Position Control for a Belt Positioning System Including Backlash

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Abstract—Gear backlash is a major challenge for dynamic and accurate positioning of drive systems. It can be overcome to some extent by increasing the accuracy of the mechanical parts, associated with major additional costs, or by means of control. In this contribution the latter approach was chosen using feedforward control based on an idealized single mass model and a nonlinear controller, accounting for elasticity and backlash. Depending on the actual contact state different controllers are developed, optimized and applied. At the example of numerical model of a belt positioning system the superiority with respect to simple PID-control is shown both in terms of positioning time as well as stability.

Keywords: position control, drive train, backlash, switching model, switching controller

I. Introduction

Positioning is one of the most frequent tasks in control engineering. Especially for the sake of efficiency and process quality high precision and dynamics are desirable. One particular problem of all drive systems including gear transmissions is gear backlash. The latter is caused by the hysteresis effect due to clearance between tooth flanks and cause position discrepancies between drive and load. In particular movements from rest suffer from contactless states without the feedback of the load.

The challenge in position control arises when the load overshoots. Then the drive has to change direction and move from one to the other contact side without significant impact of the teeth which could cause controller banging. Hence teeth clearance is a major factor for the precision and stability of positioning.

Control of drive trains has a long history in engineering. Lagerberg in [1] as well as Nordin and Gutman in [2] give a good review of the publications over the last decades highlighting different aspects such as modeling, identification and control.

Concerning the system so called physical modeling is most widely used. The three components gear ratio, elasticity and clearance are separated in idealized parts. It can be either in contact or backlash mode switching between them depending on the state of the gear. It is therefore usually called switching model.

From the controllers point of view categorization is possible in linear, nonlinear passive and nonlinear active. From the last group switching controller are particularly promising where different control laws are used for contact and backlash mode. Switching between the modes is done according to state measurement or state observation [1].

Lagerberg considered switching controllers for the application on accelerated automotive drive trains [3] and Ros- talski et al. for velocity control [4]. In both cases an additional observer was used to generate the unknown states such as the contact of the flanks.

In [5] Khan et al. proposed a multiple model adaptive strategy for velocity control of drive trains with clearance which made the state observer obsolete. In a subsequent contribution they investigated the position control of the same system [6] but suffered from excessive positioning times.

Since the approaches are only conditionally useful for position control the focus of this contribution is on the design of a dynamic and precise positioning system. After the system is presented in section II and modeled in section III it is analyzed (section IV) and controlled (section V). After the results are evaluated in section VI finally a summary as well as an outlook is given in section VII.

II. System Description

The belt drive system under consideration might usually be reduced to a small number of relevant components, see Fig. 1.

![Fig. 1. Elements of a belt drive.](image)

The central element is the belt itself which is an endless band. Considerable pretension in conjunction with the high friction coefficient between pulley and belt prohibit slip. In order to prevent vibrations of the belt it moves on a sliding surface. The motor (connected to the driving pulley by a gearbox) as well as the load (pulleys and belt) are equipped
with position measurement systems which measure the angle of the first and the linear position of the latter respectively. Hence imperfections of the pulleys and the belt will not have any influence on the precision of the positioning.

The performance requirement for the belt drive system is to reach a position tolerance of ±0.01 mm within 0.6 s on a length of 200 mm and 0.6 s. For a pulley with \( R = 125 \text{ mm} \) this is equivalent to an angular movement of 1.6 rad and a tolerance of ± 8·10⁻⁵ rad.

### III. Modeling

The mathematical modeling of the belt drive follows the simplified two mass approach proposed in [3], c.f. Fig. 2. Since the stiffness of the belt is several orders higher than the one of the gear drive this assumptions seems to be justifiable.

![Fig. 2. Two-mass gearing with backlash.](image)

\( J_m \) denotes the moment of inertia of the engine and transmission (reduced to the drive side) and \( J_l \) the moment of inertia of the belt and pulleys. The bodies are linked via an (ideal) transmission with transmission ratio \( i \) whose elastic properties are reflected in a shaft of stiffness \( k \) and damping constant \( c \). Gear clearance \( 2\alpha \) is modeled as separate element and includes the gear ratio. Additionally viscous friction on both drive and load side are included via the respective coefficients \( c_{mi} \) and \( c_l \).

