

# A Comparative Study of Backlash Compensation Methods

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## 1 Introduction

### Abstract

This paper describes three different approaches to compensation of backlash influence in a position-controlled system. The influence of backlash in a cascade structured position controller (method 1) with inner torque control loop is compensated by utilizing a fast linear torque controller, at the expense of having to measure the torque on shaft. Defining the model of backlash (method 2) allows for compensation of backlash influence by superposition of compensation signal. However, this method is more sensitive to measurement noise. Variable structure controller (method 3) (one for linear, and the other for nonlinear regime) proved well, with constraints on input signal. These methods were chosen because they were judged most appropriate with respect to constructional and computational constraints in the controlled system. Excellent system behaviors had been experimentally verified on a laboratory model, when proposed control concepts were utilized.

Compensation of backlash influence is an important issue in many industrial and military applications as well, [10]. The mathematical model of a two-mass servosystem is well known, however its parameters have to be identified for each individual system. The mechanical system, depicted in Figure 1, consists of two inertial masses, characterized by mechanical time constants  $T_{M1}$  (drive) and  $T_{M2}$  (load). Transmission elements possess elastic characteristics, conveyed through stiffness ( $c$ ) and damping ( $d$ ) factors, as well as backlash of  $2\alpha_B$ .

Method of harmonic balance was used for analysis of backlash influence. When performing such analysis, nonlinearity in the system is represented by its describing function [3]; whereas linear part of the system is replaced by a single element with a linear transfer function (Fig. 2). Control is performed utilizing the quasicontinuous-time approach, where the discrete-time control system is replaced by an equivalent continuous-time control system, [15, 11].

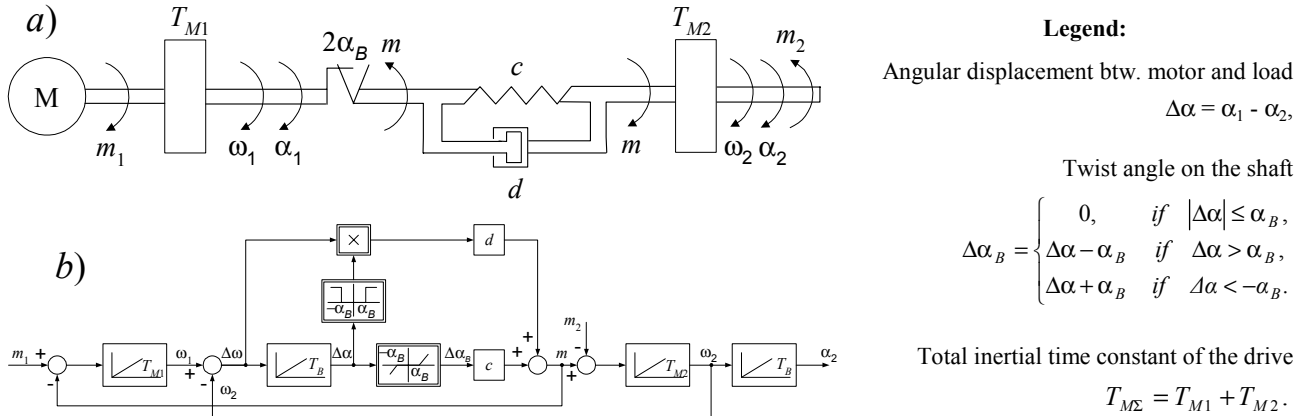


Fig. 1. A two-mass elastic system with backlash: a) system model, b) structural block diagram.

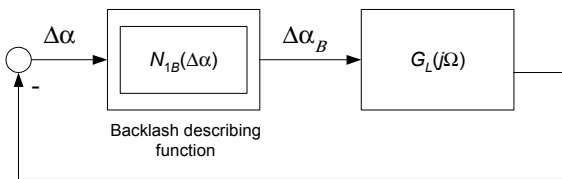


Fig. 2. Block diagram of a control system with nonlinear element, suitable for analysis by method of harmonic balance.

