Active vibration control of slewing drives with gear backlash

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Abstract Geared transmissions with high reduction ratios are widely utilized in drive technology. Common to these drives is the occurrence of considerable gear backlash. If operating scenarios with repeated reversals of the rotational direction or changing load directions occur, the backlash will be traversed. To overcome the gear play, rotation angles of several hundred degrees can be required at the motor. As a result, unwanted vibrations can occur and high dynamic loads may arise. It is shown that an active vibration control strategy based on full state feedback (LQG) allows to reduce both the dynamic overload and the time for reversing the drive. The model-based design approach relies on two angular measurements only. Multibody system simulations of an exemplary drivetrain are used to verify the possible improvements.

1. Introduction
A multitude of processes relies on the precise rotational motion of a machine around its vertical axis. Typical examples are wind turbines, radio telescopes as well as a variety of machines in material handling technology, e.g. tower cranes. Their slewing motion or yaw motion is usually carried out by means of electrical drives and multi-stage gearboxes with high transmission ratios.
Due to the inevitable elasticity of shafts, couplings and gears, these drive systems are prone to low-frequency torsional vibrations.
One source of excitation is associated with the special operating conditions of slewing drives: As a result of the operating scenario with repeated reversals of the rotational direction and changing load directions at the gearbox output, the backlash of all gear stages is compulsorily traversed. After each backlash traverse, the tooth flanks collide with a certain speed difference. This excites the system’s natural frequencies and causes high dynamic loads in the gearbox.
Large backlash and high acceleration capabilities of electric machines promote the occurrence of substantial speed differences during collision of the tooth flanks.

The angular backlash of a single gear stage rarely exceeds a few degrees. However, in typical drive configurations with multi-stage gearboxes, cumulated backlash angles of significantly larger dimensions frequently occur. In particular, the angular backlash of the low speed gear stages – in conjunction with the high gear ratio in between – can lead to substantial rotation angles of several hundred degrees at the motor shaft, that are required to overcome the gear play.
Backlash itself degrades positioning accuracy and the associated vibration-induced overload contributes significantly to fatigue and wear of drivetrain components, which should be minimized. [1,2]
For positioning drives with explicit requirements for highest precision – used e.g. in machine tools, industrial robots, radar equipment – zero or low backlash can be guaranteed by taking constructive measures. Possible approaches are the use of special gearbox concepts (strain wave gearing, cycloidal drives), or the optimization of conventional gear pairs. Among these are the use of spring-loaded gears, the precise manufacturing of the centre distance combined with tight gearing tolerances (manufacturing effort) and the use of two braced drives. [2,3]
For the majority of slewing drives in heavy industry and materials handling technology, the focus is on providing very high torques with the most robust and simple drive technology possible, so that the occurrence of greater gear backlash is accepted.

In order to achieve a precise tracking of rotational speed or angular position, feedback control systems are utilized. Linear proportional-integral (PI) control based on motor speed feedback forms the de-facto industry standard for slewing drives [1]. The simple (on-site) parameterization and the acceptable robustness justify its widespread use. However, the capability to actively dampen torsional vibrations is limited, even if no backlash is present [4]. Different authors propose the additional feedback of the shaft torque, which is referred to as “disturbance observer” [5], “Gear Torque Observer” [6] or “Brandenburg Observer” [1]. This measure should prevent limit cycles, but since it actually decouples motor and load, improvements are limited to the motor side. The control no longer affects the load side, where only friction remains to counteract vibrations [7].

Since backlash introduces two distinct modes in a drive (contact mode and backlash mode), control by means of two different controllers, each active in either contact or backlash mode would suggest itself. The switching is realized by measuring the motor and load positions and comparing them to the backlash size [8]. In [7] a switching controller is presented, where strong control action enforces a fast reestablishment of flank contact. Another concept [9,10] relies on the computation of an optimal trajectory, allowing to smoothly reach the former non-operating flank.

In [11] the switching between two different control concepts is proposed: A linear-quadratic regulator (LQR) is enabled in contact mode to achieve an active vibration control. A model predictive control (MPC) algorithm is used in backlash mode for an optimal traverse. However, all states are assumed to be measured and no disturbances are taken into account. The concept of MPC is further refined in [12], where a nonlinear system model valid in both contact and backlash mode is presented. The experimental validation reveals the sensitivity to parametric uncertainty.

