Control Study with a Pilot Crane

ARTO MARTTINEN, JOUKO VIRKKUNEN, AND RIKU T. SALMINEN

***Abstract*—Computer control of crane operations has been studied in Helsinki University of Technology since 1981, theoretically and experimentally using a pilot scale gantry crane instrumented with new types of sensors. Many classical and modern methods of identification and control have been tested by simulations and experiments, which have given a sound basis for different laboratory exercises. The pilot crane system is a comprehensive and an illustrative environment for studying identification of dynamical systems, for designing controllers of different complexity and for analyzing the behavior of mechanical systems. In this paper we are not describing any specific laboratory exercises, but rather describing the environment and the applications related to the educational purposes.**

I. Introduction

BOOM-BALANCING systems and double pendulums are favorite equipment in control engineering laboratories for demonstration and education purposes. Our choice at Helsinki University of Technology in the early 1980's was, however, a trolley crane. The pilot crane has been extensively used for research and education [6], [7].

The pilot crane offers good possibilities for educational activities and is inexpensive. The students can do experiments with the system and watch its behavior (impossible with "closed" chemical pilot processes). The natural time constants of the pilot crane are small enough to allow different identification and control experiments to be performed during typical laboratory sessions. The system is not too fast either, thus allowing visual perception of dynamic events.

The technical importance of crane control is immediately evident for the students. In contrast to the double pendulum for instance, the crane has complicated dynamics because of the flexible rope. For example, the load position and velocity cannot be measured by elementary means. Also the frame of the crane is not a stiff construction and its resonant modes are interacting with control actions, especially at high speed.

At first we briefly introduce the pilot crane and especially its instrumentation. A complete control engineering design starts from modeling and identification. After checking the validity of the developed model one has to analyze the dynamical properties of the system to be controlled in order to define the ultimate control constraints and possible control demands. Finally, different control methods are to be applied and the performance has to be analyzed. Each stage is the subject of an independent laboratory exercise for students. The contents of the exercises are not expressed explicitly. Rather, the concepts related to the exercises are stressed.

The mathematical model of the crane is presented and some interesting modeling issues are considered. Three different identification methods have been used to obtain a nonparametric frequency response estimation based on: 1) spectrum estimates from RBS experiments, 2) direct gain and phase measurement from sinusoidal input experiments, and 3) a parametric ARX-model estimation [5].

Three different control methods are surveyed. First, we study the minimum-time control strategy which has been one of the favorite subjects in the crane control literature (e.g., [3]). Second, PID-controller design is studied with the root-locus method [10]. As a third method we consider polynomial pole-placement and its adaptive application [4]. The crane is basically a nonlinear system when both hoisting and traversing are considered. For example, the rope length is an important variable of the system. Adaptive or self-tuning control is a necessity.

Several software tools are used in examining system behavior and in designing controllers. For analyzing, identification and design purposes mainly PC-MATLAB is used. SIMNON is used for nonlinear simulations. In some cases MACSYMA has been used to discretize continuous models and to solve equations in symbolic form which are further utilized in real time operations.

The Pilot Crane System

A. The Crane

The pilot crane at Helsinki University of Technology is a scale version (1 : 20) of a 30.5 ton container crane used at shipping ports. By the laws of physics the time-scale ratio of the pilot crane is reduced by $1:\sqrt{20}$. The maximum velocities and accelerations are intentionally higher than implied by the scaling because of the high-speed control goals. The down scaling is based on dimensional theories which give the gantry approximately the same structural and dynamical properties at those of the original crane.

B. Instrumentation and Computer System

The traveling and hoisting of the crane are generated by standard industrial dc-motor drives. In the industrial practice the crane driver gives the reference values of velocity to hoisting and traversing through a joystick. Manual control is also possible in the pilot crane. The velocity references can also be given by a user's application program

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The authors are with Control Engineering Laboratory, Helsinki University of Technology, 02150 Espoo, Finland.
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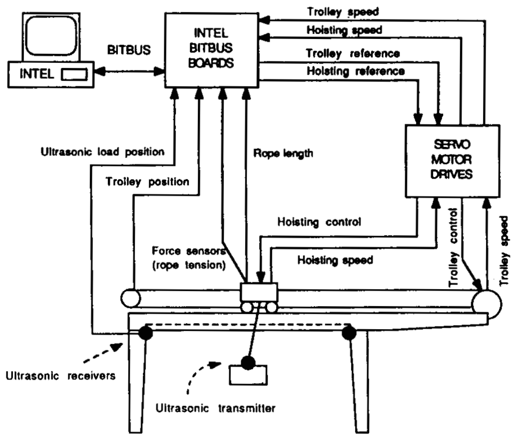

Fig. 1. Pilot crane construction and its instrumentation.

or by another joystick which is connected to the system via the computer.

