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TRAJECTORY CONTROL PERFORMANCE ANALYSIS OF EXCAVATOR-BASED SHEET-PILER SYSTEM

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Abstract: Sheet-piling process poses the problem of prescribing the pile a fixed trajectory in order to drive it correctly into the ground. When representing the complete system of the supporting boom with hydraulic actuators, the gripper and the vibratory unit, the large number of system variables implies the use of assisted control to allow a human operator to handle the operation. Based on exact kinematic transformation from boom state space variables to cartesian workspace ones, a simple control procedure linking boom variables reduces inputs to only one which determines the advance of the pile in the ground. A numerical model representing an actual industrial system has been developed. It is verified that the resulting trajectory is very stable and is leading to small tracking error in following a prescribed trajectory when compared to manual performance in similar conditions.

Keywords: sheet piling, trajectory performance control, electro-hydraulic servo control

1 INTRODUCTION.

The development of modern technology has led to conception and realization of new devices able to enhance significantly human action, especially in fields where heavy and repetitive work is implied. Such is for instance the case for earth moving, digging, and sheet-piling processes where the amount of material to be moved is very large. The system is generally a three-link boom at the end of which is fixed the adapted tool. For sheet-piling, the tool is composed of a gripper for loading the pile and displacing it to the assigned place in right position, and of a vibratory unit for hitting it into the ground, see Fig.1.

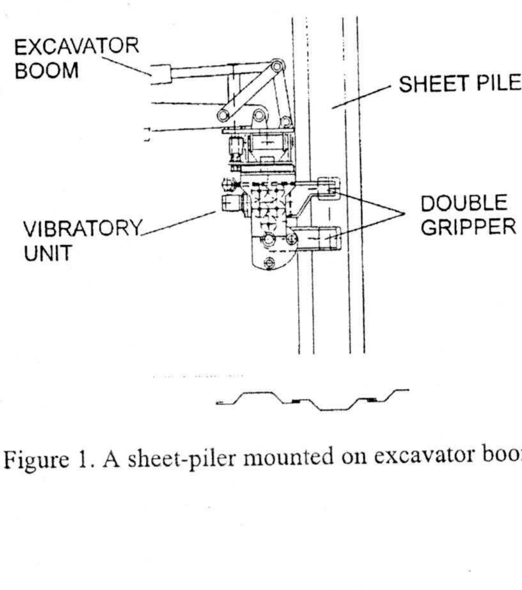

Figure 1. A sheet-piler mounted on excavator boom.

As the system is mounted on a mobile base powered by a diesel engine, its actuation is provided by extending the existing hydraulic power supply to the actuators needed to displace the end effector and to proceed to the piling operation. However the advantage of using such a device with enhanced power effect should be balanced with the larger difficulty of designing properly the system and for a human operator to use it in easy and efficient enough way. This is coming from the too large number of command variables to manipulate simultaneously, and also from the difficulty of human operator to apply the correct control action in full working space especially when the effect of these command variables are strongly coupled in output variables, and the more as the desired output values are themselves uneasy to evaluate, such as, for instance, the displacement along a preassigned direction, see Fig.2.

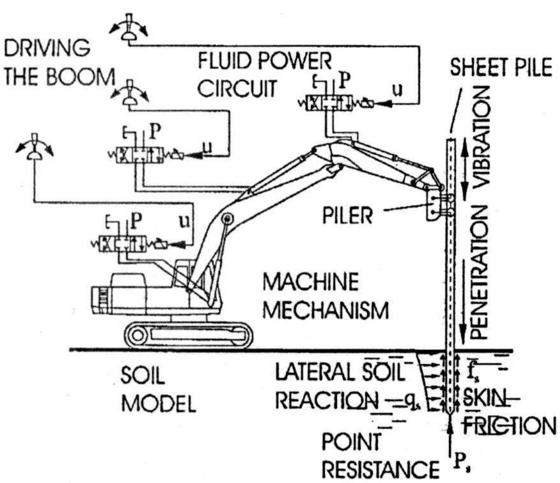

Figure 2. Man-Machine interface in excavator-based sheet-piling.