Common to these approaches is the necessity to precisely know the backlash size and especially avoid underestimation. The analyses known to the authors are concerned with backlash sizes smaller than 20°.

2. Measurements and multibody system simulation of an exemplary slewing drive

In this article, a modular control strategy based on a model-based design approach is presented, which effectively limits the torsional load of elastic drivetrains subjected to large backlash. The reduced dynamical overloads prevent mechanical failure of load-bearing machine elements and thus increase the drive’s reliability. This can be achieved by active damping of torsional vibrations and a novel backlash traverse strategy. The proposed control strategy facilitates practical implementation due to the necessity to provide only two incremental encoders.

First, the system dynamics of an exemplary slewing drive are analysed using measured data and a fully flexible multibody system simulation model. The latter allows to derive a simple reduced order model, suitable for designing a full-state feedback controller and a state estimator for the nonlinear system. Robustness and achievable load reduction are investigated in subsequent steps. The mathematical formulation is based upon a drive with a single motor for the sake of clarity. An extension to multi-motor drives is straightforward.
by following a jerk-limited speed profile. As a consequence of a defensive setpoint setting, the reversing
time is currently approx. 16 s. Nevertheless, the measured data reveal a severely increased load, which
can be attributed to the gear backlash:
As soon as the load direction or the rotational direction is reversed, the tooth flanks previously transmitting
the torque disengage and the backlash of all gear stages is run through until the former non-operating
flanks touch. The increase in load is thereby caused by two independent mechanisms:
1. The collision during the initial contact of the former non-operating flanks. The magnitude
   of the mean torque throughout the impact approximately follows from the change in mo-
   mentum $M = \Delta L/\Delta t$ with the angular momentum $J\omega$.
2. The excitation of the lowest torsional vibration mode of frequency $f_0$, leading to higher
   loads subsequent to the reestablishment of the flank contact.

The current PI speed control structure is neither capable to effectively limit the speed difference during
impact nor the subsequent torsional overload.

Based on the system’s technical drawings, a fully-flexible multi body system (MBS) simulation model
has been set up using the program environment SIMPACK. The MBS model, including its validation
process, is described in [13] as well as an in depth-analysis of the effects associated with gear backlash.
As the simulation results of one reversal of the rotational direction in figure 3 indicate, there is a good agreement between measurement and simulation. The detailed MBS model will serve to evaluate the possible improvements of a new control strategy being presented in the subsequent sections. But first, the model allows to derive a simple low-order system representation suitable for the intended model-based controller design.

Analyzing the distribution of the system’s rotational inertia reveals that the input shaft assembly (consisting of induction machine, coupling and brake disc) and the excavator superstructure (load side) represent approximately 99 % of the drivetrain’s total rotational inertia. Consequently, the gear stages and shafts may be modelled as an idealized massless torsion spring. This leads to the two-mass system depicted in figure 4, with a torsional shaft stiffness $k = 585\,\text{Nm/rad}$ and the rotational inertias $J_1 = 12\,\text{kgm}^2$ and $J_2 = 26\,\text{kgm}^2$ of motor and load, respectively. In figure 5 the transfer functions of the two-mass system are compared to the detailed MBS model. A sufficient agreement can be seen in the low-frequency range up to 10 Hz. The torsional natural frequency $f_0 = 1.34\,\text{Hz}$ of the two-mass system corresponds to that of the MBS model.

As the torsional damping of most drives is small and hardly known exactly, it is neglected in the design phase but included in the final MBS simulations (section 4) and the robustness assessment (section 3.3) as uncertain parameter. This allows to extend the two-mass system by the known dead-zone backlash model (see figure 4), which is valid for shafts without damping only [14]. Consequently the angular difference between motor and load

$$\varphi_d = \varphi_1 - \varphi_2 = \varphi_s + \varphi_b \quad (1)$$

consists of the elastic twist of the drivetrain $\varphi_s$ (proportional to the torque transmitted) and the position in the backlash $\varphi_b$. The latter is bounded by the backlash size $\alpha$ and remains constant at either of both limits in case of flank contact.