Control is based on full-order state controller, tuned according to the Hurwitz criteria, suboptimal in terms of meeting stability conditions [10, 7]. Time delays caused by D/A conversion, motor torque realization in the inner current (torque) control loop and digital speed measurement can be approximated by first-order lag terms [13, 14]. When stability analysis is performed, it can be seen that it is impossible to avoid limit cycle oscillations and have a stable behavior for such a system structure [9].

This paper is organized as follows: section 2 describes the implementation of a cascade structured position controller. Model based backlash compensation is presented in section 3, followed by the description of variable structure position controller in section 4, and presentation of the experimental research in the 5<sup>th</sup> section. A review of the results is given in the conclusion (section 6).

## 2 Cascade structured position controller

A fast, linear driving torque controller assumes the role of backlash influence compensator, since it encompasses the place where backlash exists in the system. The structure of inner control loop (driving torque control, with polynomial controller) of an elastic servosystem is represented by a block diagram in Figure 3.

Polynomial controller in Laplace ( $z$ ) domain is [13]:

$$m_{1Rd}(z) = \frac{T(z)}{R(z)} m_{1R}(z) - \frac{S(z)}{R(z)} m(z), \quad (1)$$

while the transfer function of a mechanical system model, according to Fig. 1b) (without the influence of backlash) is:

$$G_{mms}(s) = \frac{m(s)}{m_1(s)} = \frac{T_{M2}}{T_{M\Sigma}} \frac{1 + 2\zeta_2\Omega_{02}^{-1}s}{1 + 2\zeta_2\Omega_0^{-1}s + \Omega_0^{-2}s^2}. \quad (2)$$

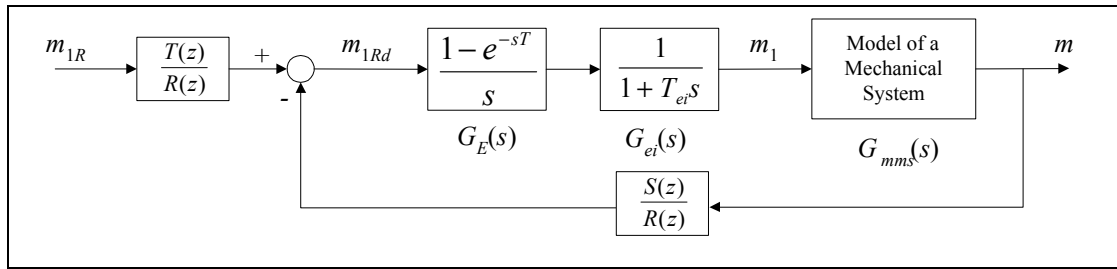


Fig. 3. Structural block diagram of driving torque control loop, with polynomial controller.

## 3 Model based backlash compensation

The grounds for this method of backlash influence compensation are: the mechanical system can be linearized, hence the influence of backlash can be cancelled, by adding a compensation signal to the output of an angular velocity controller [9].

If non-dominant time constants ( $T_{ei}$  – driving torque closed control loop and  $T$  – sample time) are neglected, using algebra of blocks it is possible to obtain an alternative representation of control system. In that case, the mechanical part of the system retains linear structure, whereas the influence of backlash is expressed by an additional component of control signal:

$$m_{bc}(z) = \left[ T_{M1} \left( \frac{z-1}{Tz} \right)^2 + K_{\omega 1} \frac{z-1}{Tz} + K_{\Delta\alpha} \right] \Delta\alpha_B(z), \quad (3)$$

Polynomial coefficients  $R(z)$ ,  $S(z)$  and  $T(z)$  are tuned so, that the driving torque control loop response to torque reference  $m_{1R}$  is fast, accurate and well damped, as well as robust and minimally sensitive to measurement noise [10].

Driving torque closed control loop is further replaced by a continuous PT<sub>1</sub> element, so the system can be modeled as a stiff drive, and the synthesis of outer (speed and position) control loops can be conducted utilizing quasicontinuous approach [10].

During synthesis, the lag due to sampling and extrapolation is modeled by a PT<sub>1</sub> element with time constant of half a sampling interval. Furthermore, the three PT<sub>1</sub> elements with ‘parasitic’ time constants are replaced by a single PT<sub>1</sub> element with an equivalent constant  $T_{\Sigma} = T + T_{em}$ . The damping optimum yields the controller parameters  $T_{e\omega}$ ,  $K_{\omega}$ ,  $K_{\alpha}$ ,  $T_I$ .

