COMPARISON OF CLASSICAL AND MODERN CONTROL APPLIED TO AN EXCAVATOR-ARM

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Abstract: The excavation of foundations, general earthworks and earth removal tasks are activities which involve the machine operator in a series of repetitive operations. Automation is likely to provide a number of benefits such as improving efficiency, quality and safety. However, a persistent stumbling block for system developers is the achievement of fast smooth movement of the excavator arm under automatic control. In this regard, the paper develops two very different design methods, a model-based, full state feedback approach and a classical frequency domain technique based on the Nichols chart. The advantages and limitations of these contrasting approaches are identified in terms of both performance and design effort.

Keywords: robot control; PI controllers; state feedback; identification.

1. INTRODUCTION

The civil and construction industries currently deploy a large number of manually controlled plants for a wide variety of tasks within the construction process, using a range of heavy hydraulic machinery including cranes, excavators, piling rigs and graders. Semi-automatic functions are now starting to be adopted as a means of improving efficiency, quality and safety, particularly when working in hazardous environments.

However, there are still very few real commercially applied examples of site-based intelligent construction robots. In this regard, a persistent stumbling block for system developers is the achievement of adequate smooth movement under automatic control. The control problem is generally made difficult by a range of factors that include highly varying loads, speeds and geometries, together with the issue of multiple hydraulic cylinders being driven by a single pump.

Furthermore, the behaviour of hydraulically driven manipulators is dominated by the nonlinear, lightly damped dynamics of the actuators, in addition to external uncertainties such as the soil-tool interaction for the case of automated excavators. These difficulties are being addressed by researchers using a wide range of classical and modern design methods: e.g. Chiang and Murrenhoff (1998); Ha *et al.* (2000); Budny *et al.* (2003); Gu *et al.* (2004).

This paper concentrates on a laboratory based excavator arm, a 1/5th scale representation of the more widely known Lancaster University Computerised Intelligent Excavator (LUCIE), which is being developed to dig foundation trenches on a building site: Bradley and Seward (1998). The 1/5th scale excavator arm provides a valuable insight into the full sized system and is being used to identify the advantages and limitations of the proposed control approaches. The study considers two contrasting design methods, a model-based full state feedback approach and a classical frequency domain technique based on the Nichols chart. By comparing the approaches in a fair unbiased way, it is anticipated that an improved understanding of the problems associated with this particular plant and of the benefits of the two design techniques will emerge.

2. MOTIVATION

Previous work using LUCIE has demonstrated the feasibility of developing a machine that will accurately dig a trench of specified dimensions. However, control was initially based on the ubiquitous Proportional-Integral-Derivative (PID) type algorithm, tuned on-line in a rather *ad hoc* manner. As a result, the nonlinear joint dynamics would sometimes yield an oscillatory response for bucket position. In order to maintain smooth control, therefore, previous research has utilised a relatively slow control action.

To make automatic excavation viable it is essential that researchers obtain response times that improve on those of skilled human operators – without this, the economic benefits of automation are limited. For this reason, recent research with LUCIE has used Proportional-Integral-Plus (PIP) methods, in order to improve the joint control and so provide smoother, more accurate movement of the excavator arm: Gu *et al.* (2004). Here, Non-Minimal State Space (NMSS) models are formulated, so that full state variable feedback can be implemented directly from the measured input and output signals of the process, without resort to the design and implementation of a deterministic state reconstructor (observer) or stochastic Kalman filter: see Young *et al.* (1987); Taylor *et al.* (2000).

Although normally evaluated in the time domain, in order to compare with PID design this paper also considers the frequency response of the PIP system.

2.1 PIP Control

The PIP controller can be interpreted as a logical extension of conventional PI/PID controllers, with additional dynamic feedback and input compensators introduced automatically when the process has second order or higher dynamics, or pure time delays greater than one sampling interval. However, PIP design has numerous advantages: in particular, its structure exploits the power of state variable feedback methods, where the vagaries of manual tuning are replaced by pole assignment or Linear Quadratic (LQ) design. Over the last few years, such PIP control systems have been successfully employed in a wide range of applications, including construction: Seward *et al.* (1997); Gu *et al.* (2004).