**Figure 3.** Measured data from multiple reversals of the rotational direction compared to the multibody system simulation results – the absolute value of the motor speed is shown to facilitate the visual comparison

**Figure 4.** Dynamic model used for the control design (left) and nonlinear characteristic representing the backlash (right)
3. Active vibration control strategy

Due to the described shortcomings of the widespread PI control, a different control structure is pursued. Figure 6 summarizes the modules of the proposed control system and the interfaces to the actual drive system.

The main objective is to actuate the drive by means of a motor torque \( M_1(t) \) that ensures tracking of the slewing speed reference \( \omega(t) \) without excessive overload due to backlash-induced vibrations. The torque setpoint value is slightly delayed by the induction machine’s underlying vector control. The latter can be modelled as first-order delay with a time constant \( T_{VC} \) of several milliseconds.

The feedback control relies on measurements at the drive. Usually the rotation angle \( \varphi_1 \) of the motor is measured, additionally the slewing angle \( \varphi_2 \) of the load side is assumed to be available. The limited resolution of incremental encoders causes a quantization error of the measurements.

The utilized linear quadratic regulator (LQR) is described in section 3.1. Subsequently, section 3.2 is dedicated to the state estimation process using a Kalman filter. A robust and practical approach for a fast and yet smooth traversal of the backlash gap is presented in section 4.

3.1. LQ-optimal state feedback

In contrast to the PI control, which is based solely on the feedback of motor speed, additional kinematic quantities are fed into the control algorithm here. This full state feedback will enable an active damping of torsional drivetrain oscillations. [4]
The starting point is the state space representation of the system model shown in figure 4:

\[ \dot{x}(t) = A_{LQ} x(t) + b_{LQ} u(t) \quad \text{with} \quad A_{LQ} = \begin{bmatrix} 0 & 0 & -k/J_1 \\ 0 & 0 & k/J_2 \\ 1 & -1 & 0 \end{bmatrix} \quad \text{and} \quad b_{LQ} = \begin{bmatrix} 1/J_1 \\ 0 \\ 0 \end{bmatrix}. \quad (2) \]

Herein the state vector \( x = [\omega_1 \, \omega_2 \, \phi_s]^T \) includes the angular velocities of motor and load, as well as the shaft twist. The system matrix \( A_{LQ} \) represents the dynamics in terms of the rigid body motion and the torsional mode of vibration, when the tooth flanks engage, i.e. in contact mode. The input vector \( b_{LQ} \) describes how the motor torque \( u(t) = M_1(t) \) is acting on the drive.

In order to arbitrarily manipulate the system’s poles – in particular the damping of the torsional mode – full state feedback is considered. This leads to the linear control law \( u(t) = -k x(t) \). The gain vector \( k \) is chosen to minimize the quadratic cost functional

\[ J(t) = \int_0^t x(\tau)^T Q x(\tau) + u(\tau)^T r u(\tau) \, d\tau \quad (3) \]

and thus defines an LQ-optimal controller [15]. The weighting matrix \( Q \) and the scalar value \( r \) serve as tuning parameters to achieve favourable control characteristic, as they balance the relative importance of states deviating from zero and the resulting control action.

As the rotational speeds of motor and load are required to track some reference slewing speed value and reach the desired steady state value, a forward gain matrix

\[ F_w = c_{LQ} (b_{LQ} k - A_{LQ})^{-1} b_{LQ} \quad (4) \]

is introduced [16]. The output vector \( c_{LQ} = [1 \, 0 \, 0]^T \) defines the state to be tracked. To maintain the requested rotational velocity, even if a non-zero load torque is acting, the disturbance gain matrix

\[ F_z = F_w c_{LQ} (b_{LQ} k - A_{LQ})^{-1} b_z \quad (5) \]

is included [16]. The input vector \( b_z = [0 \, -1/J_2 \, 0]^T \) describes how the load torque acts on the system.