In the pilot crane there are different types of measurements: hoisting and trolley motor velocities (tachometers), trolley position and rope length (digital absolute angle detectors), trolley and frame accelerations (servo accelerometers), horizontal and vertical rope tension components (strain gauges), and load position (ultrasonic measuring system).

The knowledge of the position of the load (measured or estimated) is prerequisite for feedback control. One way to measure the rope angle is to use the components of the rope tension in the pivot of the rope wheel. The momentary rope angle can be calculated from these force components by using trigonometry. Hoisting causes disturbances to the measurement.

The relative position of the load cannot be calculated reliably from the rope angle measurement in all cases. The rope is not a stiff and weightless stick. The rope angle is therefore a function of the relative load position and acceleration of the trolley. The latter relation is important in high frequency operations which are customary in the fast automatic damping of load swing. Also the elastic waves in the rope become apparent, if the load is not heavy enough and the rope is long.

Another way to measure the relative load position is to use direct acoustic sensors. An ultrasonic transmitter is installed on the load and three receivers on the trolley measure the signals with this from the load. The phase differences define the position changes of the transmitter (incremental measurement). Ultrasonic measurement is not disturbed by hoisting nor by rope vibrations. With this device, the resolution is 2 mm, which is less than that in the tension-based measurement system. The ultrasonic system is initialized by using the extended Kalman filter to calculate the absolute load position from the incremental phase differences.

The pilot crane, measuring instruments, and motor drives are connected to a distributed computer system. The computer system consists of an INTEL-310 microcomputer (80386 20 MHz processor), BITBUS remote controller boards and a BITBUS interconnection between the microcomputer and the remote controller boards. The microcomputer controls the highest level automation while the remote boards take care of the simpler controls and filtering [8]. The microcomputer has iRMX86 real-time multitasking operating system. The programs for communication and control can be written in C, FORTRAN, or PLM languages. The microcomputer is connected to a MS-DOS microcomputer by a serial bus. The control design programs and connections to more efficient computers are located in the MS-DOS microcomputer.

The computer system offers excellent educational possibilities in several fields. A student can write his own control program, e.g., in C language and link it to the rest of the system easily. The effects of sampling and anti-aliasing can be seen clearly. Digital filtering algorithms have been programmed to remote boards with integer arithmetic. The real-time operating system can be configured in several ways for communication, and the control algorithms can be distributed between the micro and the remote boards according to the communication procedures.

III. Crane Model

The model of the crane can be divided into several parts, which can be studied separately or as an aggregate. The mechanical model of the frame characterizes the structural vibrations of the gantry. The basic part of the model is the trolley-load (pendulum) dynamics. The dynamics of the motor drives and industrial controllers can be quite complex. Depending on the dc-motor controllers used the crane can be driven either by an external torque reference (torque-controlled crane) or by an external speed reference of the trolley (velocity controlled, tachometer feedback from the trolley speed).

The model of the crane forms the basis for the control design. The controller is synthesized by combining the model with the control demands. Therefore, for different purposes different models are needed; these are usually obtained by simplification and linearization of the original model. The model is basically nonlinear, if the load is hoisted at varying velocity during transfer. The original nonlinear model is used for simulation and testing purposes.