Then a control has to be worked out to automatically proceed to the operation, or at least to assist the operator in efficient enough way to make the driving routinely affordable and to give the operator usual supervisory action. Motivated by possible industrial applications, the present analysis has been performed to evaluate, for different implementable measurements, the control performance obtained when, for simplicity, the objective of following a straight line is imposed as a desired output. Numerical applications are given for an industrial Kobelco machine using previously developed simulation models[2]. The main result is that operator task is reduced to manipulation of only one driving parameter, while system behavior is very robust when following correctly its trajectory for the various measured parameters.

2. SYSTEM SERVO CONTROL.

Semi-automatic drive system introduced in [1] uses servo control techniques, see Fig. 3, where for safety requirements, the operator keeps a continuous possible action onto actuator movements, and controls manually the main boom actuator while the servo system drives the other segments.

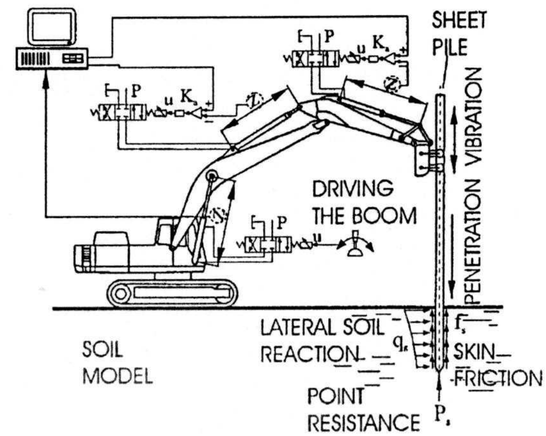

Figure 3. Excavator boom as a servo-manipulator.

Classical hydraulic servo control system consists of sensors, of feedback loops with PID type controllers, and of valves driving the hydraulic actuators, see Fig.4. Sensors in the loop can be measuring actual lenghts z_a^j for each segment j of the boom, or the relative angles ϕ_a^j between adjacent j and $j+1$ segments, or the absolute angles θ_a^j of each segment j with a fixed absolute direction. In any case, the control computer calculates the desired hydraulic actuator position z_d^j using inverse kinematic solution and measured quantities.

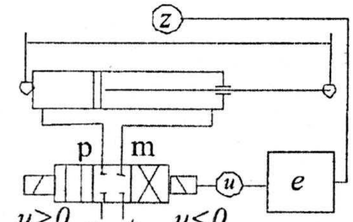

Figure 4. Actuator circuit in electro-hydraulic servo drive.

Due to system errors, there exists a difference

$$e_j = z_d^j - z_a^j \qquad (1)$$

which has to be compensated by the controller. This can be done locally in a decoupled way as long as the motion of the various segments is slow enough with respect to actuator response time. So control inputs are voltage

$$u_j = K_D^j \dot{e}_j + K_P^j e_j + K_I^j \int_0^t e_j \, dt \qquad (2)$$

with respective scalar derivative, proportional and integral gains K^j_D , K^j_P and K^j_I to the valves of corresponding j-th hydraulic actuator.

This produces a displacement of the valve spool generating oil flow in p or m directions given by

$$q_p = \mathrm{sgn}(p_s - p_p) c u \sqrt{|p_s - p_p|} \qquad (3a)$$

$$q_m = -\mathrm{sgn}(p_m - p_t) c u \sqrt{|p_m - p_t|} \qquad (3b)$$

for $u>0$ and

$$q_p = \mathrm{sgn}(p_p - p_t) c u \sqrt{|p_p - p_t|} \qquad (4a)$$

$$q_m = -\mathrm{sgn}(p_s - p_m) c u \sqrt{|p_s - p_m|} \qquad (4b)$$

for $u<0$ respectively, see Fig. 4, with $c = q_{nom} u_{max}^{-1} \Delta p_{nom}^{-1/2}$ obtained by measuring volumetric flow q_{nom} for fixed pressure difference ΔP_{nom} and full input voltage u_{max} . As only the product cu enters eqns (3ab,4ab), it is useful to scale the gains in (2) with this parameter.