The resulting overall control law including full state feedback, setpoint tracking and disturbance rejection becomes

\[ u(t) = -k x(t) + F_w u(t) + F_z z(t). \quad (6) \]

The realization of the control structure relies on the availability of the state vector \( x(t) \) and the load torque \( z(t) = M_2(t) \). While both rotational velocities may either be measured directly or processed from angle measurements, the determination of the elastic shaft twist \( \phi_s \) is more involved. As stated in equation 1, the backlash nonlinearity imposes an uncertainty \( (\phi_b) \) to the shaft twist. The magnitude of the backlash can substantially exceed the shaft twist angle, thus requiring special attention during the state estimation process described below.

### 3.2. State estimation

The estimation of the state vector can be accomplished by means of a soft-switching continuous-time Kalman filter. A system model in state space formulation

\[ \dot{\hat{x}}(t) = A_{KF} \hat{x}(t) + b_{KF} u(t) + \nu_d \quad (7) \]

forms the basis. The state transition is described in terms of a deterministic portion, capturing the system’s internal dynamics and the effects of external inputs, e.g. the actuation due to control. Effects that are not captured by the estimation model, but are present in the actual physical system, are modelled as stochastic state disturbances \( \nu_d \). This includes unmodelled high-frequency dynamics or forced vibration due to some excitation.

Additionally, measurements taken from the actual system

\[ y(t) = C_{KF} x(t) + \nu_n \quad (8) \]
are also assumed to be corrupted by measurement noise $\mathbf{v}_m$. All stochastic processes are supposed to have white noise characteristics with covariance matrices $\mathbf{V}_d$ and $\mathbf{V}_n$ for simplification [17]. In the described application, measurements $\mathbf{y} = [\varphi_1 \varphi_2]^T$ are taken at the motor and at the load of the drive. The specific state vector of the estimation model used in equation 7 is based upon the two mass system shown in figure 4.

$$\ddot{\mathbf{x}} = \begin{bmatrix} \ddot{\varphi}_1 \\ \ddot{\varphi}_2 \\ \ddot{\varphi}_3 \\ \ddot{\varphi}_4 \\ \ddot{\varphi}_5 \end{bmatrix} = \begin{bmatrix} \ddot{\varphi}_5 \\ \ddot{\varphi}_4 \\ \ddot{\varphi}_3 \\ \ddot{\varphi}_2 \\ \ddot{\varphi}_1 \\ \ddot{\varphi}_5 \end{bmatrix}$$

(9)

All estimates are denote by the tilde. In contrast to the state-space model utilized in section 3.1 for controlling the system, the filter model involves both measured states. Due to the specific requirements of a system containing backlash, also the angular position in the backlash gap $\varphi_b$ is included. Finally, the Kalman filter allows to estimate the load torque acting on the drive. [8]

The state vector used by the controller is obtained from the estimated states using a Boolean matrix

$$\mathbf{x}_{LO} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & -1 & -1 & 0 \end{bmatrix} \mathbf{x}_{KF} .$$

(10)

The dynamical properties of the drivetrain are expressed in terms of the system matrix

$$\mathbf{A}_{KF,co} = \begin{bmatrix} 0 & 0 & -k/J_1 & k/J_1 & k/J_1 & 0 \\ 0 & 0 & k/J_2 & k/J_2 & -k/J_2 & -1/J_2 \\ 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}.$$ 

(11)

which is valid if torque is transmitted, i.e. contact of tooth flanks is established. If motor and load are decoupled during backlash, the system dynamics change, leading to a different system matrix

$$\mathbf{A}_{KF,bl} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & -1/J_2 \\ 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 1 & -1 & 1/dt & -1/dt & -1/dt & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}.$$ 

(12)

No torque will be transmitted and thus the shaft twist becomes zero. This enables to augment the calculation of the backlash position, as $\ddot{\varphi}_1 - \ddot{\varphi}_2 = \ddot{\varphi}_b$ holds true. The additional parameter $dt$ governs the dynamics of the correction of $\ddot{\varphi}_b$. The actuation $\mathbf{b}_{KF} = [1/J_1 \ 0_{1,5}]^T$ due to the input torque is unaffected by the backlash nonlinearity.

The hybrid characteristics of drives subject to backlash with two distinct modes (contact and backlash) leads to a switching observer, as proposed in [8]. The switching between the two models (equation 11 and equation 12) may be realized by comparing the angle difference $\varphi_1 - \varphi_2$ to the backlash size.