The resulting differential equation system of the model is

\[
\begin{align*}
\frac{M_s}{I} &= M_m - J_m \dot{\omega}_m - c_m \omega_m, \quad (1) \\
M_s &= J_l \dot{\omega}_l + c_l \omega_l \\
\omega_m &= \dot{\varphi}_m, \quad \omega_l = \dot{\varphi}_l \\
\end{align*}
\]

with \( \omega_m = \dot{\varphi}_m, \omega_l = \dot{\varphi}_l \) cmp. [3].

\( M_m \) is the drive torque and \( M_s \) the torque acting on the shaft, which, according to Nordin et al. [7], is calculated differently depending on the gear state

\[
M_s = \begin{cases} 
  k \varphi_s + c \omega_s & \text{contact} \\
  0 & \text{backlash} 
\end{cases} 
\]

with the shaft torsion

\[
\varphi_s = \frac{\varphi_m}{i} - \varphi_e. 
\]

Note that in backlash mode no torque is transferred by the shaft. Also the backlash angle

\[
\varphi_b = \varphi_e - \varphi_l 
\]

with

\[
-\alpha \leq \varphi_b \leq \alpha
\]

and the total difference between drive (after the gearbox) and load angle are defined

\[
\varphi_d = \frac{\varphi_m}{i} - \varphi_l = \varphi_s + \varphi_b.
\]

Hence the nonlinear equations of motion (1) and (2) which can be described in contact mode by

\[
\begin{bmatrix}
\omega_m \\
\omega_l \\
\varphi_m \\
\varphi_l \\
\varphi_b \\
\end{bmatrix} =
\begin{bmatrix}
\omega_m \\
\omega_l \\
\varphi_m \\
\varphi_l \\
\varphi_b \\
\end{bmatrix} +
\begin{bmatrix}
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
\end{bmatrix}
M_m
\]

and in backlash mode by

\[
\begin{bmatrix}
\omega_m \\
\omega_l \\
\varphi_m \\
\varphi_l \\
\varphi_b \\
\end{bmatrix} =
\begin{bmatrix}
\omega_m \\
\omega_l \\
\varphi_m \\
\varphi_l \\
\varphi_b \\
\end{bmatrix} +
\begin{bmatrix}
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
\end{bmatrix}
M_m
\]

where all states except for \( \varphi_b \) are assumed to be measurable. The output equation is identical in both modes

\[
y = \begin{bmatrix}
1 & 0 & 0 & 0 & 0 \\
0 & 1 & 0 & 0 & 0 \\
0 & 0 & 1 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 \\
0 & 0 & 0 & 0 & 1 \\
\end{bmatrix}
\begin{bmatrix}
\omega_m \\
\omega_l \\
\varphi_m \\
\varphi_l \\
\varphi_b \\
\end{bmatrix}
\]

For a unique solution of the differential equation system the initial states need to be known which are assumed to be trivial (drive and load at rest as well as maximal gear clearance \( \alpha \))

\[
x = \begin{bmatrix}
\omega_m \\
\omega_l \\
\varphi_m \\
\varphi_l \\
\varphi_b \\
\end{bmatrix} = 0. 
\]

The parameters for subsequent system and control analysis are taken from positioning system for textile printing. The respective values are:

\[
\begin{align*}
J_m &= 5.4 \times 10^{-4} \text{ kg m}^2 \\
J_l &= 4.8 \text{ kg m}^2 \\
k &= 182 200 \text{ N m/rad} \\
c &= 100 \text{ N m/s/rad} \\
c_m &= 0.002 \text{ N m/s/rad} \\
c_l &= 0.1 \text{ N m/s/rad} \\
\alpha &= 4.36 \times 10^{-4} \text{ rad} \\
i &= 50
\end{align*}
\]

<table>
<thead>
<tr>
<th>TABLE I. Parameter for the two mass model.</th>
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<tbody>
<tr>
<td>( J_m )</td>
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IV. System Analysis

In the framework of an analysis in time domain the dynamic properties of the two-mass system is investigated. To this end different signals for the input $u(t)$ and the transient output $y(t)$ are chosen.