A. Nonlinear Models and Simplifications

General equations can be derived by Lagrangian mechanics. Assuming that the rope angle ϕ is small we can use the following approximations: $\sin\phi \approx \phi$ and $\cos\phi \approx 1$. Furthermore, assuming that the centripetal acceleration of the pendulum oscillation ($\dot{\phi}^2 L$) is small compared to the gravitational constant g and reducing the rotational inertias of the motor drives into the trolley and load masses, we get

$$m_T \ddot{x}_T = f_T - \phi f_H + f_F(\dot{x}_T) \tag{1}$$

$$L\ddot{\phi} = \frac{1}{m_T}[f_T - \phi f_H + f_F(\dot{x}_T)] - g\phi - 2\dot{\phi}\dot{L} \quad (2)$$

$$m_L\ddot{L} = gm_L - f_H + \frac{m_L}{m_T}\phi[f_T + f_F(\dot{x}_T)] \quad (3)$$

where L is the rope length, m_T trolley mass, m_L load mass, x_T trolley position, f_T force (torque) generated by the motor drive for the trolley, and f_H is the corresponding force for hoisting. The motor drive dynamics are not taken into account here. The friction f_F is mainly caused by the motor drives (it consists of a linear velocity dependent part and a nonlinear dry-friction part). The equations can be further approximated by assuming different special cases, e.g., $\ddot{L} \approx 0$ (constant hoisting speed) or $\dot{L} \approx 0$ (constant rope length).

The linear time-invariant model can only be used if the rope length is fixed. If the hoisting acceleration and speed are low enough, the linearized model can be used in control design assuming that the controller parameters are updated in real time according to rope length changes. Self-tuning and adaptive control have been designed for this case. If the crane is torque-controlled, which is not industrial practice today, nonlinear friction terms must be taken into account. A PID controller with fast analog velocity feedback compensates its effects.

B. Constant Rope Length

Velocity Controlled Crane: If the trolley is velocity-controlled, the rope length is fixed and the rope is assumed to be stiff, the load dynamics can be modelled as a controlled pendulum. Fig. 2 illustrates the case.

With the previous assumptions one gets from the elementary physics

$$m_L\ddot{x}_{La} = m_L g\phi \quad (4)$$

$$m_T\ddot{x}_T = f_T - m_L g\phi \quad (5)$$

where $x_{La} = x_T - x_L$ and $x_L = L\phi$. If the velocity controller is supposed to be ideal, the transfer function between the rope angle ϕ and the velocity reference v_T is

$$\frac{\phi(s)}{v_T(s)} = \frac{1}{L}\frac{s}{s^2 + \omega_0^2} \quad (6)$$

where $\omega_0^2 = g/L$. By using time normalization, $\tau = \omega_0 t$, the model is

$$\ddot{x}_L(\tau) + x_L(\tau) = \dot{v}_T(\tau). \quad (7)$$

Torque-Controlled Crane: The torque-controlled crane is a fourth-order system. The transfer function between the trolley position x_T and the torque reference f_T is

$$\frac{x_T(s)}{f_T(s)} = \frac{1}{m_T}\frac{s^2 + \omega_0^2}{s^2(s^2 + a\omega_0^2)} \quad (8)$$

where $a = m_L + m_T/m_T$. The system equations can be presented in a dimensionless form in many different ways. A simple formulation can be obtained by considering the system in a coordinate system with its origin at the center of mass.

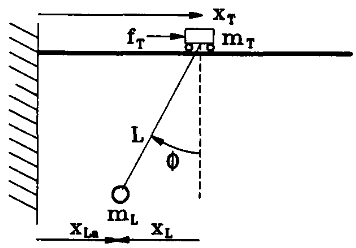

Fig. 2. Ideal pendulum and its notations.

$$\ddot{x}_c(\tau) = a^{-1}u(\tau) \quad (9)$$

$$\ddot{x}_T(\tau) + x_T(\tau) = u(\tau) \quad (10)$$

$$u(\tau) = \frac{f_T(\tau)}{(m_L + m_T)\omega_0^2}. \quad (11)$$

The time normalization is here $\tau = \sqrt{a}\omega_0 t$. Equation (9) describes the dynamics of the center of mass x_c (of the trolley-rope-load system). Actually it is the law of conservation of linear momentum, independent of any kind of internal forces of the trolley-rope-load system. Thus, it is independent of the load suspension. Equation (10) is the model of the pendulum.

Equations (9) and (10) are simple but not so common in control engineering practice. The entire system has four poles on the imaginary axis, two of them at the origin. The two subsystems are dynamically independent, connected only by the control variable u (scaled).

IV. Identification Experiments

The low damping of the oscillatory modes is very evident especially in the frequency domain. The oscillatory modes can also be examined easily in the time domain. Therefore, the pilot crane offers a very efficient environment for studying different estimation methods based on frequency-domain or time-domain considerations.

