FORCE SENSOR BY DRIVING HYDRAULIC CYLINDERS: IDENTIFICATIONOF INERTIAL PARAMETERS AND TESTS

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Abstract: Force feedback sensors are useful for the planning of robotic digging trajectories. In particular, when combined with force-control algorithms it becomes possible to sense buried objects and to determine the weight of excavated materials. The proposed force sensor system makes use of hydraulic cylinder pressure and thereby measures machine force indirectly. Successful implementation of such an approach will eliminate the need for expensive, direct-force sensors. Measurements of pressures in candidate cylinders were compared with laboratory-measured stroke and force. To measure force in every position and during the motion of a backhoe, kinematic, static and dynamic inertial parameters of the bucket, arm and other links have to be considered. From these data a friction model of the hydraulic cylinders can be developed. The present work involves the determination of these friction parameters for an excavator arm using position measures of the bucket and boom, cylinder pressures and the attitude of the excavator boom. Combined, these represent the "dynamic tare" of system. The work includes a practical approach to filtering regression matrix to do not measure accelerations. The methods allows one to carry out inertial parameters for a full machine, in order to verified design and drawning of costruction machine or to use as input for simulation of machine dynamics.

Keywords: digging robot, force sensor, identification, inertial parameters

Introduction

In previous studies [1,2] static and dynamic analyses of force sensors for digging robots and on-board force-sensing systems for construction machines were presented. The force sensor is based on measurement of pressures in two hydraulic cylinders and angular positions of two links, e.g. with an excavator, the driving cylinders and position of bucket and boom respectively, were measured about an absolute inertial reference point.

Using two cylinders it is possible to compute variable pairs, e.g. force/force-position or force/force-direction, by considering the planar movement of two links. This concept is applicable to many types of construction machines having two or more hydraulic cylinders.

The continuous measurement of soil-bucket interaction force is required to predict interaction force or for real-time planning of bucket trajectories.

Our method of force measurement does not require any additional sensors; it makes use of sensors already available on most construction machinery. It does, however, require new static and inertial parameters for the mechanism links and their respective accelerations and velocities.

Errors in the determination of hydraulic cylinder force by using cylinder pressure measurements are introduced because of the presence of static and viscous frictions. the

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proposed new method includes the determination of these parameters.

We use Newton-Euler notation to describe the parameters affecting the dynamic equations of the excavator arm. The parameters are arranged in regression matrix form (eq 1), while the introduction of a linear filter and convolution theorem allowed us to transform the regression matrix into a new system independent of acceleration of the link. This independence avoids the requirement for measuring accelerations of the links (bucket, stick, etc.).

The present work is focused on experimental implementation of this approach, considering practical integration of sensors and software. Tests and experiments were made using a small size excavator (Figure 1) which was modified to use proportional servovalves and was powered by an electric motor. Parameter identification was done off-line using general purpose PC hardare and computing software.

1. Dynamic Analysis and Filtering

Dynamic analysis of the excavator arm shown in Figure 1 was undertaken using Newton-Euler notation.

Figure 1 shows the arm of small size excavator modified to fit transducers

The following calculations consider the dynamics of a system comprised of a bucket and stick, without considering the contribution of the boom to the overall excavator dynamics.

Considering the angular positions of the bucket and stick, respectively θ_2 and θ_l with reference to the horizontal, and the torques trasmitteed by their driving cylinders τ_2 and τ_l, the Newton-Euler dynamics of these links is shown with the system.

$$\begin{bmatrix} \tau_1 \\ \tau_2 \end{bmatrix} = \begin{bmatrix} I_{11} & I_{12} \\ I_{21} & I_{22} \end{bmatrix}\begin{bmatrix} \ddot{\theta}_1 \\ \ddot{\theta}_1 + \ddot{\theta}_2 \end{bmatrix} + \begin{bmatrix} C_{11} & C_{12} \\ C_{21} & C_{22} \end{bmatrix}\begin{bmatrix} \dot{\theta}_1^2 \\ \dot{\theta}_2^2 \end{bmatrix} \quad (1)$$
$$- \begin{bmatrix} h_{11} & h_{12} \\ h_{21} & h_{22} \end{bmatrix}\begin{bmatrix} m_1 g \\ m_2 g \end{bmatrix} + \begin{bmatrix} l_{11} & l_{12} \\ l_{21} & l_{22} \end{bmatrix}\begin{bmatrix} F_x \cos\alpha_x \\ F_x \sin\alpha_x \end{bmatrix}$$

