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Postprint

This is the accepted version of a paper presented at *25th IEEE/RSJ International Conference on Intelligent Robots and Systems (IROS)*, OCT 07-12, 2012, Algarve, Portugal.

Citation for the original published paper:

La Hera, P., Rehman, B., Morales, D. (2012)

Electro-hydraulically actuated forestry manipulator: Modeling and Identification.

In: *2012 IEEE/RSJ International Conference on Intelligent Robots and Systems (IROS)* (pp. 3399-3404). New York: IEEE conference proceedings

IEEE International Conference on Intelligent Robots and Systems

<http://dx.doi.org/10.1109/IROS.2012.6385656>

N.B. When citing this work, cite the original published paper.

Permanent link to this version:

<http://urn.kb.se/resolve?urn=urn:nbn:se:umu:diva-72717>

Electro-hydraulically actuated forestry manipulator: Modeling and Identification

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Abstract—We present results of modeling dynamics of a forestry manipulator, in which we consider its mechanics, as well as its hydraulic actuation system. The mathematical model of its mechanics is formulated by Euler-Lagrange equations, for which the addition of friction forces is straightforward. Dynamics of the hydraulic system is modeled upon first principle laws, which concern flow through orifices and fluid compressibility. These models lead to a set of equations with various unknown parameters, which are related to the inertias, masses, location of center of masses, friction forces, and valve coefficients. The numerical values of these parameters are estimated by the use of least-square methods, which is made feasible by transforming the models into linear representations. The results of simulation tests show a significant correspondence between measured and estimated variables, validating our modeling and identification approach.

I. INTRODUCTION

Modern forestry operations use state-of-art systems and high-tech machinery to meet the mechanical and engineering challenges of harvesting and logging trees in a safe and environmentally responsible manner. The cranes of these machines are an especial type of mechanical manipulators, which employ hydraulic actuation to produce motions. They are engineered to be manually maneuvered by joysticks, having the human operator as the control unit. During daily work, the driver is demanded to perform various tasks simultaneously, which include visualization, recognition, selection, controlling the crane, and positioning the vehicle. This level of work demand is stressful, since the operator has to deal with an excess of information, and take various decisions at high pace.

As an attempt to support human drivers, and in view of increasing the machine productivity [1], forestry based industry and researchers have analyzed the possibility to automate the routine motions performed in these tasks [2], [3]. The forwarder family of cranes used for logging has been established as a benchmark platform to understand the fundamental challenges of designing this (semi) automatic solution. From the industrial point of view, crane manufacturers have entrepreneur the design of forwarder cranes that can support the implementation of feedback control systems [4]. Researchers, on the other hand, have provided different solutions regarding automatic control design, study of human crane operation, and trajectory planning techniques. Many of

these concepts have been experimentally validated in modern laboratories dedicated to the development of future forestry technologies [5], [6], [7], [8].

Formally, forwarder cranes can be regarded as electro-hydraulically actuated mechanical systems. Theoretically, control design for such systems has proven to be challenging, due to the complex nonlinear nature of the process dynamics. In literature, we recognize the establishment of two well defined brands of development for these systems. On one hand, there exist a vast number of publications with quite mature content regarding modeling and controller design for hydraulic servo systems. Such terminology is attributed to the interconnection between valve and cylinder (or rotor), and the main objective is to control the actuator dynamics, e.g. [9], [10], [11], [12], [13], [14]. On the other hand, we have hydraulic manipulators, for which mainly model-free controllers, resembling conventional decentralized joint control, are found [15], [16], [17], [6], [8], [18], [7]. This class of controllers are usually desirable due to their simplicity of implementation. However, experimental studies reveal that their efficiency is limited to joint trajectories with slow velocity profiles [17]. As the speed of motion reaches high (human) profiles, or when the payload changes dramatically, they lack of damping capabilities and stability.