The simple transfer function model between the input (speed or torque reference) and the output signals (trolley speed and position, load position) can be derived from the dynamical equations. These simple models can then be compared to the results from identification experiments. The flexible frame introduces unmodeled structural resonance vibrations which can be seen clearly from the experimental results.

Three different input-output model identification procedures were applied. The first method is based on spectral analysis where the transfer function estimate is a ratio of two spectral estimates (the cross spectrum between input and output, and the input spectrum).

$$\hat{G}(e^{i\omega}) = \frac{\hat{\Psi}_{yu}(\omega)}{\hat{\Psi}_u(\omega)} \quad (12)$$

As a second method a sinusoidal input signal at the frequency ω_0 was fed into the system and the gain and the phase of the system were calculated from the following correlations

$$I_C(N) = \frac{1}{N}\sum_{t=1}^{N} y(t)\cos\omega_0(t) \qquad (13)$$

$$I_S(N) = \frac{1}{N}\sum_{t=1}^{N} y(t)\sin\omega_0(t) \qquad (14)$$

where N is the length of the data set. Because of the oscillatory nature of the crane system care must be taken with the sinusoidal input signals. The gain and phase can be then calculated from the equations

$$|\hat{G}_N(e^{i\omega})| = \frac{\sqrt{I_C^2 + I_S^2}}{\alpha/2} \qquad (15)$$

$$\hat{\varphi}_N = -\arctan\frac{I_S}{I_C} \qquad (16)$$

where α is the amplitude of the input signal. The drawback of this method is that the experiment must be repeated for every frequency point of interest. But, on the other hand, it gives accurate estimates. Moreover, when the input frequencies are close to the resonance and antiresonance frequencies one can easily see the attributes of these frequencies in time domain.

The third method was the ARX-model identification method which results in polynomial estimates $\hat{A}(q^{-1})$ and $\hat{B}(q^{-1})$ from the equation

$$A(q^{-1})y(t) = B(q^{-1})u(t) + n(t) \qquad (17)$$

for discrete time transfer functions, where n is a noise variable. Both the FFT-based spectral analysis and the ARX-identification were based on the RBS test signals, which does not have necessarily enough power at the antiresonancy frequencies to excite the system dynamics effectively. If the antiresonancy frequencies are known in advance one can easily get a more accurate spectrum by adding sinusoidal input signals at those frequencies. All the RBS-test signals are generated by PC-MATLAB and fed to the crane system via the INTEL microcomputer. Both the spectral estimation and ARX-identification were carried out by *system identification toolbox* routines of PC-MATLAB package.

As an example, Fig. 3 shows the Bode gains of the torque-driven trolley speed in the frequency domain. Comparison of the theoretical response (the steady-state gain is adjusted to the measured ones) with the measured responses indicates that there is no major difference at low frequencies. At higher frequencies the difference is larger due to the structural resonance. Similar frequency domain analysis was carried out for the entire crane system [5]. As can be seen in Fig. 3 the correlation method is the only method, which clearly distinguished the antiresonance (zero gain) point.

V. Control Strategies

The crane system in practice should fulfill several requirements. The transfer of loads must be fast and accurate. The system must adapt to different loads and to changing rope length. The control actions must be reasonably smooth, and not giving extra stresses and fatigue to the mechanical structures. The weighting of the criteria depends on the application. In the following three different control methods are introduced: minimum-time control, a simple PID control and polynomial pole-placement control.

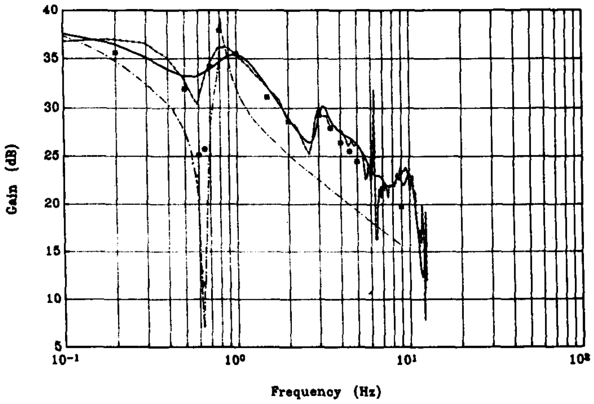

Fig. 3. Bode gain curves of the torque-controlled crane (the output is trolley speed): spectral analysis (dashed lines), correlation method (star), ARX-identification (solid line), theoretical trolley-load model (dash-dotted line). The resonance of the frame can be seen on higher frequencies.