3. ACTUATOR SPACE APPROACH.

In order to properly command the system to give the pile a motion along an inclined straight line with angle θ_d with respect to horizontal in the vertical plane of the boom in its geometric workspace, it is necessary to transform intrinsic system coordinates to cartesian absolute ones. Letting l_j the geometric fixed lengths of boom segments, the coordinates of boom end joint P are expressed as

$$\boldsymbol{R} = l_1 \begin{Bmatrix} \cos(\phi_1) \\ \sin(\phi_1) \end{Bmatrix} + l_2 \begin{Bmatrix} \cos(\phi_1 + \phi_2) \\ \sin(\phi_1 + \phi_2) \end{Bmatrix} \qquad (5)$$

shown detailed in Fig. 5.

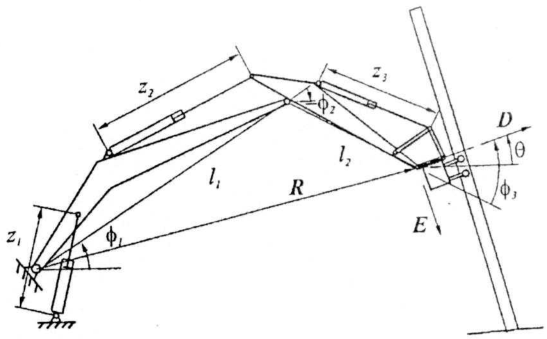

Figure 5. Cartesian path of the boom end.

The corresponding inclination angle θ of the normal to the pile at P given by

$$\theta = \phi_1 + \phi_2 + \phi_3 \qquad (6)$$

should be equal to θ_d first, and subsequent displacement of P along required straight line is expressed by the condition $\boldsymbol{R}_d = \boldsymbol{R}_i + d\boldsymbol{E}_d$ where $\boldsymbol{E}_d = \{\sin(\theta_d) \;\; -\cos(\theta_d)\}^T$ is the unit vector along desired direction, $\boldsymbol{R}_i$ the initial position of the boom end and d is a running feeding parameter. Let $\boldsymbol{D}_d = \{\cos(\theta_d) \;\; \sin(\theta_d)\}^T$ be the vector perpendicular to $\boldsymbol{E}_d$, then $\boldsymbol{R}_d \cdot \boldsymbol{D}_d = \boldsymbol{R}_i \cdot \boldsymbol{D}_d$ takes the form of a classical trigonometric equation giving ϕ_2^d and ϕ_3^d in terms of ϕ_1 and θ_d

$$\phi_2^d = \mathrm{Atan}\,2(b,a) \pm \mathrm{Atan}\,2\left(\sqrt{a^2+b^2-c^2}, c\right) - \phi_1 \qquad (7)$$

with $a = l_2 \cos(\theta_d)$, $b = l_2 \sin(\theta_d)$ and $c = \boldsymbol{R}_i \cdot \boldsymbol{D}_d - l_1 \cos(\theta_d - \phi_1)$.

For fixed θ_d eqn (7) allows to get ϕ_2^d if ϕ_1 is given from measurements, and from eqn (6)

$$\phi_3^d = \theta_d - \phi_1 - \phi_2 \qquad (8)$$

Plots of $\phi_2^d\,(\phi_1)$ and $\phi_3^d\,(\phi_1)$ are in figures 6,7.

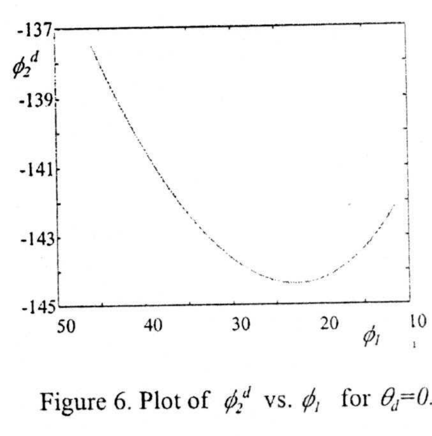

Figure 6. Plot of ϕ_2^d vs. ϕ_1 for θ_d=0.