Simulations have proved that the inner control loop (driving torque control) has to be designed for aperiodic response, for overall stability of the system.

The influence of altering the equivalent time constant  $T_{em}$  had later been investigated, with the conclusion that it is advisable to design the inner control loop (torque control) as slow as possible, without jeopardizing the overall system stability. That way, the system would last longer, because of lesser control efforts.

which replaces the influence of the physical element *backlash* in the system. Simulated output is the same as for the system with backlash.

The extension to the described approach was an attempt to use that idea for compensation of backlash influence, as shown in Figure 4. A servosystem with linear position controller was expanded with nonlinear backlash compensator (compensation branch). This way, compensation was done in closed loop, and the compensation signal  $m_{bc}$  had been generated from angular difference on the shaft,  $\Delta\alpha$  (s).

Obviously, when measured signal of angular acceleration is used for control, one can expect higher sensitivity to measurement noise, hence an impulse-like behavior of compensation signal. Therefore, compensator output can be filtered to some extent, with the aim of “flattening” the shape of compensation signal, without significant influence on compensation quality. A 3<sup>rd</sup> order discrete-time Butterworth filter was used because of its simplicity, yet satisfying performance.

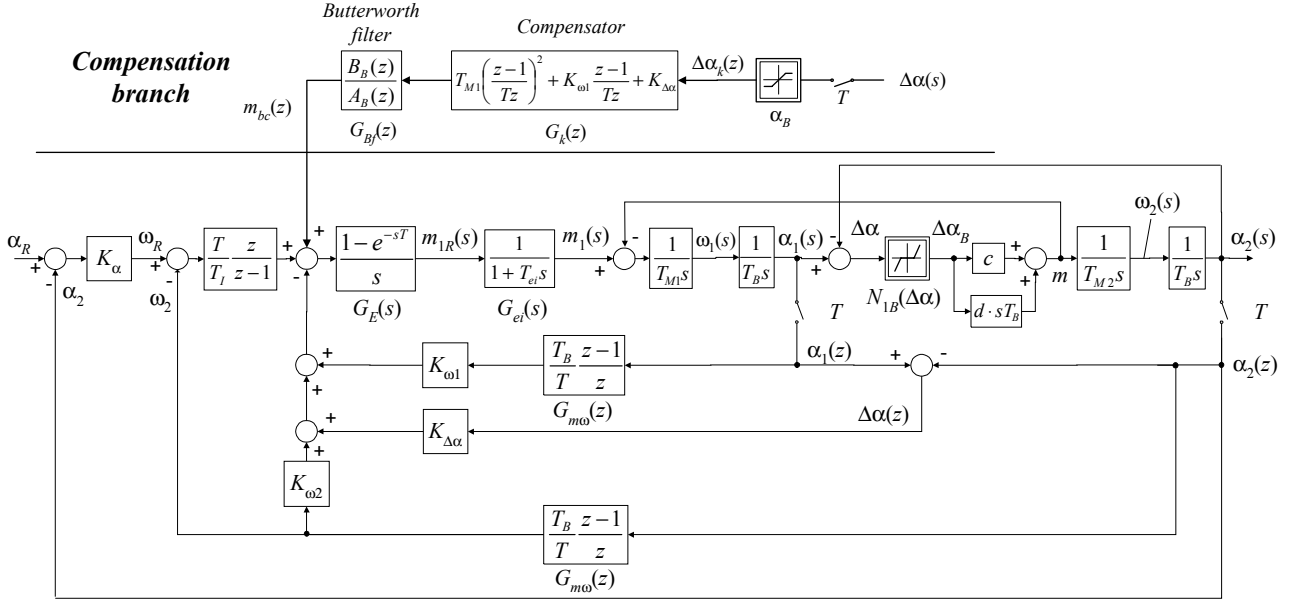


Fig. 4. Structural block diagram of a position controlled servodrive, with model-based backlash compensation algorithm.