2.2 Classical/PID Control

Although modern design methods, such as NMSS/PIP, are often said to yield performance or robustness benefits, most industrial feedback control systems are nonetheless based on classical methods such as the Nichols chart. There are perhaps two reasons why such frequency domain techniques remain popular in industrial practice. In the first instance, they provide good designs in the face of uncertainty in the plant model. For example, if a system has poorly understood resonances at high frequency, the design can be compensated to alleviate their effects. Secondly, in the absence of a formal model of the system, experimental information can be used directly for design purposes – there is no need for intermediate processing of the data to arrive at a system model. In other words, measurements of the output amplitude and phase of a system exited by a sinusoidal input can be used directly to design the control system. Whilst the wide availability of powerful computers renders this second advantage less important than in the past, the design methods remain extremely effective and popular.

3. CONTROL OBJECTIVES

In order to gain insight into the design problem and to compare the true advantages of PIP over a properly tuned classical controller, the present paper develops and evaluates control systems based on both NMSS/PIP and classical methods. For the preliminary study reported here, the boom angle of the 1/5th scale digger arm is considered. The control objective is to achieve the fastest stable response between specified angles with ideally no overshoot. Furthermore, it is desirable to have integral action to ensure steady state tracking even in the event of load disturbances. Finally, both controllers are implemented with a sampling time of 0.083s, the fastest permitted by the existing experimental set-up.

4. FULL STATE FEEDBACK DESIGN

In order to develop a linear NMSS/PIP control algorithm, a linearised representation of the system is required. Here, the small perturbation behaviour is usually approximated by a linear transfer function model, as follows,

$$y_k = \frac{b_1 z^{-1} + \cdots + b_m z^{-m}}{1 + a_1 z^{-1} + \cdots + a_n z^{-n}} u_k = \frac{B(z^{-1})}{A(z^{-1})} u_k \qquad (1)$$

where y_k is the angle of the boom joint (degrees) and u_k is the applied voltage, expressed as a percentage in the range -1000 to +1000. Positive and negative inputs open or close the boom joint respectively. Here, $A(z^{-1})$ and $B(z^{-1})$ are appropriately defined polynomials in the backward shift operator $z^{-i} y_k = y_{k-i}$. For convenience, any pure time delay of $\delta > 1$ samples can be accounted for by setting the $\delta - 1$ leading parameters of the $B(z^{-1})$ polynomial to zero, i.e. $b_1 \ldots b_{\delta-1} = 0$.

The present research utilises the Simplified Refined Instrumental Variable (SRIV) algorithm to estimate the model parameters: Young (1985). However, an appropriate model structure first needs to be identified, i.e. the most appropriate values for the triad [n, m, δ]. The two main statistical measures employed to help determine these values are the coefficient of determination R_T^2, based on the response error; and YIC (Young's Information Criterion), which provides a combined measure of model fit and parametric efficiency, with large negative values indicating a model which explains the output data well, without over-parameterisation.

Note that these statistical tools and algorithms have been assembled as the CAPTAIN toolbox within the Matlab® software environment and can be downloaded from: www.es.lancs.ac.uk/cres/captain.

It is easy to show that the model (1) can be represented by the following linear Non-Minimal State Space (NMSS) equations,

$$\begin{aligned} \mathbf{x}_k &= \mathbf{F}\mathbf{x}_{k-1} + \mathbf{g}u_{k-1} + \mathbf{d}y_{d,k} \\ y_k &= \mathbf{h}\mathbf{x}_k \end{aligned} \tag{2}$$

where {**F**, **g**, **d**, **h**} are given by Young *et al.* (1987). The $n+m$ dimensional *non-minimal* state vector $\mathbf{x}_k$, consists of the present and past sampled values of the input and output variables, i.e.,

$$\mathbf{x}_k = \begin{bmatrix} y_k & y_{k-1} & \cdots & y_{k-n+1} \\ & u_{k-1} & \cdots & u_{k-m+1} & z_k \end{bmatrix}^T \tag{3}$$