where

$$I_{11} = I_1 + m_2 r_2 l_1 \cos(\theta_2 + \alpha_2) + m_2 l_1^2 + m_1 r_1^2$$
$$I_{12} = I_2 + m_2 r_2 l_1 \cos(\theta_2 + \alpha_2) + m_2 r_2^2$$
$$I_{21} = m_2 r_2 l_1 \cos(\theta_2 + \alpha_2)$$
$$I_{22} = I_2 + m_2 r_2^2$$

$$h_{11} = r_1 \cos(\theta_1 + \alpha_1)$$
$$h_{12} = r_2 \cos(\theta_1 + \theta_2 + \alpha_2) + l_1 \cos\theta_1$$
$$h_{21} = 0$$
$$h_{22} = r_2 \cos(\theta_1 + \theta_2 + \alpha_2)$$
$$C_{11} = m_2 r_2 l_1 \sin(\theta_2 + \alpha_2)$$
$$C_{12} = m_2 r_2 l_1 \sin(\theta_2 + \alpha_2)$$
$$C_{21} = m_2 r_2 l_1 \sin(\theta_2 + \alpha_2)$$
$$C_{22} = 0$$
$$l_{11} = l_1 \sin\theta_x$$
$$l_{22} = l_1 \cos\theta_x + l_x$$
$$l_{21} = 0$$
$$l_{22} = l_x$$

and I is the inertia matrix, h is the gravity matrix, F_x is the interaction force on the bucket.

This system was arranged in matrix regression form as:

$$\Delta\tau = W(\ddot{\theta}, \dot{\theta}, \theta)\Phi \quad\quad (2)$$

where $\Delta\tau = \begin{bmatrix} \tau_2 & \tau_1 - \tau_2 \end{bmatrix}^T$, and the inertial parameters are defined as (3).

$$\Phi = \begin{bmatrix} I_2 + m_2 r_2^2 \\ I_1 + m_2 l_1^2 + m_1 r_1^2 \\ m_2 r_2 \cos \alpha_2 \\ m_2 r_2 \sin \alpha_2 \\ m_2 l_1 + m_1 r_1 \cos(\alpha_1) \\ m_1 r_1 \sin(\alpha_1) \end{bmatrix} \qquad (3)$$

To solve equation system (2) without the measurement of angular accelerations of links, we had to filter equation (1) linearly. By convolution, the filtered torques become

$$\tau_f = \dot{f} * h + f(0)h - fh(0) + f * g$$

were f is the linear stable filter, $*$ is the sign of convolution,

$$h = \frac{d}{dt}\left(I(\theta)\dot{\theta}\right)$$
$$g = -\dot{I}(\theta)\dot{\theta} + C(\dot{\theta},\theta)\dot{\theta} + G(\theta)$$

In this way a filtered regression form of the arm dynamics can be written as:

$$\Delta\tau_f = W_f(\dot{\theta},\theta)\Phi \qquad (4)$$

where the vector of parameters Φ does not change, since the filter is linear.

Until now we have used the "torque" term, but it is produced by the cylinder force and crank length. The torques are related to hydraulic pressures of the cylinder by the following equation

$$\tau = FL\sin\left(\arctan\left(\frac{f\sin\theta'}{L - f\cos\theta'}\right)\right) \qquad (5)$$

where L is the crank length, f the distance between the link and cylinder pins, and θ' is the angle between those two. This relationship shows that the torque depends on the angle θ' (fig.2), which is related to the θ_i angles of links.

However, it is also possible to use a polynomial approximation of $d\theta_i/dL_i$, which

represents the relationship between the angles θ_i and the rod extensions L_i.

The friction of the hydraulic cylinder is difficult to model and compute, because it depends on many factors, for example the types of seal and load pressure. However the same method of identification of inertial parameters allows for the contribution of friction too.
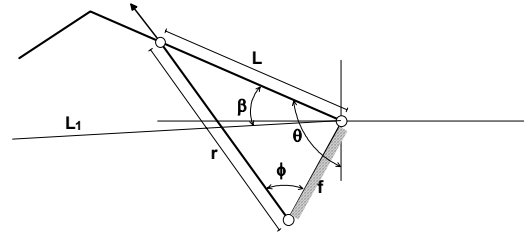

Figure 2: Typical example of mechanism driven by hydraulic cylinder, where r is the variable length of cylinder, L and f the fixed lengths.