#### 4 Variable structure position controller

The basic idea for this backlash compensation method is the fact that the dynamic model of the system, when the shaft is inside the neutral zone, is different than the linear model. That fact leads to the conclusion it would be possible to compensate the backlash influence with a dual-structure controller: one structure for linear, and the other for nonlinear mode of work [15].

As long as the system is in linear mode of work, a classic linear load-position ( $\alpha_2$ ) controller is active. When the shaft enters the neutral zone, the problem of stiff transmission can be considered. Driving motor position ( $\alpha_1$ ) is controlled in that period, with the goal of bringing the shaft as quickly as possible, but *aperiodically* to the boundary of neutral zone.

Controller structure switching algorithm is revealed on a flow chart in Figure 5. The first choice depends on the current position of the shaft. If the shaft is not within the neutral zone, the 'main' controller (load position controller  $\alpha_2$ ) is active. When the shaft enters the neutral zone, which happens during accelerating or braking, drive position controller ( $\alpha_1$ ) is activated. The position of driving motor at that moment  $\alpha_{10}$  is proclaimed initial and recorded, upon which it is used for formation of reference. If the system is accelerating, it is first necessary to aperiodically lead the shaft outside of the neutral zone and then let the load position controller take over. For that purpose, reference is set to  $\alpha_{1R} = \alpha_{10} + \alpha_B$ . Start-up is detected when there is a position difference larger than  $\alpha_B$ , while load mass is still static. If the system is in non-linear regime because it needs to start braking, drive position controller has the task to stop the driving motor in initial position  $\alpha_{10}$ . During that interval, the load mass crosses the backlash and the system is stopped because the 'main' controller (load position controller  $\alpha_2$ ) is active.

Once the shaft enters the neutral zone, a faster driving motor

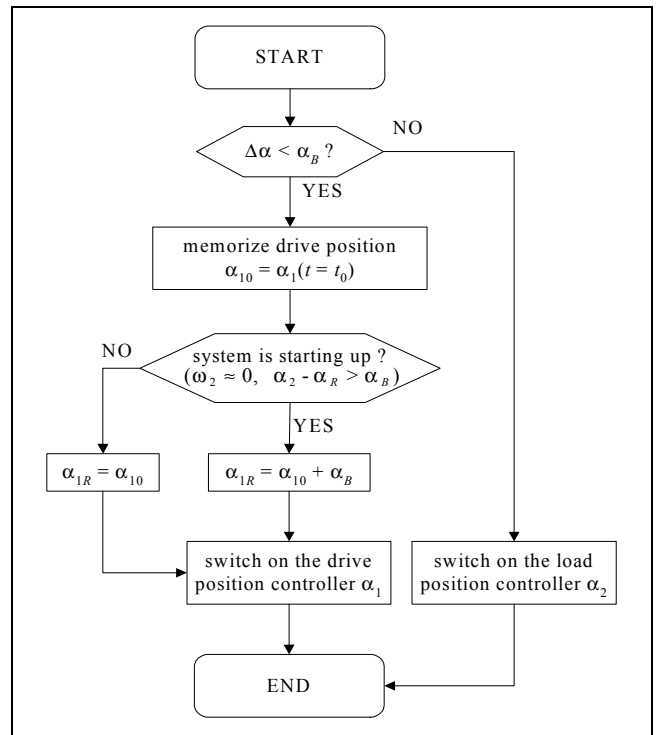


Fig. 5. Controller structure switching algorithm flow chart.

position controller is switched on (its equivalent time constant  $T_{el}$  is four times smaller than that of a closed load-position control loop time constant  $T_{e\alpha}$ ). The driving motor quickly accelerates in the opposite direction, the shaft exits the neutral zone and the main controller is activated again.

Clearly, the two distinct phases are: forced exit from neutral zone and linear mode of work. The method showed good results for different values of backlash and various inertial ratios, with the only limitation on minimal reference being  $\alpha_R \approx 10 \alpha_B$ .

## 5 Experimental results

Experiments described in this paper were conducted on a laboratory model of a servosystem designed for the purpose of analyzing elasticity, backlash and friction effects in various mechanisms and devices [2, 4]. The photograph and the functional scheme are shown in Figures 6 and 7, respectively. The servosystem consists of two brushless servodrives (driving and load motor), adequately designed mechanical components (shafts, devices for generation of friction and backlash), sensors (angular displacement, speed and coupling torque) and a digital control system. The driving torque is generated by brushless servodrives with rated motor torque  $M_n = 14.8$  (Nm) and rated speed  $n_n = 2000$  (rpm), while the angular displacement is measured by a precise incremental encoder with a resolution of 120000 pulses per revolution.