Here, $z_k = z_{k-1} + \{y_{d,k} - y_k\}$ is the *integral-of-error* between the reference or command input $y_{d,k}$ and the sampled output y_k. Inherent type 1 servomechanism performance is introduced by means of the *integral-of-error* state z_k. If the closed-loop system is stable, then this ensures that steady-state tracking of the command level is inherent in the basic design. The control law associated with the NMSS model (2) takes the usual State Variable Feedback (SVF) form,

$$u_k = -\mathbf{k}\mathbf{x}_k \tag{4}$$

where $\mathbf{k} = [f_0 \; f_1 \cdots f_{n-1} \; g_1 \cdots g_{m-1} \; -K_I]$ is the SVF control gain vector. In more conventional block-diagram terms, the SVF controller (4) can be implemented as shown in Fig. 1, where it is clear that it can be considered as one particular extension of the ubiquitous PI controller, where the PI action is, in general, enhanced by the higher order forward path and feedback compensators $1/G(z^{-1})$ and $F(z^{-1})$,

$$\begin{aligned} F(z^{-1}) &= f_0 + f_1 z^{-1} + \cdots + f_{n-1} z^{-(n-1)} \\ G(z^{-1}) &= 1 + g_1 z^{-1} + \cdots + g_{m-1} z^{-(m-1)} \end{aligned} \tag{5}$$

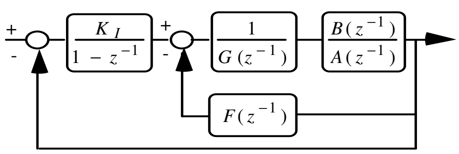

Fig. 1 PIP control block diagram.

However, because it exploits fully the power of SVF within the NMSS setting, PIP control is inherently much more flexible and sophisticated, allowing for well-known SVF strategies such as closed loop pole assignment, with decoupling control in the multivariable case; or optimisation in terms of a Linear-Quadratic (LQ) cost function of the form,

$$J = \frac{1}{2}\sum_{i=0}^{\infty}\left\{\mathbf{x}_i^{\mathrm{T}}\mathbf{Q}\mathbf{x}_i + r u_i^2\right\} \tag{6}$$

where $\mathbf{Q} = diag[q_1 \ldots q_n \; q_{n+1} \cdots q_{n+m-1} \; q_{n+m}]$ is a diagonal state weighting matrix and r is an additional scalar weight on the input. The gains are obtained recursively from the Algebraic Riccati Equation.

4.1 Application to the boom angle

The SRIV algorithm combined with the *YIC* and R_T^2 identification criteria discussed above, reveal that a first order model with two samples time delay provides the best estimated model and most optimum fit to the data across a wide range of operating conditions, i.e.,

$$y_k = \frac{b_2 z^{-2}}{1 + a_1 z^{-1}} u_k \tag{7}$$

where y_k is the angle and u_k is the applied voltage scaled from -1000 to 1000. Here, it should be pointed out that the SRIV algorithm is utilised with a_1 fixed *a priori* at -1, so that only the numerator parameter is estimated. This is because the arm always acts like an integrator, i.e. there is no movement when the input is zero. This assumption is supported by an initial modelling study, which usually returns a_1 close to unity. Furthermore, $b_2 = 0.01915$ yields the best fit for the full range of positive and negative applied voltages. In other words, the linear model (7) represents the ‘average’ behaviour of the nonlinear system, and yields the most robust linear PIP control performance found to date.

In this case, the linear NMSS equations are given by equation (2) with $\mathbf{x}_k = [y_k \quad u_{k-1} \quad z_k]^T$. Finally, trial and error experimentation using step responses predicted using the linear model, suggests that setting $\mathbf{Q} = \mathrm{diag}\,[3000 \;\; 1 \;\; 500]$ and $r = 0.1$ yields a suitably fast control algorithm with no overshoot.

5. CLASSICAL CONTROL DESIGN

Numerous tools exist for classical frequency design, including Bode plots, Nyquist plots and Nichols charts. For the purposes of the present study, the Nichols chart is used to design a PI controller. The chart allows the designer to represent the open-loop gain, phase and frequency data on a single diagram (the latter is implicit, as each point on the chart corresponds to a single frequency).

For a deeper explanation of the chart and associated design techniques, see e.g. pp. 412-417 Franklin *et al.* (1994) or pp. 505-519 Johnson *et al.* (2002). Here it is sufficient to say that the key feature of the Nichols Chart is that, having plotted the open-loop response, it is possible to design closed-loop systems that meet phase and gain margin requirements.

Furthermore, the chart gives an indication of the closed-loop bandwidth and peak gain (the latter two from the contours of constant M or "M-circles").