To attain improving performance, it is of interest to apply model-based motion control strategies, employing models of robot kinematics and dynamics [19]. Within this context, modeling and parameters identification are key elements to realize such a design. Therefore, the initial interest is to verify that, despite the complex mechanical structure and hydraulic hardware, standard engineering procedures for robot modeling and parameter estimation are applicable for describing and simulating dynamics of these machines. The following presentation can be regarded as a technical report of results along these lines, which presents a detailed review of our experience modeling dynamics of the mechanics of a forwarder crane and its experimental validation.

II. MODELING

The manipulator used for our study is a downsized version of a typical forwarder crane¹, but similar in configuration and

¹All dimensional parameters required in all the coming set of equations are reported in [15].

dynamics (see Fig. 1). This machine has been fully equipped to realize various experimental studies, as reported in [20].

Fig. 1. Laboratory crane installed at the Department of Applied Physics and Electronics, Umeå University.

A. Dynamics modeling

We apply the Euler-Lagrange equation as a formalism used to systematically describe the robot dynamics [19], [21], [22]. Based on such a procedure, the computation of the Euler Lagrange equations of motion leads to a second order set of differential equations of the form

$$M(q)\ddot{q} + C(q, \dot{q})\dot{q} + G(q) = B(q)\tau, \quad (1)$$

where $M(q)$ denotes a symmetric and positive-definite matrix of inertias, $G(q)$ the gravity vector, $C(q, \dot{q})$ the matrix of Coriolis forces, and $B(q)$ is a matrix that allocates the external forces to joint torques.

Fig. 2. Crane dimensions and masses description in the sagittal plane.

Considering motions in the plane, as shown in the schematics of Fig. 2, the model (1) takes the form

$$M(q_i, \xi) \begin{bmatrix} \ddot{q}_2 \\ \ddot{q}_3 \\ \ddot{q}_4 \end{bmatrix} + C(q_i, \dot{q}_i, \xi) \begin{bmatrix} \dot{q}_2 \\ \dot{q}_3 \\ \dot{q}_4 \end{bmatrix} + G(q_i, \xi) = \begin{bmatrix} \tau_2 \\ \tau_3 \\ F_4 \end{bmatrix}, \quad (2)$$

where $q = [q_2, q_3, q_4]^T$, i.e. the slewing angle q_1 is not considered, and ξ represents the crane inertial parameters. In addition, and assuming no frictional forces, the set of equations in (2) can be transformed and represented linearly in the elements of the base parameters ξ that constitutes the minimum set [19], i.e.

$$\varphi(q(t), \dot{q}(t), \ddot{q}(t)) \cdot \theta = \tau, \quad (3)$$

where $\varphi(\cdot)$ denotes the regressor written in terms of the measurable variables $[q, \dot{q}, \ddot{q}]$, $\theta = [\theta_1, \dots, \theta_7]^T$ the minimum set of inertial parameters, which components are

$$\theta_1 = m_{FL} \cdot r_{cm_{FL}}^2 + I_{zz_{FL}} \quad (4)$$

$$\theta_2 = r_{cm_{SL}} \cdot m_{SL} \quad (5)$$

$$\theta_3 = m_{SL} \quad (6)$$

$$\theta_4 = m_{Gr} \cdot r_7 + r_{cm_{TL}} \cdot m_{TL} \quad (7)$$

$$\theta_5 = m_{TL} + m_{Gr} \quad (8)$$

$$\theta_6 = m_{Gr} \cdot r_7^2 + m_{TL} \cdot r_{cm_{TL}}^2 + \dots \\ I_{zz_{SL}} + I_{zz_{TL}} + m_{SL} \cdot r_{cm_{SL}}^2 + I_{zz_{Gr}} \quad (9)$$

$$\theta_7 = m_{FL} \cdot r_{cm_{FL}}, \quad (10)$$

and τ the vector of generalized input torques and forces, which is equal to the right hand side of (2).