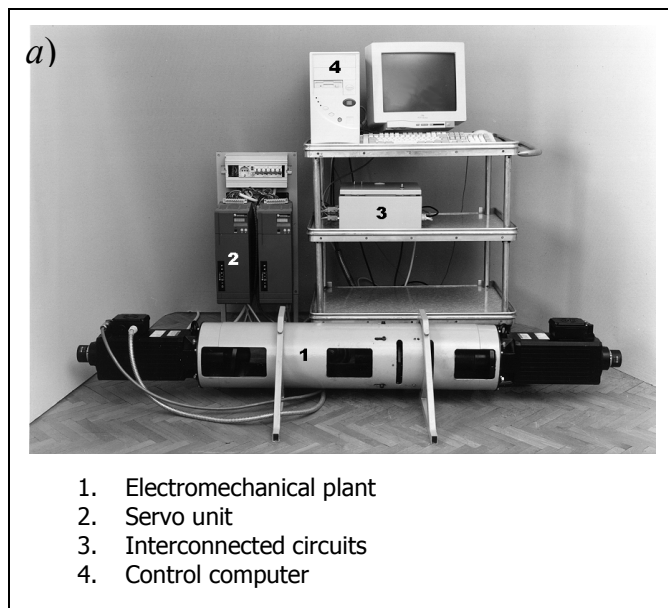


Fig. 6. Laboratory model photo.

The existence of friction in laboratory model complicates the analysis, in the sense it causes the driving motor to be “ahead of” the load in static mode, and keeps the shaft twisted. The difference between stationary values of controller output ( $m_{1R}$ ) and driving torque ( $m$ ) equals the friction torque. Backlash is influential (and compensated in manners described in sections 2, 3 and 4) in transient state. On the other hand, due to stabilizing effects of friction in a combination with backlash, the limitation on minimal reference value ( $\alpha_R > 10\alpha_B$ ) discovered in simulations, was eliminated. Friction compensation was not included because friction ultimately did not dominantly shape the behavior of the system, furthermore it affected all backlash compensation methods the same way.

Regarding experimental results for the first analyzed method of backlash compensation, the system with a cascade structured position controller was in limit cycle oscillatory mode. The reason for such behavior is the fact that introducing a torque sensor adds some elasticity to the

system, thus causing the inner (torque) control loop to be unable to respond quickly enough and compensate backlash influence. One can conclude that the application of torque controller is only sensible when neither the transmission mechanism, nor the torque sensor are very elastic. A significant improvement of the implementation is possible if semiconductor torque-measuring tapes are used (with up to 50 times larger sensitivity than the standard tapes used), or incorporating the torque-measuring element in the base of mechanical system [4].