The design approach taken here involves 4 steps:

1. Collect frequency response date from the open-loop system across a range that covers the frequencies of interest.
2. Plot the data on a Nichols Chart using Matlab/Simulink.
3. Design appropriate compensation for the system to meet the design goals.
4. Implement and evaluate the control response.

Each of these steps is covered in more detail below.

5.1 Frequency Response (steps 1,2)

Due to hardware limitations the controller must be implemented digitally with a maximum sampling rate of 0.083s. This places a limit on the maximum frequency that can be considered for control design. Hence, the frequency range of interest is 0.01Hz up to a maximum of 6Hz (where the latter value is the approximate Nyquist frequency). The frequency response tests consider the mid-point of the boom operating range (25degrees) with the input magnitudes selected to ensure that the boom angle traversed a large part of its range (at low frequencies) and that a measurable output signal was achieved at the higher frequencies.

Due to issues associated with applying a spectrum analyser to the rig, the frequency analysis for this study was undertaken in Matlab/ Simulink on a previously developed non-linear simulation of the plant: see Shaban *et al.* (2004) for details of this model. The frequency response data can be seen on the Nichols chart in Fig. 2.

5.2 Design of a PI Compensator (step 3)

The intent is to design a PI controller taking the following continuous-time form,

$$u = \frac{K_p(T_i s + 1)}{T_i\, s} e \tag{8}$$

where the angle error is defined by $e = (y_d - y)$, while K_p is the proportional gain and T_i is the integral time. For implementation, equation (8) is converted to a discrete-time incremental form, in order to avoid the potential problem of integrator wind-up.

Selection of the proportional gain and integral time aims to ensure stability and provide a fast rise time, whilst avoiding overshoot. To meet these objectives with sufficient robustness, a gain margin (GM) of 6dB and a minimum phase margin (PM) of 60 degrees are chosen. Furthermore, to ensure minimal overshoot, the compensated frequency response should not cross far beyond the 0dB closed-loop gain circle. Hence, the design steps are: (i) increase the gain until gain or phase margin is compromised or the response crosses into the region enclosed by the 0dB closed-loop gain circle, (ii) add integral action at a frequency around 10 times slower than the cross-over frequency.

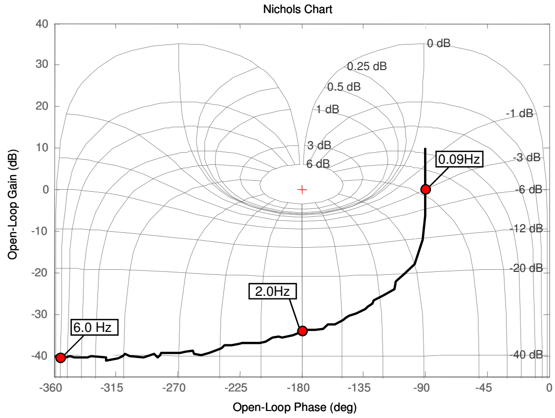

Fig. 2. Open-loop frequency response for boom angle, showing a gain margin of 33dB (at 2Hz) and a phase margin of almost 90degrees at 0.09Hz.

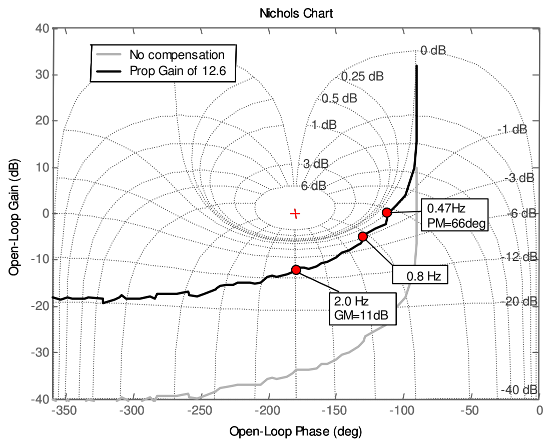

Fig. 3. Open-loop frequency response for boom angle, showing the compensated response (black) alongside the uncompensated system (grey).

However, it is clear immediately from Fig. 2 that any attempt to add integral action will cause crossing of the 0dB closed-loop gain circle. Furthermore the system already presents type 1 performance. Hence only step 1 is followed and the integral gain is set to zero (i.e. the integral time is infinite).