From traditional Coulomb friction theory

$$F_f = f_c \, \text{sgn}(v) + f_v v \qquad (6)$$

where v is the velocity of the rod, we have two friction coefficients, the f_c Coulomb friction coefficient and the f_v viscous friction coefficient. Remembering that $\tau d\theta = Fdl$, the previous coefficients form the vector of friction parameters :

$$\Phi_f = \begin{bmatrix} f_{c2} & f_{v2} & f_{c1} & f_{v1} \end{bmatrix}^T \qquad (7)$$

2. Inplementation

To filter the regression matrix a simple stable first order filter was used, having the simple transfer function $f(s) = 1/s + 1$ and the following impulse response in the time domain $f(t) = e^{-t}$.

The regression matrix was convolved by this filter, obtaining the filtered regression matrix, independent of accelerations.

In practice we chose the following transfer function:

$$F(s) = a/s + a ,$$

with a cutting frequency of 20 Hz, a sample frequency of 100 Hz and $a \cong 125 \ rad/s$.

The identification process was based on Least Square Methods considering the measured noise of torque, ε, only.

$$\Delta\tau_f = W_f(\dot{\theta}, \theta)\Phi + \varepsilon$$

Moreover, the components of the torque vector

$$\Delta\tau = \begin{bmatrix} \tau_2 & \tau_1 - \tau_2 \end{bmatrix}^T$$

are decoupled, so every link or equation can be considerd as a single input, single output system to estimate.

The measurement hardware consisted of traditional strain gauges and high pressure transducers. These were used to measure hydraulic pressure in both chambers of the cylinders, two resistive angular position trasducers and a tilt sensor.

One position sensor was placed on the stick-boom pin, the other one was placed on the pin of the crankshaft that transformed the linear motion of rod into angular motion (see fig.1). The angular position sensor location was chosen for practical reasons, so that the relationship between the force produced by the cylinder and the angular position of the bucket was known, based on the kinematics of the bucket mechanism.

The tilt sensor (Accustar) placed on the boom of our small excavator provided the absolute inertial reference to measure the angular positions of links. Additionally it allowed the determination of the boom angular position and velocity.

The signals were acquired by a standard multifunction National Instrument I/O board for PCs and processed using Matlab-Simulink. The same software was used for off-line system identification routines.

As discussed previously, the solution to filter dynamic system and its regression form was chosen to avoid the use of acceleration sensors or the noisy and inaccurate double time derivative of angular displacement. However the regression form needs to measure or estimate angular velocity.

Because the simple method of backward differencing does not yield good performance, the angular velocities were estimated using the Savitzky-Golay filter.

In practice this filter, performing a least squares linear regression fit, is a mobile window of $2nw+1$ number of samples of angular position that is fitted with an m-order polynomial.

$$\theta(t) = p_1 t^m + p_2 t^{m-1} + \cdots + p_m$$

The time derivative or angular velocity is derived at the same time. To avoid high-order calculus the filter parameters were chosen as: $nW=3$ and $m=2$. The velocity is thus given as:

$$v = 2p_1 t + p(2) .$$

Also in this case the filtering was done off-line.

First, static inertial parameters were determined in the static configuration; afterward the fitting of all static, dynamic and friction parameters was conducted during dynamic conditions, that is, by moving the arm of the excavator.

In the first case we used the normal regression matrix because the static conditions do not depend on accelerations, while under dynamic conditions the filtered regression matrix was used.

Many references on identification of dynamic parameters of industrial robots suggest the use of specified trajectories to minimize the conditioning of the data, but in our case it was impossible to plan suitable trajectories that properly simulate a digging cycle under manual control.

3.   Results of estimation

The tests to estimate parameters of the excavator arm can be divided in two parts: inertial static parameters in static condions, and all inertial and friction parameters in dynamic conditions.

3.1   Static estimation

Static estimation was carried out by putting the excavator arm, and in particular the stick and bucket, in various positions, where angular position and torques for each joint was measured with acceleration and velocity of the links equal zero.

This set of measurements was repeated for a number about 150 samples (positions), to have small variation in the results. We avoided those positions corresponding or close to the endstop of the rods.

In this first part we chose to not include the static friction parameters in the estimation, because the static friction depends on many factors including the characteristics of the hydraulic fluid, the type of seal, and the time that the rod remains in the same position, making difficult to have a reliable model of friction.

The results of the static estimation are summarized in Table 1,

i	ϕ_i [kgm]	$\hat{\sigma}_i$ [kgm]
S1	6.17	2.95
S2	12.05	4.45
S3	103.31	23.67
S4	22.78	7.08

Table 1: Parameter estimation of static conditions

where Φ_i and σ_i are the vectors of static parameters and their standard deviations respectively.