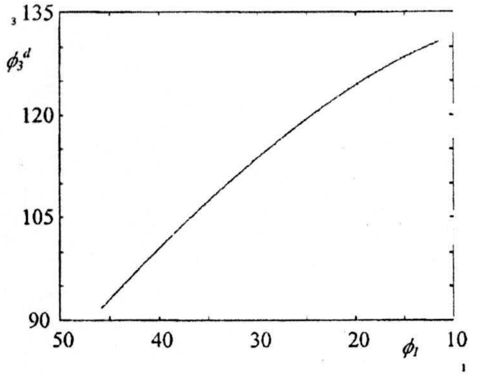

Figure 7. Joint variable ϕ_3^d vs. ϕ_1 for $\theta_d=0$.

This organization dictates the way the system can be operated, in the sense that ϕ_1 has to be parametrically moved for fixed θ_d whereas ϕ_2^d and ϕ_3^d are just obtained as followers, thus reducing the operator task to act ϕ_1 only.

This appears to be the more justified as in the present approach, the operator is guiding the penetration of the pile in the ground while the adjustment of system geometry to comply with piling constraint is taken care of by the previous command. Such a natural splitting is the more convenient as it fundamentally respects the priorities and it can satisfy security requirements. However, account should also be taken of the fact that if the system is now made much easier to command, it may be more unstable, ie uneasier to control, because of inadequation of relative domains in work space covered by system variables, a situation here aggravated by the importance of environment reaction, due to soil structure, onto the piling system. Usual simple response to this difficulty is to allow the gains in controllers to be adjustable by adaptive modification [3]. Here practical experience shows that both damping and stiffness properties of the ground crossed by the pile in lateral direction vary by a large amount as a function of piling depth. Higher gains are allowed with larger penetration with same stability margin. However, as the procedure may be of limited effect, this question raises more fundamentally the problem of the fit of actual system design to the chosen task, showing clearly that both control and design should be clustered into what is now called optimal design.

For completeness, it is necessary to transform the angular representation $\{\phi_1 \;\; \phi_2 \;\; \phi_3\}$ of the system into the more appropriate hydraulic actuator variables, ie the lengths $\{z_1 \;\; z_2 \;\; z_3\}$ of actual actuators position. The solution is depending upon the geometric structure of the link organization. Calculations are performed for a typical revolute joint linking two triangular adjacent elements with given value of the three sides and actuated by a linear drive, see Fig.8.

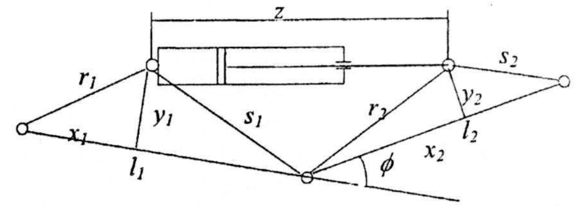

Figure 8. Variables in joint transformation..

One then gets for angle-to-length transformation

$$z = \sqrt{s_1^2 + r_2^2 - 2 s_1 r_2 \cos(\gamma - \phi)} \qquad (9)$$

and for length-to-angle transformation

$$\phi = \gamma - \arccos\left(\frac{z^2 - s_1^2 - r_2^2}{2 s_1 r_2}\right) \qquad (10)$$

where $x_i = \frac{1}{2} l_i^{-1}\left(s_i^2 + l_i^2 - r_i^2\right)$, $y_i = \left(r_i^2 + x_i^2\right)^{\frac{1}{2}}$ and $\gamma = \pi - \arcsin(y_1/s_1) - \arcsin(y_2/r_2)$.

It can be shown that similar but much more complicated transformation pair exist for the bucket joint with the four-bar-linkage mechanism.

4. ANGULAR SPACE APPROACH.

Difficulties in creating reliable control software in the trajectory following problem has led the piler manufacturer to develop more compact formulas, where the angular sensors measuring whether absolute link orientations or relative joint angles could directly control the servo valves without utilizing the angle-to-length transformation (9). In case of absolute ones the system configuration is shown in figure 9.