According to the state equation (9) and (10) respectively, $M_m$ is chosen as the input. At the same time not all states are visible at the output, cf. Eqn. (11). In the simulation however all variables are accessible.

Let us assume a sinoid development of the input. Frequency and amplitude are chosen in a way that the load displacement is $\varphi_l = 1.6$ rad within $T = 0.4$ s.

In a first step a frictionless system ($c_m = c_l = 0$) is considered. Fig. 3 shows the development of the drive torque $M_m$ and the shaft torque $M_s$ for $0 \leq t \leq 2T$. Due to the initial conditions at $t = 0$ no torque is transmitted to the shaft. But already after a short moment contact mode is reached and the $M_s$ increases rapidly.

The progress of angle and angular velocity of drive and load are depicted in Fig. 4. The small remaining velocities provoke continuing impacts.

In Fig. 5 the development of $\varphi_s$ and $\varphi_l$ as well as their time derivations are shown. The continuing movement of the load indicates the need for active control. As soon as the drive torque becomes negative the drive decelerates and loses contact to the load. Only when contact is established on the negative contact side also the load starts to slow down.

Alternatively the system might be considered in the frequency domain neglecting transient effects. Choosing harmonic input $u$ the system response $y(\omega)$ in terms of amplification and phase shift can be found.

![Fig. 3. Temporal development of drive and load torque.](image1)

![Fig. 4. Temporal development of engine and load angle.](image2)

![Fig. 5. Temporal development of backlash, difference and shaft angle.](image3)

Although the complete system is nonlinear the dynamics of contact and backlash mode alone are linear. A harmonic analysis is therefore meaningful as long as the system does not change mode. This is e.g. the case for the entire acceleration phase, cf. Fig. 4.

The starting point for the harmonic analysis is the equation of motion in matrix form

$$M \cdot \ddot{x} + D \cdot \dot{x} + K \cdot x = f. \quad (13)$$

If the generalized force $f$ is harmonic the steady-state solution of above equation is

$$x = x_{\text{max}} \cdot e^{j\mathbf{\Theta}t} \quad (14)$$

with

$$x_{\text{max}} = \begin{bmatrix} x_{\text{max,1}} \\ \vdots \\ x_{\text{max,n}} \end{bmatrix}, \quad \mathbf{\Theta} = \begin{bmatrix} \theta_1 \\ \vdots \\ \theta_n \end{bmatrix}. \quad (15)$$
Here \( n \) is the number of degrees of freedom, \( x_{\text{max}} \) the corresponding amplitude and \( \Theta \) the phase. Eqn. (14) may be also written in the form of

\[
x = x_{\text{max}} \cdot (\cos \Theta + j \sin \Theta)e^{j\Omega t}
\]

or

\[
x = (x_1 + jx_2)e^{j\Omega t}
\]

with

\[
x_1 = x_{\text{max}} \cdot \cos \Theta, \quad x_2 = x_{\text{max}} \cdot \sin \Theta.
\]

There \( x_1 \) corresponds to the real and \( x_2 \) to the imaginary part of the displacement vector \( x \). Simple algebra yields absolut value and phase of the displacement

\[
x_{\text{max},n} = \sqrt{x_{1,n}^2 + x_{2,n}^2},
\]

\[
\Theta_k = \arctan \frac{x_{2,n}}{x_{1,n}}.
\]

In analogy also the load is assumed to be a complex vector

\[
f = f_{\text{max}}e^{j\Omega t}
\]

or

\[
f = (f_1 + jf_2)e^{j\Omega t}.
\]

Substitution of Eqn. (17) and (21) into (13) yields

\[
(-\Omega^2M + j\Omega D + K) \cdot x e^{j\Omega t} = f_{\text{max}} e^{j\Omega t}
\]

and after division by the harmonic function

\[
(K - \Omega^2M + j\Omega D) \cdot x = f_{\text{max}}.
\]

Knowing the load vector \( f \) above equation can be directly solved yielding

\[
x = (K - \Omega^2M + j\Omega D)^{-1} \cdot f_{\text{max}}
\]

