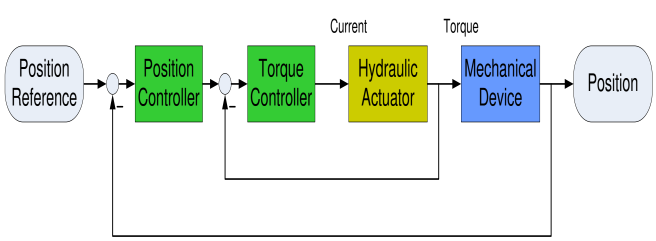


Fig. 9. Two stages cascade control.

In Fig. 9 the outer loop controller calculates the reference torque τ_{ref} needed to drive the manipulator along the predefined joint trajectory θ_{ref}. The inner loop controller takes this torque reference τ_{ref} and computes the servo-valve input current u needed to make the true torque τ asymptotically track τ_{ref}. Since τ asymptotically tracks τ_{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}.

VI. Experimental results

A. Angular position control

Fig. 10 shows the performance of the controller explained in V-A together with a feed-forward friction compensation. The response of the hydraulic torque is also depicted in Fig. 10(b). Although the tracking to a reference signal is somewhat accurate (maximum error of 0.15 radians), the hydraulic torque shows clearly a non-smooth behavior. In the case of the end effector, nevertheless, hydraulic oscillations are not dramatic. In the case of different other links, however, these oscillations are propagated to the mechanical plant provoking a lose of efficiency at high speed.

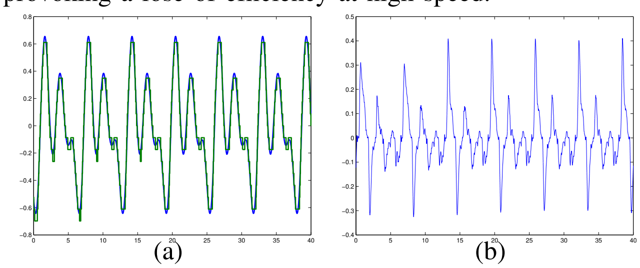


Fig. 10. (a) Reference signal and gripper position superimposed; (b) The resulting normalized torque.

B. Pressure control

As an example of performance of the controller described in V-B, Fig. 11 shows the tracking of the hydraulic torque according to some reference signal that is composed by a summatory of sinusoidal signals of different frequencies. The mismatch is almost unseen (around 8%), which supports that accurate hydraulic torque control can be achieved, despite of model uncertainties. More robust control methods can be designed as discussed in [4].

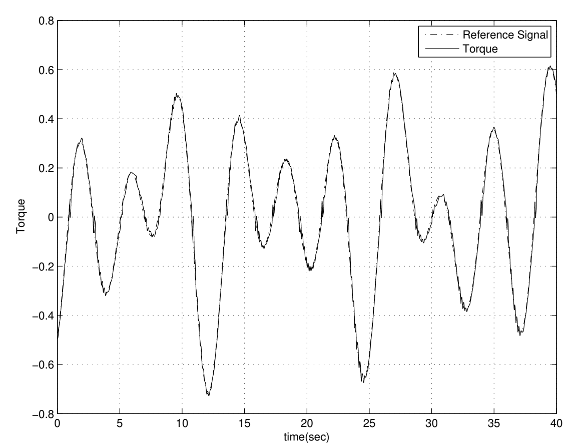


Fig. 11. Tracking of the hydraulic torque (normalized to the range $[-1, 1]$) to a Reference Signal.

C. Cascade Control Algorithm

Fig. 12 shows the same experiment as Fig. 10, when cascade control has been applied. As seen from Fig. 13 and Fig. 10(b) an improvement of the torque smoothness is achieved. The tracking of the signal is improved as well, having a maximum deviation of 0.05 radians.

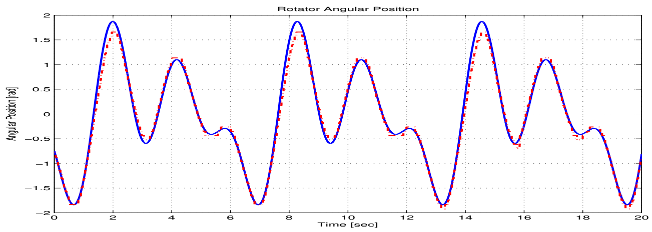


Fig. 12. Two stages cascade control.

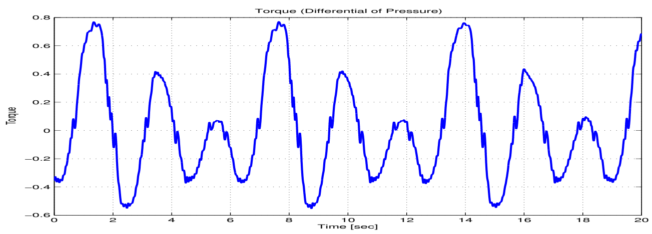


Fig. 13. Two stages cascade control.

VII. Conclusions and future work

The primary target of this article was to show experimental results of system modeling and control design for the case of a hydraulic rotary actuator used at the end effector of forestry cranes. It has been demonstrated that our sensing device solution of low resolution allows us to control this link satisfactorily well by the help of pressure measurements. For this purpose a model of the system was identified throughout data driven modeling. A cascade controller was design, which is used for accurate torque control and damping of oscillations produced by the hydraulic dynamics response. The internal torque controller allows us to achieve faster motions, which is not feasible by the application of a single loop controller.

There is much that can be done to continue this project. At present, this sensing device solution showed very promising results and there is an intention to include cameras as part of the sensing devices. This inclusion would help to correct the bias that is provoked due to the rough working environment, i.e. hitting the ground, trees, etc.

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