B. Modeling friction forces

A model often used to represent friction forces considers the static Coulomb - viscous friction, and it is formulated as

$$\tau_i^F = f_{c,i} \cdot (\dot{q}_i) + f_{v,i} \cdot \dot{q}_i, \quad (11)$$

where $i = \{2, 3, 4\}$, and corresponds to the i th link of the robot, f_v denotes the viscous friction, and f_c the Coulomb friction [23]. Hydraulic systems exhibit appreciable asymmetric friction forces, i.e. the constant parameters used to represent friction change according to the direction of motion, i.e.

$$f_v = \bar{f}_v + \Delta f_v \cdot (\dot{q}), \quad (12)$$

$$f_c = \bar{f}_c + \Delta f_c \cdot (\dot{q}), \quad (13)$$

where the terms with a $\bar{\cdot}$ denote the mean values, and their variational values Δ depend on the direction of motion given by the $(\dot{q})$. Introducing the above equations into the general form (11) yields a more complete model for friction forces:

$$\tau_i^F = \bar{f}_{c,i} \cdot (\dot{q}_i) + \Delta f_{c,i} + \bar{f}_{v,i} \cdot \dot{q}_i + \Delta f_{v,i} \cdot |\dot{q}_i|, \quad (14)$$

which is able to capture the most relevant effects of the nonlinear friction phenomena.

C. Combining the models of dynamics and friction forces

Due to the linearity of the friction parameters in (14), i.e.

$$\tau_i^F = \underbrace{[(\dot{q}_i) \ 1 \ \dot{q}_i \ |\dot{q}_i|]}_{\varphi_i^f} \cdot \underbrace{\begin{bmatrix} \bar{f}_{c,i} \\ \Delta f_{c,i} \\ \bar{f}_{v,i} \\ \Delta f_{v,i} \end{bmatrix}}_{\theta_i^f}, \quad (15)$$

the introduction of frictional forces into the linear model (3) is straightforward. To show this, we form a complete set of friction forces for the three links considered, i.e. $q = [q_2, q_3, q_4]^T$, as follows

$$\tau^F = \underbrace{\begin{bmatrix} \varphi_2^f & 0 & 0 \\ 0 & \varphi_3^f & 0 \\ 0 & 0 & \varphi_4^f \end{bmatrix}}_{\varphi^f} \cdot \underbrace{\begin{bmatrix} \theta_2^f \\ \theta_3^f \\ \theta_4^f \end{bmatrix}}_{\theta_f}, \quad (16)$$

where the zero vectors have dimension 1-by-4. Thus, considering that frictions forces act opposed to the torques in the right hand side of (3), a more complete linear model is given by

$$[\varphi(q(t), \dot{q}(t), \ddot{q}(t)) \quad \varphi_f(\dot{q}(t))] \cdot \begin{bmatrix} \theta \\ \theta_f \end{bmatrix} = \tau, \quad (17)$$

which results by the combination of (3) and (16).

D. Mapping cylinder forces to joint torques

Using the geometry of the machine, the mapping between the motion of the hydraulic cylinders given angles of the joints can be explicitly found. This mapping $x_i = f(q_i)$, can be used for a) deriving the joint torques, and b) computing the velocity of the pistons given angular velocity. To this end, we consider the virtual work principle [19], to define that

$$\tau_i \cdot dq_i = F_i \cdot dx_i, \quad (18)$$

which yields an equality for the joint torque as

$$\tau_i = F_i \cdot \frac{dx_i}{dq_i}, \quad (19)$$

for the links $i = 2, 3$, as required in (2)². The actuators forces (see Fig. 3) can be calculated as [13]

$$F_i = A_{A,i}p_{A,i} - A_{B,i}p_{B,i}, \quad i = \{2, 3, 4\}, \quad (20)$$

where $A_{A,i}$ and $A_{B,i}$ denote the areas of chambers A and B respectively, and $p_{(\cdot)}$ the measurements of their corresponding pressures, which in the machine are available through pressure transducers.

Fig. 3. Hydraulic single-rod cylinder [13].