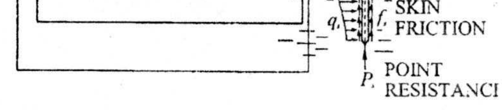

Figure 9. Boom control in angular space.

Suppose we are using now the absolute angles illustrated in figure 10.

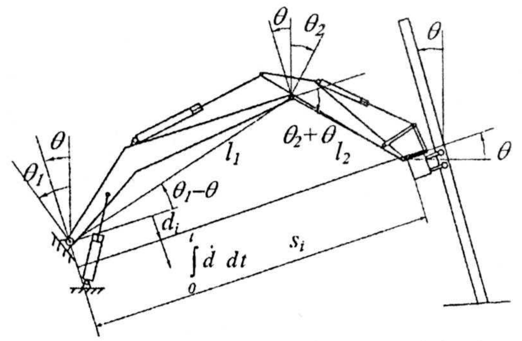

Figure 10. Boom kinematics in space of absolute angles.

Rotating the cartesian coordinate system by angle θ to the left allows to write two conditions related to the requirement of controlling both the rate and the trajectory of feeding motion. The first condition will be satisfied if the desired angular position of the main boom is calculated from

$$\theta_1^d = \arccos\frac{d^2+s_i^2+l_1^2-l_2^2}{2l_1\sqrt{d^2+s_i^2}} - \mathrm{Atan}\,2\frac{d}{s_i} + \theta_d \qquad (11)$$

where the desired position in feeding direction is

$$d = d_i + \int_0^t \dot{d}\, dt \qquad (12)$$

with $d_i = -l_1\sin(\theta_1^i - \theta_d) + l_2\sin(\theta_2^i + \theta_d)$. Secondly, condition to keep the initial normal direction from the joint of the main boom to the end joint constant reads $l_i\cos(\theta_1-\theta)+l_2\cos(\theta_2+\theta)=s_i$ and leads to another relation for the desired bucket boom angle.

$$\theta_2^d = \arccos\left[\frac{s_i - l_1\cos(\theta_1-\theta_d)}{l_2}\right] - \theta_d \qquad (13)$$

where $s_i = l_1\cos(\theta_1^i-\theta_d)+l_2\cos(\theta_2^i+\theta_d)$ has now been fixed to the initial situation indicated by *i*.

What then can be done in the servo loop, is to directly transmit servo errors in angular positions

$$e_j = \theta_j^d - \theta_j \quad ;j=1,2 \qquad (14)$$

and in piler orientation

$$e_3 = \theta_d - \theta \qquad (15)$$

to the valve controllers for governing actuators without using the angle-to-length transformation as shown in figure 11.

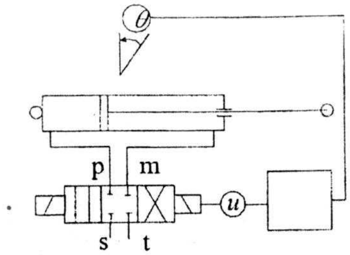

Figure 11. Servo loop in direct angular sensing method.

Advantages of this approach are in the simplicity of the equations but drawbacks arise from the fact that the gains in the controllers depend nonlinearly on system state.

5. PERFORMANCE OF TRAJECTORY CONTROL.

The elements of present model have been added to previously developed simulation model of piler system [1]. They include feedback loops, actuator transformation expressions, controller formulae and inverse kinematic transform, ending up on a 9 solid bodies, 16 joints, 5 hydraulic actuators, 4 proportional valves and 10 hydraulic volumes dynamical model. From this model, system performance can be analyzed under various working conditions and with different input parameters. In particular, the advantage of full nonlinear kinematic representation and the appropriateness of servo system parameters can be tested by the resulting tracking error. Two cases have been analyzed, semi-automatic driving with u_l= *const.* in actuator space and full-automatic driving with $\dot{d}$ = *const.* in absolute angular space. Large servo errors in angular positions as mapped to joint space shown in Fig. 12 at beginning of piling sequence indicate that stability constraints on PID controllers are limiting the gains to too small values for them to adjust to the rapid change in feeding speed of the hydraulic actuators related to the motion of the various boom segments.