In order to avoid estimation errors associated to harsh switching between different dynamic models as in [8], a different approach is presented. A smooth transition of state estimates can be accomplished by means of a Kalman filter

$$\dot{\mathbf{x}} = \mathbf{A}_{KF}\mathbf{x} + \mathbf{B}_{KF}\mathbf{u} + \mathbf{K}_{KF}(t)(\mathbf{y} - \mathbf{C}_{KF}\mathbf{x})$$

(13)

with a time-varying gain matrix [17]

$$\mathbf{K}_{KF}(t) = \mathbf{P}(t)C_{KF}^TV_n^{-1}(t).$$ 

(14)

In addition to the usual time-dependency of the estimation error covariance $\mathbf{P}$, expressed as [17]

$$\dot{\mathbf{P}}(t) = -\mathbf{P}(t)C_{KF}^TV_n^{-1}(t)\mathbf{C}_{KF}\mathbf{P}(t) + \mathbf{A}_{KF}\mathbf{P}(t) + \mathbf{P}(t)A_{KF}^T + \mathbf{V}_d(t)$$

(15)
here also the noise covariances $V_n(t)$ and $V_d(t)$ are explicitly chosen to vary. After every switching event between the two models $A_{KF,co}$ and $A_{KF,bl}$ a transition phase is defined, during which the noise covariances gradually approach its final values for contact mode or backlash mode. This effectively reduces the impact of the filter’s fast correction term $K_{KF}(y - C_{KF} \hat{x})$ and puts emphasis on the slower system model.

The estimation performance of the proposed soft-switching Kalman-Filter can be analysed be means of an MBS simulation of the drive. Due to the separation principle, the simulation of the state estimator does not rely on the use of the state-feedback controller [15]. Thus, the simulation data of the PI-controlled load case described in section 2 is used. Only the motor and load angle, as well as the control input are provided to the Kalman filter. The estimation results and the actual signals taken from the MBS simulation are compared in figure 7.

It is important to note, that the simple internal model of the Kalman filter (two-mass system) is distinct from the detailed MBS model. The latter one includes the realistic distribution of inertia across the drive components, as well as the individual backlash of each gear mesh. Moreover, the MBS-model introduces higher frequency dynamics and disturbances from gear meshing frequencies. Finally, the angle “measurements” are artificially quantized to emulate the effect of a limited angular resolution of incremental encoders with 5000 pulses per revolution (ppr) at the motor and 50 ppr at the load side.

As the results in figure 7 indicate, both rotational velocities are estimated correctly. The reduced resolution of the load angle limits the feasible dynamics of the load speed estimation.

The backlash traverse during the simulation is apparent in the lower right diagram. Even during this phase the filter provides a reliable estimate of the elastic shaft twist $\varphi_s$ (top right diagram). Herein the small step at $t = 2$ s is caused by the switching from contact mode to backlash mode. The higher frequency signal components present in the MBS simulation (e.g. due to resonance with the second torsional natural frequency at $t = 12$ s) are effectively suppressed.

![Figure 7](image_url)

**Figure 7.** Estimated states and actual values during a reversal of the rotational direction, simulated by means of the MBS model under PI control
3.3. Active damping capability and robustness

One goal of the presented control concept is to increase the damping of the first torsional mode, which can be excited by load fluctuations or the gear meshing frequency of the low speed gear stages. Figure 8 shows the amplitude of the gearbox’ shaft torque, when torque fluctuations of defined frequency act at the load side. The standard PI controller is compared to the full state feedback control based on estimated states (LQG control). Apparently, the shaft torque magnitude can be decreased by up to 50 %. The maximum improvement is achieved if the load torque fluctuations match the system’s first torsional natural frequency.