²Recall that the telescopic displacement q_4 is linear, and therefore only its cylinder force F_4 is needed in the dynamics (2)

E. Modeling the hydraulic actuation system

In this machine, the hydraulic servo system consists of a directional proportional control valve, and differential single rod cylinders used as actuators to produce forces. Each piston's motion can be controlled by regulating the oil flow rates $Q_{A,i}$ and $Q_{B,i}$ in each chamber, which mathematical approximation is given by [13]:

$$\left. \begin{aligned} Q_{A,i} &= c_1 u_i \sqrt{P_s - P_{A,i}} \\ Q_{B,i} &= -c_2 u_i \sqrt{P_{B,i} - P_r} \end{aligned} \right\} u_i \geq 0 \quad (21)$$

$$\left. \begin{aligned} Q_{A,i} &= c_3 u_i \sqrt{P_{A,i} - P_r} \\ Q_{B,i} &= -c_4 u_i \sqrt{P_s - P_{B,i}} \end{aligned} \right\} u_i < 0 \quad (22)$$

where $i = \{2, 3, 4\}$, and corresponds to either of the links, and u_i is the electrical control input. P_s and P_r are the supplied and return pressures respectively, while $P_{A,i}$ and $P_{B,i}$ are the chamber pressures in the hydraulic cylinders, as shown in Fig. 3. The valve coefficients are denoted by c_1 , c_2 , c_3 and c_4 , and they are related to the physical properties of the valve, which numerical values are usually not provided by the manufactures. The hydraulic pressures at both chambers can be defined by the differential equations:

$$\dot{P}_{A,i} = \frac{\beta}{(V_{hA,i} + A_{A,i}x_{p,i})} (Q_{A,i} - \dot{x}_{p,i}A_{A,i}), \quad (23)$$

$$\dot{P}_{B,i} = \frac{\beta}{(V_{hB,i} + A_{B,i}(s_i - x_{p,i}))} (Q_{B,i} + \dot{x}_{p,i}A_{B,i}),$$

with $V_{hA,i}$ and $V_{hB,i}$ being the volumes of oil on each chamber when the cylinder is fully closed, β the fluid Bulk modulus, s_i the cylinder's maximum length, and $x_{p,i}$ the piston position, which is calculated according to:

$$x_{p,i} = (x_i - x_{m,i}), \quad (24)$$

where x_i is the current length of the hydraulic cylinder, and $x_{m,i}$ its minimum length. The linear displacements of the cylinders are derived in [20], from which the piston's linear velocity can be calculated as³:

$$\dot{x}_{p,i} = \dot{x}_i = \frac{\partial x_i}{\partial q_i} \dot{q}_i. \quad (25)$$

The force produced by the hydraulic actuator can be calculated as (20), which time derivative is:

$$\dot{F}_i = \dot{P}_{A,i}A_{A,i} - \dot{P}_{B,i}A_{B,i}. \quad (26)$$

Substituting (23) into (26) allows to define that:

$$\dot{F}_i = \beta \dot{x}_{p,i} \left(\frac{A_{A,i}^2}{V_{A,i}} + \frac{A_{B,i}^2}{V_{B,i}} \right) + \beta \left(\frac{A_{A,i}}{V_{A,i}} Q_{A,i} + \frac{A_{B,i}}{V_{B,i}} Q_{B,i} \right), \quad (27)$$

where

$$\begin{aligned} V_{A,i} &= (V_{hA,i} + A_{A,i}x_{p,i}), \\ V_{B,i} &= (V_{hB,i} + A_{B,i}(s_i - x_{p,i})). \end{aligned}$$

³The telescopic link q_4 is directly proportional to its piston's displacement.

Furthermore, replacing (21)-(22) into (27) defines the forces according to the direction of motion:

$$\begin{aligned}\dot{F}_i^+ &= -\beta \dot{x}_{p,i} \left(\frac{A_{B,i}^2}{V_{B,i}} + \frac{A_{A,i}^2}{V_{A,i}} \right) + \beta \psi_1(u_i, P_{A,i}, P_{B,i}), \\ \dot{F}_i^- &= -\beta \dot{x}_{p,i} \left(\frac{A_{B,i}^2}{V_{B,i}} + \frac{A_{A,i}^2}{V_{A,i}} \right) + \beta \psi_2(u_i, P_{A,i}, P_{A,i}),\end{aligned}$$