Since mode change is not taken into account the a two-mass system without gear clearance is used to find the matrices for Eqn. (13)

\[
\begin{bmatrix}
    x_{1,n} & x_{2,n}
\end{bmatrix}
\begin{bmatrix}
    x_{1,n} & x_{2,n}
\end{bmatrix}^T + \begin{bmatrix}
    f_{\text{max}} & f_{\text{max}}
\end{bmatrix}^T + \begin{bmatrix}
    \frac{\Omega m}{k} & \frac{\Omega m}{k}
\end{bmatrix}^T + \begin{bmatrix}
    \frac{\Omega m}{k} & \frac{\Omega m}{k}
\end{bmatrix}^T = \begin{bmatrix}
    M_{\text{m}} & 0
\end{bmatrix}
\]

Fig. 6 represents the bode plot of drive and load for the linear two-mass system.

31 Hz and 66 Hz, the frequencies of the first (in-phase) and second (out-of-phase) vibration mode, seem to be characteristic frequencies, resulting in pronounced changes of amplitude and phase. At the lower characteristic frequency the engine movement is damped by the load (vibration elimination) whereas at the higher one it is amplified (resonance).

At low exciting frequencies the displacements are large (integrating behavior of the system) but in phase. The constant difference of the amplitude characteristics stems from the gear ratio \( i = 50 \) since \( x_{\text{m}} = x_{l,i} \). Engine and load move synchronous like a rigid body. Consequently the excitation should not contain frequency components near 66 Hz. Ideally the highest frequency should be below \( \leq 10 \) Hz which is still compatible with target settling times of 0.6 s.

V. Control

The aim of control is to provide the suitable actuating variable (drive torque) which brings the controlled variable as closely as possible to the reference input.

The concept used for the belt drive is based on a combination of feedforward and feedback control. The feedforward part here is responsible for the belt’s feed according to some desired trajectory whereas the feedback part should eliminate any deviation. Fig. 7 depicts the scheme of the control loop.
For the design of the feedforward control it is assumed that an actuating function \( M_m \) can be found which theoretically produces the exact trajectory of \( \hat{\phi}_l \). The error \( e_{\phi_l} \), which is due to disturbances and modeling errors is controlled by an additional part \( \Delta M_m \). Hence the complete actuating variable is

\[
M_m = \hat{M}_m + \Delta M_m.
\]

(27)

In the sense of a flatness based control design the feedback part of the actuating variable has to be expressed as function of the reference input and its derivations [8]. The necessary system inversion is for sake of simplicity not done for the two-mass system but an equivalent one-mass model where the engine is reduced on the load side.

\[
(J_m i^2 + J_l) \ddot{\omega}_l + (c_m i^2 + c_l) \dot{\omega}_l = M_m i.
\]

(28)

To this end the flat output \( y_f \) is introduced, corresponding to the system state for single degree of freedom systems

\[
\begin{align*}
y_f &= \varphi_l, \\
\dot{y}_f &= \dot{\varphi}_l = \omega_l, \\
\ddot{y}_f &= \ddot{\omega}_l = -\frac{c_m i + c_l}{J_m i + J_l i} \dot{\omega}_l + \frac{1}{J_m i + J_l i} M_m (31)
\end{align*}
\]

From Eqn. (31) the necessary drive torque for the feedforward control can be calculated from the flat output

\[
\hat{M}_m = (J_m i + J_l i) \hat{\dot{y}}_f + (c_m i + c_l i) \hat{\ddot{y}}_f.
\]

(32)

In the present case when the flat output \( y_f \) and the controlled variable \( \varphi_l \) coincide, the actuating variable can be calculated directly from the desired trajectory. For the latter especially polynomials are suitable due to the smoothness of their derivations. Following boundary conditions have to be respected

\[
\begin{align*}
\hat{y}_f(0) &= 0, & \hat{\dot{y}}_f(0) &= 0, & \hat{\ddot{y}}_f(0) &= 0, \\
\dot{y}_f(T) &= 0, & \ddot{y}_f(T) &= 0, & \hat{\ddot{y}}_f(T) &= 0,
\end{align*}
\]