A. Minimum-Time Control

Minimum-time control strategy has been the subject of several studies (see, e.g., [3]). In principle, an ideal crane can start from the initial position and to stop on the final position by this open-loop strategy, but the operation is sensitive to disturbances and, moreover, the control actions are rough, giving rise to extra stresses and fatigue of materials. For educational purposes, however, we can reduce the maximum allowed for trolley speed and torque and define very illustrative optimal control laws.

From (7) we get

$$\dot{x}(\tau) = \begin{bmatrix} 0 & 1 & 0 \\ -1 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} x(\tau) + \begin{bmatrix} 1 \\ 0 \\ 1 \end{bmatrix} v_T. \qquad (18)$$

The minimum-time solution can be derived from the Hamiltonian

$$H[x(\tau), v_T(\tau), p(\tau), \tau] = 1 + p^T(\tau)[Ax(\tau) + bv_T(\tau)] \qquad (19)$$

where A and b are corresponding matrices in (18) and p is the costate

$$\dot{p}(\tau) = \frac{\partial H}{\partial x}[x(\tau), v_T(\tau), p(\tau), \tau] \qquad (20)$$

with the assumptions: initial state $x(0) = [0\ 0\ 0]^T$, final state $x(t_f) = [0\ 0\ x_T^0]^T$ and control constraints $|v_T(\tau)|$

$\leq v_0$. The Hamiltonian is in this case simple

$$H = 1 + p_1 x_2 - p_2 x_1 + p_1 v_T + p_3 v_T \tag{21}$$

Pontryagin's minimum principle indicates that $p^{*T}(\tau) b v_T^* \leq p^{*T}(\tau) b v_T$, thus $(p_1^* + p_3^*) v_T^* \leq (p_1^* + p_3^*) v_T$. If $p_1 + p_3 \leq 0$ then $v_T^* = v_0$, and if $p_1 + p_3 > 0$ then $v_T^* = -v_0$. Because all of the eigenvalues of A have nonpositive real parts, an optimal control exists.

The solution can be described easily on a phase plane using state trajectories (the exact numerical solution is not that easy). The behavior of x_3 is evident ($\int_0^\tau v_T d\tau$). The interesting behavior of x_1 and x_2 are defined by equations

$$\dot{x}_1(\tau) = x_2(\tau) + v_T^*(\tau) \tag{22}$$

$$\dot{x}_2(\tau) = -x_1(\tau) \tag{23}$$

which give the equation for the trajectories

$$\frac{dx_1(\tau)}{dx_2(\tau)} = \frac{x_2(\tau) + v_T^*(\tau)}{-x_1(\tau)}. \tag{24}$$

The (x_1, x_2)-plane solutions of (24) are two groups of circles with centers at $(-v_o, 0)$ and $(v_o, 0)$. Fig. 4 illustrates the phase-plane behavior, when a minimum-time solution has been applied. Switchings occur when the trajectories along the circles intersects. The simulation results are shown in Fig. 5.

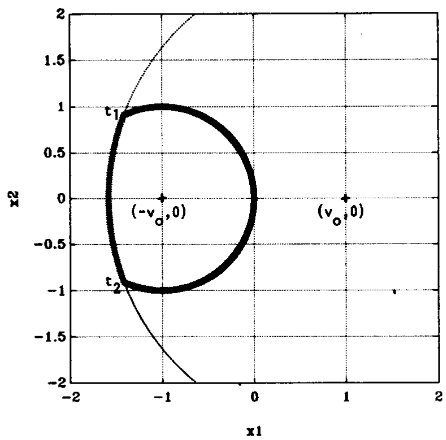

Fig. 4. Time-optimal phase plane trajectories of x_1 and x_2.

B. Conventional PID-Control

A PID-control is not at its best for this kind of batch operation. In cascade control with two *P*- or PD-controllers stability can be found, but the damping is poor and the result is sensitive to the initial conditions of the load and the distance to be run. This is especially true with the torque-controlled crane. With the speed-controlled crane acceptable results can be achieved, which is also confirmed experimentally [10].