where $(\cdot)^\pm$ represents positive or negative input signal, and

$$\begin{aligned}\psi_1 &= \left(\frac{c_1 u_i \sqrt{P_s - P_{A,i}} A_{A,i}}{V_{A,i}} + \frac{c_2 u_i \sqrt{P_{B,i} - P_r} A_{B,i}}{V_{B,i}} \right) \\ \psi_2 &= \left(\frac{c_3 u_i \sqrt{P_{A,i} - P_r} A_{A,i}}{V_{A,i}} + \frac{c_4 u_i \sqrt{P_s - P_{B,i}} A_{B,i}}{V_{B,i}} \right)\end{aligned}$$

such that, the force can be denoted in a compact form as:

$$\dot{F}_i = \underbrace{\dot{F}_i^+ \left(\frac{1 + \text{sign}(u_i)}{2} \right)}_{u_p} + \underbrace{\dot{F}_i^- \left(\frac{1 - \text{sign}(u_i)}{2} \right)}_{u_n}. \quad (28)$$

To define (28) in the linear form:

$$A \cdot X = \Upsilon, \quad (29)$$

we can represent the parameter vector:

$$X = [1/\beta \quad c_1 \quad c_2 \quad c_3 \quad c_4]^T, \quad (30)$$

and consider that,

$$\begin{aligned}z_{1,i}(t) &= -\frac{A_{A,i} u_i(t) \sqrt{P_s - P_{A,i}(t)}}{(V_{h_A,i} + A_{A,i} x_{p,i}(t))}, \\ z_{2,i}(t) &= -\frac{A_{B,i} u_i(t) \sqrt{P_{B,i}(t) - P_r}}{(V_{h_B,i} + A_{B,i} (s_i - x_{p,i}(t)))}, \\ z_{3,i}(t) &= -\frac{A_{A,i} u_i(t) \sqrt{P_{A,i}(t) - P_r}}{(V_{h_A,i} + A_{A,i} x_{p,i}(t))}, \\ z_{4,i}(t) &= -\frac{A_{B,i} u_i(t) \sqrt{P_s - P_{B,i}(t)}}{(V_{h_B,i} + A_{B,i} (s_i - x_{p,i}(t)))},\end{aligned}$$

to define the regression matrix:

$$A = [\dot{F}_i(t) \quad z_{1,i}(t)u_p \quad z_{2,i}(t)u_p \quad z_{3,i}(t)u_n \quad z_{4,i}(t)u_n], \quad (31)$$

and the observation vector as:

$$\Upsilon = -\dot{x}_{p,i} \left(\frac{A_{B,i}^2}{V_{B,i}(x_{p,i}(t))} + \frac{A_{A,i}^2}{V_{A,i}(x_{p,i}(t))} \right). \quad (32)$$

III. ESTIMATION OF MODEL PARAMETERS

Recalling that dynamics of the robot, as well as the hydraulic system, can be linearly written in the form

$$\Phi \cdot \Theta = \Sigma, \quad (33)$$

and that measurements are recorded at each t_i , with $i = \{1, \dots, T\}$, an overdetermined matrix of the form

$$\begin{bmatrix} \Phi(t_1) \\ \Phi(t_2) \\ \vdots \\ \Phi(T) \end{bmatrix} \cdot \Theta = \begin{bmatrix} \Sigma(t_1) \\ \Sigma(t_2) \\ \vdots \\ \Sigma(T) \end{bmatrix}, \quad (34)$$

can be formed for finding an estimate $\hat{\Theta}$ of Θ that fits the model (34). There are various mathematical forms to define this concept, such as the least-square estimate, which is conceptually the value of Θ that minimizes the residual of the vector $|\Sigma - \Phi\Theta|$. This can be formulated as

$$\min \|\Sigma - \Phi\Theta\|^2. \quad (35)$$

The classical least-square estimate has the unique solution given by

$$\hat{\Theta} = (\Phi^T \Phi)^{-1} \Phi^T \Sigma = \Phi^\dagger \Sigma, \quad (36)$$

where $(\Phi^T \Phi)^{-1} \Phi = \Phi^\dagger$ is known as the pseudo-inverse of Φ .