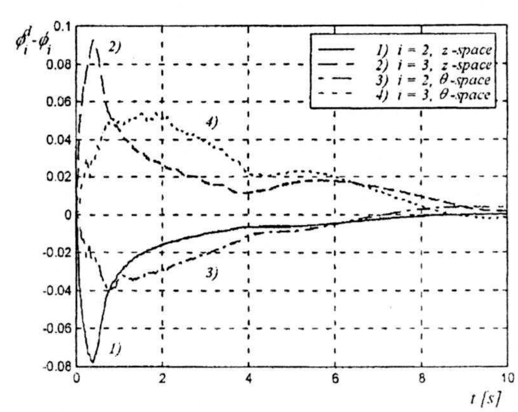

Figure 12. Error in joint angles vs. time.

One way would be to stiffen the PID by changing adaptively the gains as a function of positions but as pointed above the resulting obtained nonlinear controller may not be stable in complete workspace. A more efficient approach is to reconsider system structure itself and redesign basic structure parameters such as the anchoring point of the second segment to reduce large initial length variation.

The corresponding trajectory tracking error in piler orientation θ, see Fig. 13, shows expectable initial large value and also in the course of the piling.

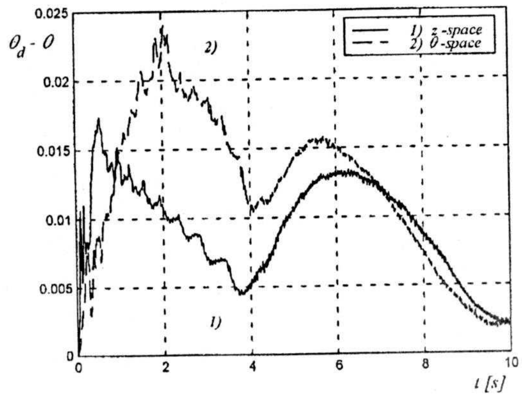

Figure 13. Error in piler orientation vs. time.

This may be related to piling speed as seen on Fig.14. As these errors are related to lateral efforts on the fixture system of the piling effector onto the boom, their minimization would demand the piling speed to be controlled by the operator. However, as the error is uneasy to evaluate, it seems more appropriate to have also a force controller which limits this speed at the same time to a value compatible with the compliance of the fixture system.

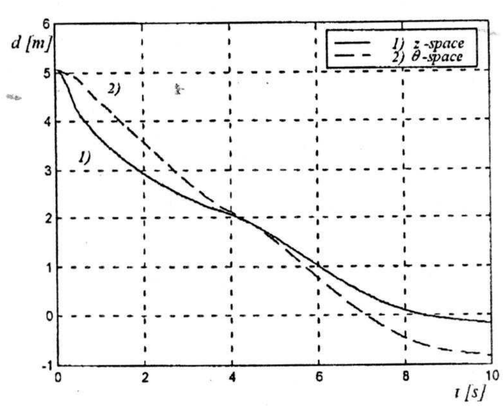

Figure 14. Pile position vs. time.

6. CONCLUSION.

Complete three link plus gripper and vibratory unit sheet-piler system has been modeled including exact kinematic transformation from system to cartesian workspace one, hydraulic actuator dynamics with local PID type controllers and reaction from the ground onto the end effector. Dynamical equations for this system have been analyzed under the realistic constraint of imposing the pile to follow a preassigned straigth line trajectory. Because of the too large number of input variables to manipulate in the case of direct action, a more affordable assisted dynamical situation where there is only one input variable guiding the penetration of the pile into the ground. Numerical application to an industrial Kobelco device shows that if the system can closely track the desired trajectory, there still exists an error due to incompatibility between input feeding speed $\dot{d}$ and actually realized penetration speed leading to difficulties in adjusting the feeding force for optimizing the penetration record. This suggests that previous purely manual feeding control through variable ϕ_1 may not be sufficiently accurate from process dynamical point of view as the operator is only seeing the pile movement, and that more adapted feeding speed control would appropriately improve again the present results.

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