The results are illustrated in Fig. 3, which shows the system with a proportional gain of 12.6 and no additional integral action. It can be seen that the GM is 11dB and the PM is 66degrees, exceeding the specified design criteria by some way. It is also clear that the bandwidth of the closed-loop system will be about 0.8Hz and that the peak gain (which is related to the system overshoot in response to a step input) tends to 0dB only at very low frequencies. These figures suggest a step response of approximately 1.25s with no overshoot.

5.3 Implement and Test (step 4)

The PI controller was implemented in the following discrete-time form with input saturations at +/-1000.

$$u_k = K_p(y_{d,k} - y_k) \qquad (9)$$

The results are discussed in Section 6 below.

6. RESULTS

The response of the (experimental) excavator arm to a step in the command input from an initial angle of –3degrees to a final position of 20degrees is illustrated in Fig. 4. As would be expected, both controllers yield zero error in the steady state (Type-1 tracking). The time to reach the set-point for the PIP controller is approximately 0.8s compared with around 1.2s for the proportional controller. However, there is a small overshoot with the PIP controller that does not appear in the case of the P control.

This system is known to have significant non-linearity that would cause steps of different amplitude to be transiently different. Therefore, it is also important to compare the two control systems over steps of varying magnitude and direction. Typical results of such a test can be seen in Fig. 5.

The figure shows that when responding to steps in the positive direction, PIP control is always fastest to reach the set-point. By contrast, in the negative direction the P control is faster, although in the latter case the P control exhibits a small overshoot.

The PIP design at first responds quickly to a negative change in the set point, before subsequently overcompensating. The latter problem temporarily drives the output away from the set-point by around 30%. This is presently being investigated by the authors: e.g. by adjusting the weighting terms. It is also worth noting that the control input signal is significantly less active in the case of the P control than PIP.

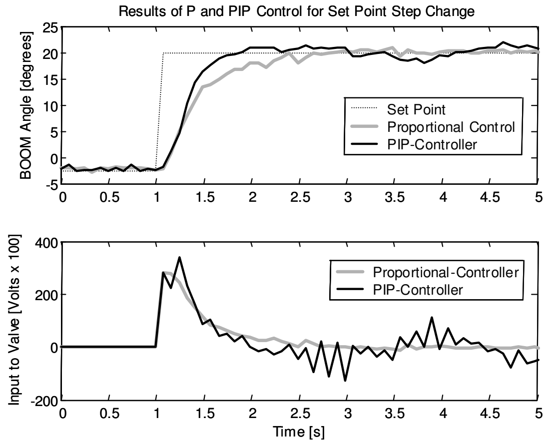

Fig. 4 Experiment showing the response of the P and the PIP controllers to a step change in set point.

In an attempt to improve the understanding of the two approaches, their respective design domains are considered. Figs. 6 and 7 (overleaf) illustrate the frequency domain (classical) and theoretical time domain (PIP) responses. It is clear that, *as would be expected given the same design objectives*, both responses are almost identical. In particular, the gain and phase margin on the Nichols chart (often taken as measures of robustness) are approximately the same. The rise times are also very similar, with the P control predicted to be a fraction faster than the PIP to reach steady state. Interestingly, this latter observation is reversed in the *implementation* results where PIP tends to be faster.

Indeed, the reason for the differences between the two approaches during the implementation results, which are likely to be caused by the different control structures utilised, requires further investigation: a decision as to which control is "better" for this application remains ambiguous.