(33)

with

\[
y_0 = 0 \text{ rad}, \ y_c = 1.6 \text{ rad}, \ T = 0.4 \text{ s}.
\]

(34)

Hence a polynomial of order 7 has to be used

\[
\hat{y}_f(t) = a_0 + a_1 t + a_2 t^2 + a_3 t^3 + a_4 t^4 + a_5 t^5 + a_6 t^6 + a_7 t^7
\]

(35)

whose coefficients are derived from the boundary conditions (33). Fig. 8 depicts the actuating variable in the interval \( 0 \leq t \leq 2T \) as well as its spectral decomposition. Evidently the most prominent components are well below 10 Hz.

Fig. 7. Block scheme of the control system.

Fig. 8. Temporal development of the actuating variable and amplitude spectrum.

The controller design is based on the principle of the switching control. Hence different controllers are used for contact and backlash mode. An additional logic is switching between them.

The contact controller is actually responsible for eliminating the control error whereas the backlash controller should establish contact as soon as possible, preferably without causing a jerk. The inputs are denoted depending on the state

\[
\Delta M_m = u = \begin{cases} 
 u_{co} & \text{if } |\varphi_d| \geq \alpha \text{ contact} \\
 u_{bd} & \text{if } |\varphi_d| < \alpha \text{ backlash} 
\end{cases}
\]

(36)

The switching condition of the controller in this case is identical to the one of the model. There are actually many possibilities to define the switching condition but \( \varphi_d \) is especially practical.

In order to find an appropriate control law the relevant quantities are considered in detail. The contact controller eliminates the difference between load position and reference position. The error is defined as

\[
e_{\varphi_l} = \varphi_l - \hat{\varphi}_l
\]

(37)

with

\[
e_{\varphi_l} = \begin{cases} 
 > 0 & \text{load ahead of reference position}, \\
 = 0 & \text{load at reference position}, \\
 < 0 & \text{load behind reference position}.
\end{cases}
\]

(38)
The backlash controller however eliminates the difference between shaft and contact side of the load. To this end the difference angle $\Delta \varphi$ is introduced, differentiating between positive $(\varphi_l + \alpha)$ and negative $(\varphi_l - \alpha)$ contact side

$$\Delta \varphi = \frac{\varphi_m}{i} - (\varphi_l \pm \alpha) = \varphi_d \mp \alpha \quad (39)$$

with

$$\Delta \varphi = \begin{cases} \varphi_d - \alpha & \text{for positive contact side} \\ \varphi_d + \alpha & \text{for negative contact side} \end{cases} \quad (40)$$

Depending on the sign of the error the engine has to be brought into contact with the respective contact side of the load. The resulting conditions for the difference angle are

- if $e_{\varphi_1} > 0$ then $\Delta \varphi = \varphi_d + \alpha > 0$
- if $e_{\varphi_1} = 0$ then $\Delta \varphi = 0$
- if $e_{\varphi_1} < 0$ then $\Delta \varphi = \varphi_d - \alpha < 0$

The actual control laws chosen are PD-controllers for both the contact mode and the backlash mode

$$u_{co} = - [K_{c_{o1}} \quad K_{c_{o2}}] \cdot \begin{bmatrix} e_{\varphi_1} \\ e_{\omega_1} \end{bmatrix}$$
$$u_{bl} = - [K_{b_{l1}} \quad K_{b_{l2}}] \cdot \begin{bmatrix} \Delta \varphi \\ \Delta \omega \end{bmatrix} \quad (42)$$

with

$$e_{\omega_1} = \omega_l - \hat{\omega}_l, \quad (43)$$
$$\Delta \omega = \frac{\omega_m}{i} - \omega_l. \quad (44)$$

The control parameters $K_{c_{o1}}$ and $K_{b_{l1}}$ respectively concern the positioning error and $K_{c_{o2}}$ and $K_{b_{l2}}$ respectively the velocity error. Considering the velocity error prevents too fast positioning by the contact controller and jerks by the backlash controller.