The bare swinging can be compensated by a *P*-controller in both the torque- and speed-controlled case. In the torque-controlled crane an unstable system results if position also has to be controlled. Instead, in the speed-controlled crane a cascade control systems (Fig. 6) results in a stable solution. The control system seems to work satisfactorily with pure proportional controllers. The transfer function for the inner closed loop is

$$\frac{x_T(s)}{\phi_r(s)} = K_a \frac{s^2 + \omega_0^2}{s\left(s^2 + \dfrac{K_a}{L} s + \omega_0^2\right)} \tag{25}$$

where ϕ_r is the reference signal and K_a the gain for the inner loop. The controller parameters can be set according to the closed-loop root loci. At first a proper K_a has to be found. Fig. 7 shows the root loci of the inner loop (K_a varies) and the outer loop (K_p varies). A reasonably good damping ratio can be achieved for the closed-loop system.

C. Pole-Placement Control

In observable and controllable systems one can place closed-loop poles anywhere on the complex plane. The robustness and noise sensitivity depends on the controller. To avoid realization problems different methods have been studied. Ackerman's state-space method [1] and Åström's and Wittenmark's polynomial approach [2] have been applied successfully [6], [7].

Mathematical Model Used: Let's consider polynomial pole-placement control for the velocity-controlled crane [4]. In order to control both the swing angle ϕ and trolley position x_T simultaneously, an artificial output $\Omega = x_T + p\phi$ is generated. Thus, the transfer function is

$$\frac{\Omega(s)}{r_T(s)} = K_v \frac{\left(1 + p\dfrac{K_\alpha}{L}\right)s^2 + \omega_0^2}{s(1 + \tau_v s)(s^2 + \omega_0^2)} \tag{26}$$

where r_T is the velocity reference signal given by the computer and p is a weighting factor used as a tuning parameter, τ_v is the time constant of the motor drive. K_v and K_α are gain parameters. If we assume $\tau_v = 0$, then we can scale the time as in (7). Hence, the system equations will be independent of any rope length changes. Therefore, an adaptive control law (gain scheduled) can be achieved either by changing sampling rate or, when using the more accurate model (26), by updating the whole control law according to the rope length changes. In the first case the sampling period should be proportional to $\sqrt{L}$ and some updating of gains are necessary. Both cases work well, as confirmed by experiments.

Pole-Placement Controller: A digital pole-placement controller is used to control the system (26). The discretized model is

$$\Omega(k) = \frac{B(q^{-1})}{A(q^{-1})} r_T(k - 2). \tag{27}$$

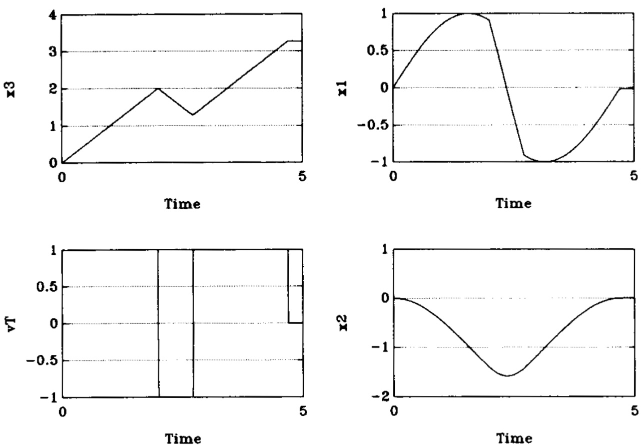

Fig. 5. Minimum-time solution of the load transfer.

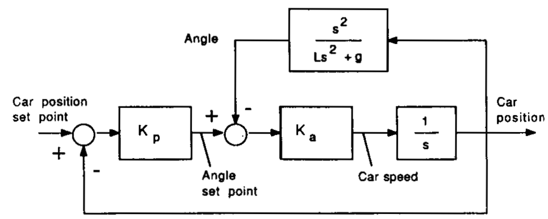

Fig. 6. The cascade control of the car position and the swinging angle.