3.2   Dynamic estimation

For the estimation of all parameters in dynamic conditions, the signals were sampled at 100 Hz for 40 seconds, converted and processed, to be estimated off-line as described previously.

The final ersults are shown in Table 2:

parameters	Estimated value	Standard deviation
ϕ_{D1}	100.32 kgm^2	25.7 kgm^2
ϕ_{D2}	186.73 kgm^2	33.4 kgm^2
ϕ_{S1}	3.50 kgm	1.54 kgm
ϕ_{S2}	13.71 kgm	3.28 kgm
ϕ_{S4}	71.81 kgm	10.07 kgm
ϕ_{S4}	35.56 kgm	5.68 kgm
ϕ_{f1}	52.23 kgm/s^2	25.5 kgm/s^2
ϕ_{f2}	31444.87 kg/s	1257 km/s
ϕ_{f3}	35.67 kgm/s^2	20.6 kgm/s^2
ϕ_{f4}	30167 kg/s	1320 kg/s

Table 2: Parameter estimation in dynamic conditions

One way to value the quality of estimation is to compare the measured and estimated torque data.   Figure 3 shows the measured and estimated torques related to the bucket, while Figure 4 shows a comparison of stick torques.
By analyzing the residual vector, the difference between measured and predicted torques were computed.   The mean square error and percentage error are given in Table 3.

torque	Mean square error	Percentage error
τ_2	43.12 Nm	26.7 %
τ_1	73.54 Nm	18 %

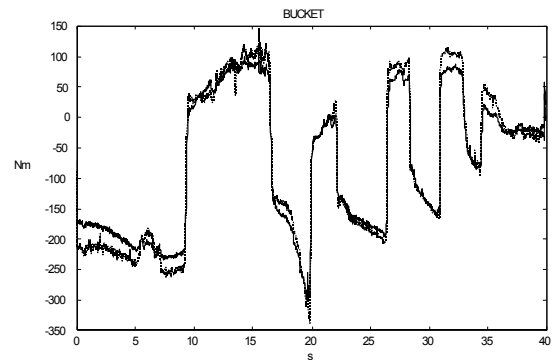Table 3: Mean Square and percentage eerors of stick torques.

Figura 3 Comparison between measured (full line) and estimated (dashed) torques of bucket
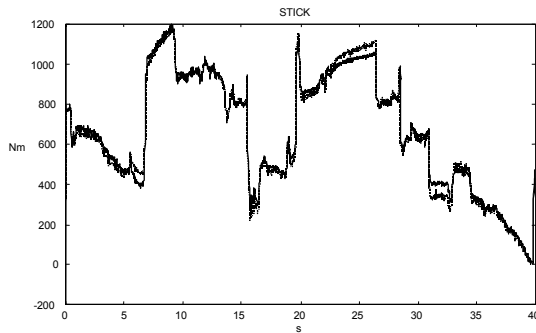
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Figura 4 Comparison between measured (full line) and estimated (dashed) torques of stick

As we can note, the torque of the bucket shows a larger error with respect to the stick torque. This is probably caused by many factors: the bucket motion needs only small forces and low pressures, so the influence of the joint and rod seal frictions will have more uncertain values and the variation is therefore higher; second, the angular position is measured indirectly. Also, the full scale range of the pressure sensors utilized was high in comparison to the pressures needed to actuate the bucket.

4. Conclusions

This work purses two fundamental objectives: first to develop an economical force sensor for contruction machinery; and second, to develop a method to measure the masses, moments of inertia, and friction of parts, when it is not easy to compute these parameters, or to validate models by running mechanical dynamic simulation software.

To achieve these objectives, a small excavator, powered by electric motor and controlled by servo valves, was equiped with a suitable sensor suite. For machine dynamics, a filtered regression approach was studied.

In these first tests we can see that the parameters indentified in dynamic conditions and considering cylinder friction are more accurate with respect to static conditions.

We can note that the estimated torques follow the trend of measured torques, but it is evident that the errors on the torque related to the bucket are large with respect to the error of stick torque.

We were unable during the course of the research to find the reasons for these discrepancies. These parameter values depend on a number of things, from selecting the appropriate pressure sensor range to software routines, hydraulic system configuration, and signal processing techniques.

This complexity requires an accurate calibration of all these factors, and that takes time. The initial results suggest that with further work this technique can be made to work, thus providing an economical sensor approach for control of machinery.

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