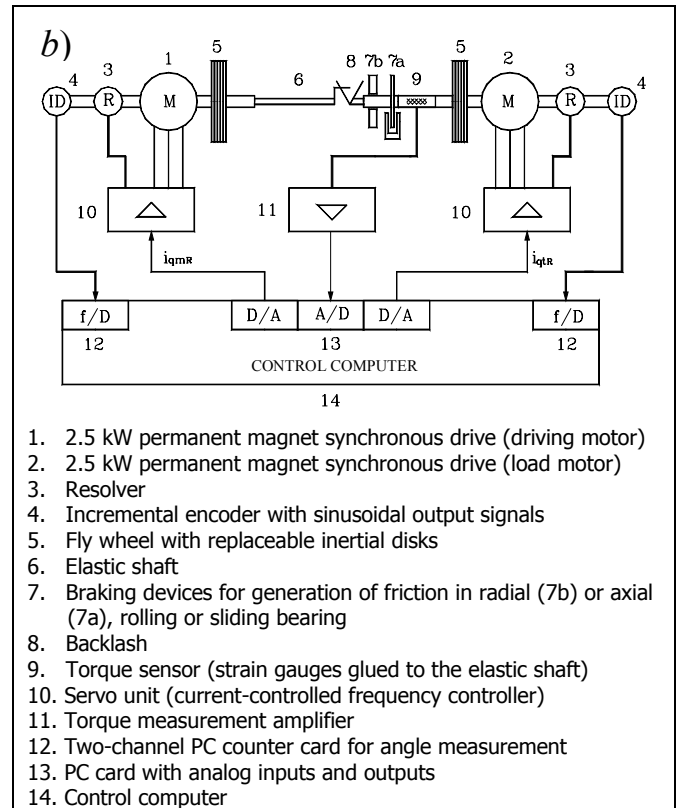


Fig. 7. Laboratory model functional scheme.

The effects of the remaining two proposed backlash compensation methods on a servodrive can be observed on Figure 8. Experimental results are shown for model based backlash compensation (1), and a variable structure position controller (2).

The shaft had first been positioned to the boundary of the neutral zone, after which reference was imposed. The system was monitored from that moment. The first phase looks the same as for a non-compensated system. For model based backlash compensation, when the shaft enters the neutral zone (braking commences), the signal  $\Delta\alpha_k$  on compensator input is changed (until that moment it was constant,  $\Delta\alpha_k = \alpha_B$ ) and due to double differentiation, a pulse appears on its output. As a result, driving motor rapidly accelerates in the opposite direction, the shaft exits the neutral zone and, because of its elasticity small oscillations appear on the opposite side of backlash. When angular difference gets stabilized on a value  $\Delta\alpha = -\alpha_B$ , the system is again in linear mode and the compensator has no influence on its behavior.

Two major phases can be distinguished:

- forced exit from neutral zone, and
- linear mode of work.

For the reason of lowering the control signal peaks, a simple third-order filter is introduced. Considering the filter cut-off frequency  $\omega_b$ , the pressure on the system decreases when it is lower, but there is a limit as to what extent  $\omega_b$  can be pushed. Its further lowering deteriorates system behavior and leads to instability. Increasing  $\omega_b$  allows larger peaks of controller output, without impact on the overall system performance.

Behavior of a system with variable structure position controller is described in chapter 4.

It can be concluded that both methods are successful at a price of forcing the controller output  $m_{1R}$ .

Experiments were also conducted for different inertial ratios (when drive inertia was larger than load inertia,  $r = 0.2$ ; and a 'better' case when load was heavier than the drive,  $r = 5$ ). Backlash influence was very well compensated in all cases.

## 6 Conclusion

This paper deals with three approaches to backlash effect compensation in servodrives. Considering the fact that the algorithms tend to "linearize" the system, two major phases of work are distinguishable: forced exit from backlash zone, followed by work in linear mode. All methods proved robust with respect to different inertial ratios, and using simple engineering practices helped significantly lower the peaks of control signal. This was important taking into consideration the durability of transmission elements in controlled servodrives.

Experimental results did not exactly match simulated ones, because of friction, but it did not have a major impact on system behavior. It can be concluded that the investigated methods proved successful in compensating the influence of backlash in servosystem transmission elements. The main advantage of the method with cascaded controller structure was the fact it is essentially a 'hardware' solution, while the controller algorithm is basically quite simple. Major disadvantage being the need for fast and accurate measurement of torque on the shaft, which is not often feasible. Model-based compensation method showed the largest flexibility with respect to different inertial ratios and values of backlash, at the same time being more sensitive to measurement noise (because of measured position signal differentiation) and having a slightly more complex control algorithm. Variable structure controller was least demanding from measurement standpoint, but also least applicable for larger values of backlash or unfavorable ratios of inertia.