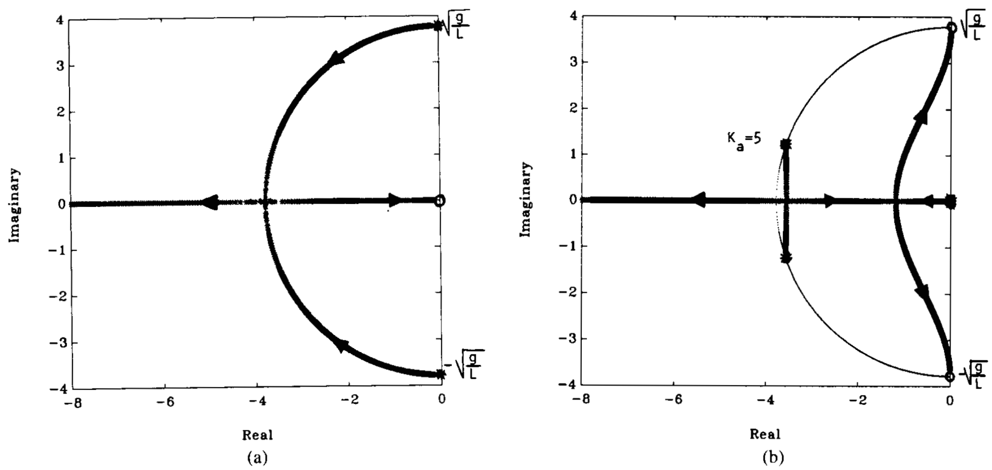

Fig. 7. (a) Closed-loop poles of the inner system as a function of K_a. (b) Closed-loop poles of the cascade-controlled system as a function of K_p when K_a is fixed.

Depending on the choice of p the zeros of B are either on the unit circle or on the real axis. B is divided into two parts $B = B^- B^+$ where B^+ is monic having all its zeros strictly inside the unit circle and B^- has all its zeros on or outside the unit circle. If $p < -(L/K_\alpha)$, then deg B^- = deg $B^+ = 1$, else deg $B^- = 2$ and deg $B^+ = 0$. The controller polynomials R and S can be solved from the Diophantine equation

$$A_M T = AR + B^- S_q^{-2}. \tag{28}$$

The transfer function (26) is discretized and the Diophantine equation (28) solved symbolically and converted automatically to real-time FORTRAN-code using MACSYMA. Symbolic equations can be updated fast in real

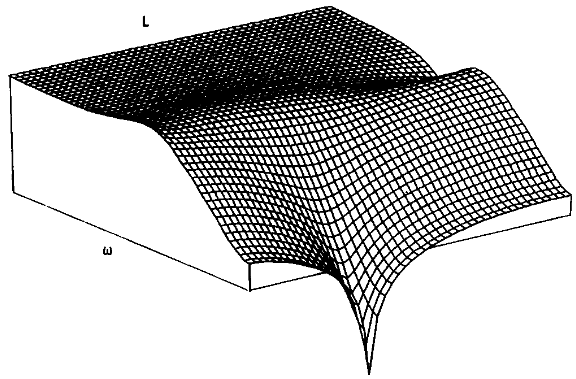

Fig. 8. Bode diagram of the sensitivity (to output noise) function.

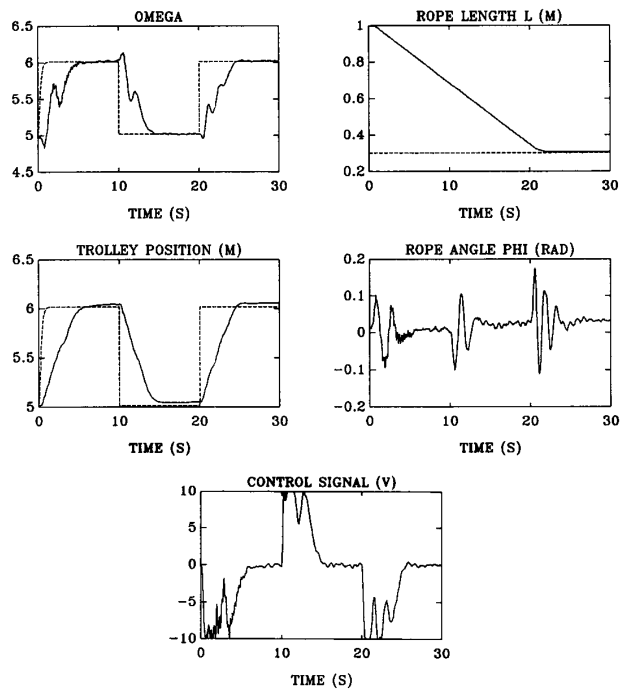

Fig. 9. Control experiment with the pilot gantry.