Since jerks are unavoidable during the positioning phase, the feedback controller is activated only after the end of the feedforward phase at $T = 0.4$ s. The input variables ($e_{\varphi_1}, e_{\omega_1}, \Delta \varphi, \Delta \omega$) can all be derived from the measured quantities ($\varphi_m, \varphi_l, \omega_m, \omega_l$) and the reference quantities ($\hat{\varphi}_l, \hat{\omega}_l$). The backlash controller additionally calculates the sign of $e_{\varphi_1}$ according to (38).

VI. Results

The control parameters are determined by recursive simulation of the positioning process and optimization. An example for a set fulfilling the design requirements is

$$K_{c_{o1}} = 280, K_{c_{o2}} = 0.1, K_{b_{l1}} = 180, K_{b_{l2}} = 4 \quad (45)$$

Fig. 9 shows the development of the location and velocity error during the positioning process in the interval $0 \leq t \leq 2T$.

Due to the missing feedback control until the end of the gross positioning process at $t = T$, during this phase non-negligible deviations from the planned trajectory occur. Starting from $t = T$ the positioning error is compensated by feedback control. In the case under consideration the load eventually overshoots ($e_{\varphi_1} > 0$). The engine has therefore to be positioned at the negative contact side by the backlash controller. At $t = 0.47$ s contact is established and the load is backing off.

Fig. 9 shows that the error is excellently stabilized. At $t = 0.6$ s the load is clearly within the perimitable tolerance of $\pm 8 \cdot 10^{-5}$ rad. Since the controller tries to compensate even the smallest deviation from the target position the drive never comes to rest. The load alternatingly overshoots whereas the deviation becomes continuously smaller. At $t = 2T$ they are in the order of magnitude of $10^{-6}$ rad.

Alternatively the controller could be deactivated after the load is within certain thresholds of the desired position. This analysis however was not done in the framework of the current contribution.

In comparison PID-control was not able to stabilize the load. Even infinitively small positioning errors triggered controller action resulting in gear banging. Also removing the I-part (PD-control) did not bring remedy. The best results using PID-control are shown in Fig. 10. Note the unstable system behavior after the end of feedforward action at $t = T$ s.
Finally the focus is given to the switching between the two controllers. To this end controller input $e_{\phi}$, switching criteria $\varphi_d$ and the actuating variable $M_m$ are considered in the interval $T < t \leq 3T$, cf. Fig. 11. Contact state is depicted with gray background.

At $t = 0.5$ s e.g. the positioning error is positive. The engine is already at the negative side of the load ($\varphi_d = -\alpha = -4.36 \cdot 10^{-4}$ rad) and the contact controller is already backing off the load ($M_m < 0$). Although the torque is continuously reduced, the load overshoots. Starting from $t = 0.53$ s the sign of the positioning error changes and $e_{\phi} < 0$ which triggers a positive torque ($M_m > 0$) and hence backlash.

Now the backlash controller is activated which tries to bring the engine back into contact as fast as possible. At the beginning $M_m$ is very high but soon to be reduced (finally even changing sign) to prevent a jerk. At $t = 0.62$ s contact is established and $\varphi_d = \alpha = 4.36 \cdot 10^{-4}$ rad. Control is given back to the contact controller. This process is repeated until $e_{\phi} = 0$ or the controller is deactivated.

VII. Conclusions

Using a combination of feedforward and feedback control the positioning requirements could be satisfactorily fulfilled. The nonlinear switching controller is an excellent candidate for this task ensuring fast, precise and stable positioning. The controller possesses, in contrast to PID control, the necessary intelligence to handle the strong nonlinearity in the system.

The drawback of the approach lies in the need for an accurate system modeling. Especially the clearance needs to be known exactly. Additionally minute measurement of drive and load position is necessary. Further controller parameters cannot be derived from simple guidelines but need to be adjusted with the help of simulations. All in all the simulations underline the potential of the approach for practical use.

For further investigations modeling and control of the system should be done in more detail. Next to clearance friction plays a crucial role and should be investigated and considered in detail. Especially dry friction is changing the system behavior significantly.

Further it would be interesting to discretize the controller design. Sensor resolution and sampling frequency might influence the quality of positioning. Not taken into account, even stability of the control system might suffer.

References