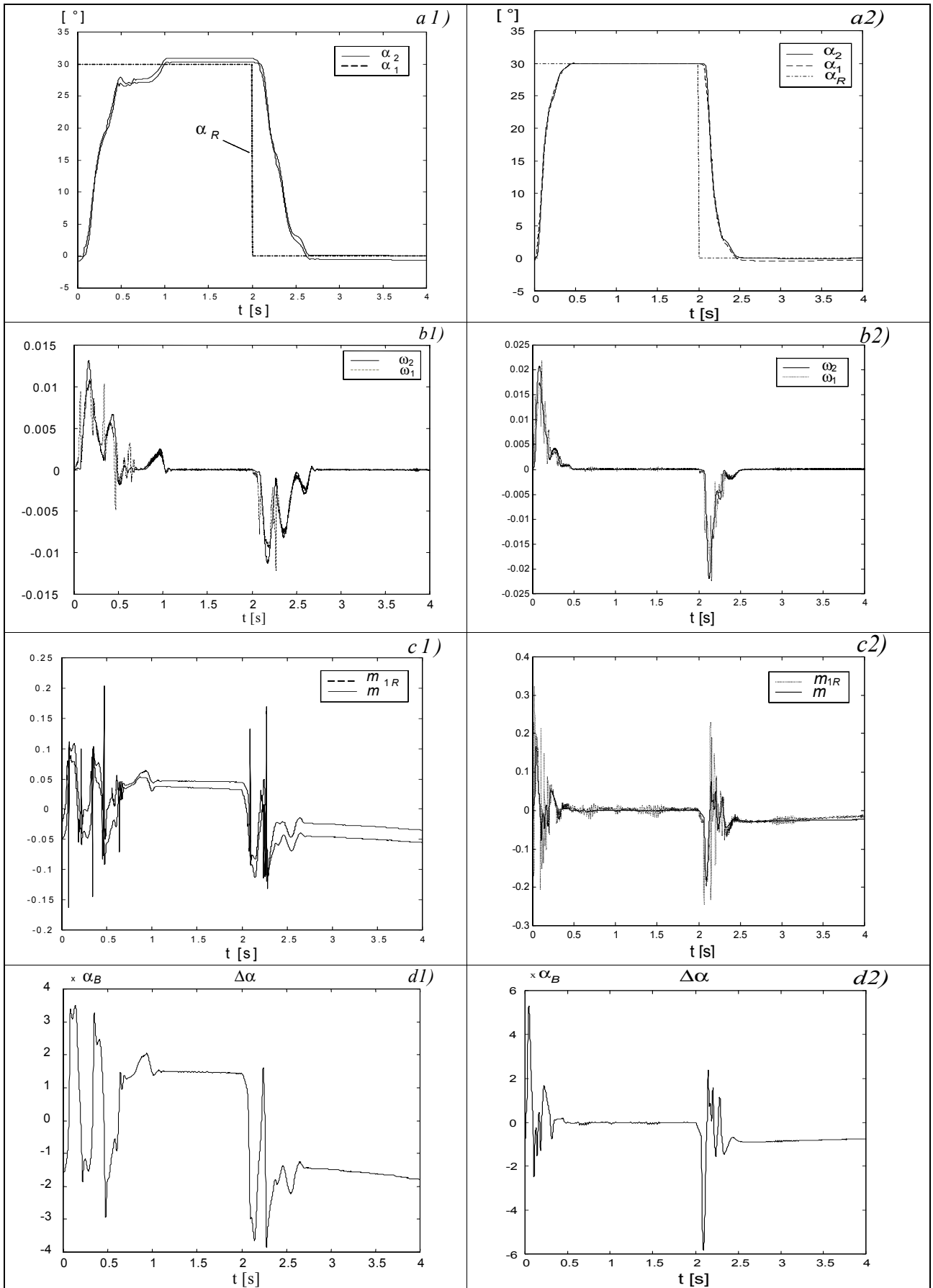
Successfulness of model-based compensation method was later used as a basis for developing an autotuned position controller with backlash compensation [8].

The scope of this paper is restricted to the methods of backlash compensation that were analyzed in deep and later implemented in a real-life industrial and military system.

Consequently, it does not attempt to be a comprehensive survey of all existing methods, rather an engineering road sign to several solutions applicable in various environments. Therefore, the reader is encouraged to further explore some other methods, described in [12].

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**Fig. 8.** Experimental responses of a position controlled system with: model-based backlash compensation (1); a variable structure position controller (2). 1)  $r = 1,63$ ;  $\alpha_B = 0,43^\circ$ ;  $\omega_b = 500$  Hz 2)  $r = 1,63$ ;  $\alpha_B = 0,43^\circ$ .