time. The desired closed-loop denominator A_M and the observer polynomial T are design parameters as well as the sampling time Δt and parameter p. When $p > -(L/K_\alpha)$, the system model will have two dipoles, which will cause serious ill conditioning. When $p < -(L/K_\alpha)$, the situation is much better. The choice of the T-polynomial is crucial for the design. Fig. 8 presents the Bode diagram of the sensitivity function when the rope length varies from 0.2 to 1.2m. Reasonable noise reduction can be achieved with all rope lengths.

Control Results: An adaptive control law is implemented in the INTEL computer. The continuous-time model is discretized symbolically and the corresponding discrete-time model is updated according to the rope length changes. In this sense the adaptation can be considered as a feedforward compensation from the rope length.

Fig. 9 presents an experiment based on the polynomial pole-placement method, where the reference signal is a sequence of step changes (dashed line in "omega"). During the time of operation the rope length decreases from 1 to 0.3m. The nonminimum phase behavior of the output Ω can be clearly seen. Because of the rather coarse rope angle measurement system and the simplified model, there are some remaining oscillations in the steady-state of ϕ and a small bias in the trolley position x_T. The saturations of the control signal (speed reference) $r_T(t)$ prohibit the compensation of ϕ during the acceleration phases.

VI. Conclusion

The pilot crane with its instrumentation is briefly introduced. The crane is well suited for educational experimentation. It offers a uniform environment for a complete control engineering design starting from modeling and ending with tuning of different controllers. The environment is not simple, giving therefore possibilities for demanding laboratory exercises. A versatile pilot crane system can be built at rather low cost.

Instead of specific laboratory work, we described the potential possibilities of the crane environment for practical education of control engineering. The modeling issues were considered and different identification methods were compared.

Different control algorithms have been applied to the load transfer in the pilot crane. The experiments show that the controllers must adapt automatically to the rope length. The paper deals mainly with the velocity-controlled crane, which is a standard in industry. An illustrative open-loop minimum-time control was derived for the load transfer. A simple root-locus method was used to tune a cascade type of PID controller, which worked satisfactorily with fixed rope length. Different pole-placement methods suffered from sensitivity and robustness problems. However, good results were achieved after ill conditioning is understood and avoided. As an example we described an adaptive pole-placement control law which was tested successfully.

Because of the complex nature of the system there are also many other applications which can be tested with the pilot crane. For example different state estimators may be easily applied. Also the effects of the frame flexibility can be studied.

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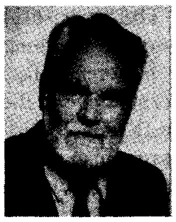
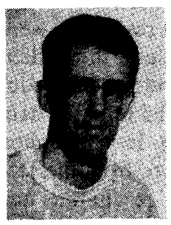
Arto Marttinen received the M.S. degree from Tampere University of Technology in 1982 and the Lich.Tech. degree from Helsinki University of Technology, Otaniemi, in 1987, both in control engineering.

Currently he is finishing the Doctor of Tech. degree in control engineering. From 1982 to 1987 he was a research scientist and since 1987 he has been a laboratory manager at the Control Engineering Laboratory of Helsinki University of Technology. Currently he is on leave, working as a director of Control CAD company. He is a coauthor of two text books on control engineering (in Finnish) and over 30 scientific and technical papers.



Jouko Virkkunen received the M.S. degrees in mechanical engineering and technical physics in 1956 and 1957, respectively, the Lic.Tech. degree in 1960, and the Doctor of Technology degree in technical physics in 1966, all from Helsinki University of Technology, Otaniemi.

He has been an Associate Professor in physics since 1966, and in control engineering since 1974 at Helsinki University. He is coauthor of four books on nuclear engineering and control engineering (in Finnish) and of about 100 scientific and technical papers.



Riku T. Salminen received the M.S. degree in electrical engineering from the Helsinki University of Technology, Otaniemi, in 1989.

He has been a Research Scientist at the Control Engineering Laboratory, Helsinki University of Technology, since 1989. His current interest is robust control of cranes and manipulators.