Special attention has to be drawn to the effects of parametric uncertainty. It is well known, that LQR and Kalman filter have guaranteed stability margins, while their combination (LQG) does not [18]. Any model imperfection and especially time delays can severely degrade control performance or even cause instability. Therefore the parameters of the controlled system are complemented by margins of uncertainty (table 1). The multitude of frequency response functions resulting from the uncertain parameters are represented by shaded areas in figure 8. The load sensitivity functions in the left diagram suggest, that considerable damping can be accomplished even if the control model significantly deviates from the actual system. The transfer functions from load disturbance to control effort in the right diagram reveal the higher bandwidth of the LQG controller, but indicate no excess control actions due to the load variations.

![Figure 8. Frequency response functions indicating the active vibration damping capabilities (left) and the associated control effort (right) for PI control and LQG control considering parametric plant uncertainty](image)

**Table 1.** Parametric uncertainty of the drivetrain

| parameter          | nominal value | relative uncertainty |
|--------------------|---------------|----------------------|
| rotational inertia $J_1$ | 12 kg m²      | ±20 %                |
| rotational inertia $J_2$ | 26 kg m²      | ±20 %                |
| torsional stiffness $k$   | 585 Nm rad⁻¹  | ±50 %                |
| damping constant $d$       | 0.5 Nm s rad⁻¹| −100 % + 1000 %      |
| time constant $T_{VC}$    | 10 ms         | ±100 %               |

4. Backlash traverse strategy

The second objective of the slewing speed control is to reduce the overload and the excitation that originate from the backlash traverse. As stated in section 2, the speed difference between load and motor during flank contact is decisive for these effects and thus should be limited.

A prerequisite for the proposed approach is the coarse knowledge of the backlash size. It may either be gained from measurements, estimated using a separate Kalman filter [8] or processed from the signal $\hat{\phi}_b$ introduced in section 3.2. Furthermore, the angular velocities and positions of motor and load are required.
The backlash traverse strategy shown in figure 9 consists of three phases:

Ⓐ Fast backlash traverse

First, the motor speed reference value is reduced by $\Delta \omega_s$ relative to the load speed $\omega_s$. The choice of the speed difference depends on the admissible control effort and the desired duration of the backlash traverse.

Ⓑ Preparation of flank contact

If the backlash position $\varphi_b$ exceeds a threshold value, the speed difference between motor and load is reduced to $\Delta \omega_b$ in preparation of the expected flank contact. The speed difference is chosen to traverse the remaining backlash gap sufficiently fast and yet limit the ensuing overload.

Ⓒ Contact phase

After re-establishing the flank contact in all gear meshes, the targeted final slewing speed can be reached.

Finally, the overall control structure consisting of the LQG control and the backlash traverse strategy is used in an MBS simulation for reversing the drive’s rotational direction to demonstrate its functionality. In figure 10 the motor speed and shaft torque are shown and compared to the present PI controller. Currently, the process takes 16 s due to a defensive jerk-limited reference speed profile. A faster reversal is preferable. However, the simple adjustment of the PI controller’s reference speed profile to reduce the duration for a motor speed reversal to 12 s would result in an unacceptable strain in the drivetrain (see figure 10). Moreover, the PI control is unable to reach the target speed without overshooting, thus entering backlash again.

Figure 9. Reference speed profile for backlash traverse (left) and corresponding backlash position (right)

Figure 10. MBS simulation of three different reversals of the rotational direction – the absolute value of the motor speed is shown to facilitate the visual comparison
In contrast, the proposed control is capable of reducing the time for reversing the rotational direction by 25%, while almost completely avoiding torque overshoot. The increased shaft torque in the second half of the simulation results from the higher acceleration torque necessary for the faster reversal.

5. Conclusion

The present article provides a practical solution to two common problems in drive technology. Geared transmissions with high reduction ratios cause both large backlash and elasticity in drivetrains. As a result, unwanted vibrations can occur and high dynamic loads may arise. A modular control strategy is proposed, which is based on full-speed feedback in order to actively dampen torsional vibrations. A soft-switching state estimator is presented, allowing to reconstruct the necessary states for feedback based on two measured kinematic quantities. The overload resulting from the abrupt contact of tooth flanks after a backlash traverse is reduced by means of a simple and robust strategy. MBS simulations and a robustness assessment in the frequency domain prove the concept’s effectiveness and reliability. The practical implementation is facilitated due to the limited sensor demand and the moderate computational cost. Future work will cover the realization on an actual drive system.

6. References

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