A. Recording data and averaging

In the machine, different trajectories can be realized by decentralized PD feedback gains [17], i.e.

$$u_i = -K_p(q_i - q_i^{ref}) - K_d(\dot{q}_i - \dot{q}_i^{ref}) + F_f(\dot{q}_i), \quad (37)$$

where K_p denotes the values of proportional gains, K_d the values of derivative gains, and F_f is a feedforward term used for avoiding dead-zones present in the hydraulic system. The control signal u_i is the electrical input level to the electro-hydraulic servo valve [17], and the estimation of velocities and accelerations is done by the use of Kalman filtering, as suggested in [24]. The reference trajectories are designed to be periodic, and the data is recorded for various periods, so that it can be averaged to improve the signal-to-noise ratio.

B. Estimation and validation of parameters from (17)

Various trajectories were recorded to evaluate different ranges of motion. Considering (36), a set of parameters $[\hat{\theta}, \hat{\theta}_f]^T$ was found. Some additional data sets were recorded for validation tests, two of which are presented in Fig. 4 - 5. The simulation results are shown in Fig. 6 - 7. The validation shows a comparison of the averaged measured torque $\bar{\tau}_m$, versus the torque computed by the model (17), once the values of the unknown parameters have been estimated. We can see that major dynamics are successfully recovered by the model, with minor uncertainty not captured due to unmodeled dynamics.

C. Estimation and validation of parameters (30)

For simplicity, we present results for the first link, i.e. q_2 . The data used for estimation is presented in Fig. 8. In this figure, the first column shows three different reference trajectories $q_2^{ref}(t)$, at different frequencies. The second column shows the control input (37), while the estimation of (26) is shown in the third column. Applying the set of estimated parameters (30), a comparison of the left hand side of (29), with its right hand side is shown in Fig. 9. These results show agreement between the recorded and the estimated responses of the hydraulic model.

Fig. 4. First example of data used for validation. First row: Average trajectories in [rad]. Second row: Averaged velocity in [rad/sec]. Third row: Estimated acceleration in [rad/sec²]. The x-axis is time in seconds.

Fig. 5. Second example of data used for validation. First row: Average trajectories in [rad]. Second row: Averaged velocity in [rad/sec]. Third row: Estimated acceleration in [rad/sec²]. The x-axis is time in seconds.

IV. CONCLUSIONS

We have presented results on modeling and parameters estimation, which were successfully carried on a manipulator used in forestry machines. It is shown that despite the complex interaction between the hydraulic and mechanical processes of the robot, the system's dynamical response can be described by first principle laws. To model friction forces we have considered a map consisting of Coulomb and viscous friction, which are relevant to describe nonlinear dynamics near zero velocities. The geometry of the machine is used to explicitly map the forces of the cylinders to torques on the joints, and to compute their linear velocities given measurements of angular velocities.

To calibrate the model, and to find reliable approximations of unknown parameters, we apply the conventional least-square method. To this end, we proposed a transformation

Fig. 6. Validation with the first data set from Fig. 4. The plot shows the measured averaged torque (black line) vs. estimated torque (red line), where the two graphs on top are in [Nm], and the bottom graph is in [N]. The x-axis is time in seconds.

Fig. 7. Validation with the second data set from Fig. 5. The plot shows the measured averaged torque (black line) vs. estimated torque (red line), where the two graphs on top are in [Nm], and the bottom graph is in [N]. The x-axis is time in seconds.

of the system dynamics into a linear form, possible due to the linearity of the model parameters. To capture the data, we apply decentralized PD control, which has been designed only for the purpose of system identification.

The results allow to assess the correctness of the estimated parameters, and let us conclude that despite minor differences, the model found is able to capture the most relevant dynamics involved in a motion. This statement is valid for the mechanics, as well as for the hydraulic actuation system.

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Fig. 8. The data used for estimation. First column: Reference trajectories $q_2^{ref}(t)$ [rad]. Second column: Control signal $u_2(t)$. Third column: Estimated derivatives of force $\dot{F}_2$. The x-axis is time in seconds.

Fig. 9. Validation results: Plot shows the recorded responses (red line) vs. the estimated responses (black line). The x-axis is time in seconds.

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