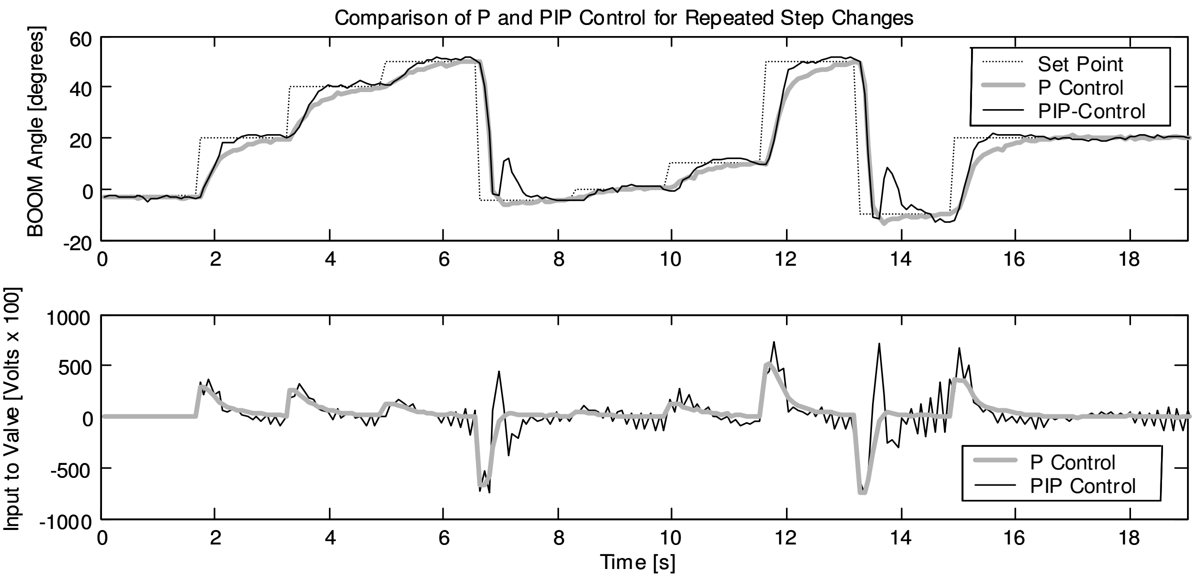

Fig. 5 Experimental results showing the response of the Proportional and the PIP controller to a series of varying amplitude changes to the set point.

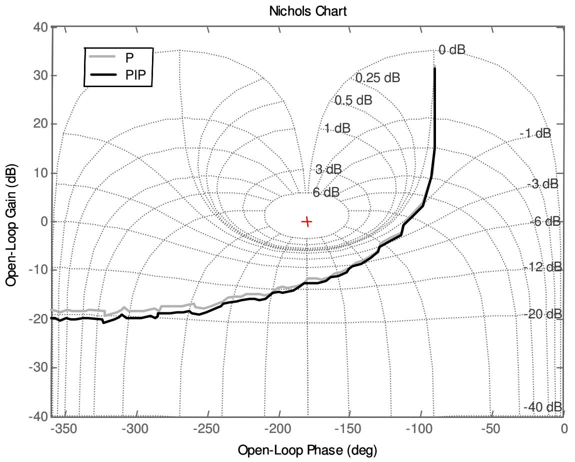

Fig. 6 Frequency response comparison of P and PIP controllers.

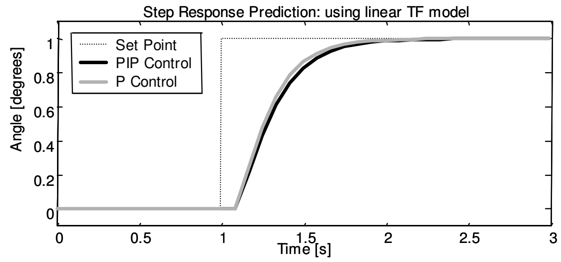

Fig. 7. Predicted Step Response (using ideal linear model of equation 7) comparison of P and PIP

7. CONCLUSIONS

The aim of the research is to provide a balanced comparison between the chosen modern and classical approaches to control system design, when applied to an excavator arm joint. The two control schemes considered are: full state feedback Proportional-Integral-Plus (PIP) control; and P control designed using a frequency domain technique based on the Nichols chart. Both approaches yield acceptable results with little to choose between them.

In general, it might be expected that PIP with (in this case) three gains would have been superior to the P control. This was not found to be the case here. Despite the plant's two sample time delay, it is not clear that the higher order PIP approach offers any significant improvement over simple P control in this case. Note, however, that the results here are limited to boom movement in air – digging (and the large unmeasured disturbances connected with the process) may well alter things. On completion of such experiments, the results will be reported in a future publication.

The excavator arm represents a difficult control problem due to both the demanding design specifications and the plant non-linearity. It may be possible to improve the PIP result with a different approach to tuning the weights in (6), whilst augmenting the classical design with a derivative (or phase advance) action might similarly improve the performance. In addition, a non-linear approach to controlling the arm is currently under investigation (Shaban *et al.* 2004). Naturally the successful approaches must then be extended to the other joints and evaluated for practical digging experiments. Finally, it is intended to repeat the classical design approach and analysis using a spectrum analyser connected directly to the excavator.

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