Simulation Model of a Torque-Limiting Clutch with Adjustable Design Parameters to Investigate the Release Behavior

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Abstract. For the design of torque-limiting clutches in the early phases of product development, simulation models are necessary which allow an adjustment of the release behavior by defining the design. Known and accessible simulation models do not allow profile variation of torque-limiting clutches.
Based on an analysis of a torque-limiting clutch, quantitative relationships between the design parameters and the release behavior were investigated and implemented in a simulation model. The developed model represents the torque defined by the profile as a function of the angular position and the direction of rotation. The simulation model is parameterized for instance via the geometry of the profile and the spring stiffness. The model takes into account friction behavior across all components.
With this model, the behavior of the torque-limiting clutch can be simulated and an adaptation of the shape of the clutch to its function can be estimated. Verification results show that the model can be used for early design and application in testing.

Introduction

In product development, design parameters are defined in order to realize a desired function [1]. This requires an understanding of the relationship between design parameters and the resulting behavior.

For the derivation of target values of the design parameters (described in the following as design targets) and the testing of development variants during development, both test benches and simulations are used. Increasingly, simulations are used to predict design parameters. Simulations are used to build up an understanding of the system and to define design targets. This is helpful if subsystems are not yet physically available and only simulation models can be used for early validation in the development process. The key is not only to be able to investigate principles, but also to map design details in the simulation model in order to enable a preliminary assessment of the components and to define design targets. Such simulations exist for the testing of subsystems in the automotive, aerospace and electrical equipment industries, among others. The simulation models of the subsystems provides the basis for investigations on test benches. Particularly for safety-critical components, such as overload or torque-limiting clutches, knowledge of the relationships [2; 3] between design and function is important for preliminary design.

Challenge. In the state of research, simulation models of overload clutches already exist [2–6]. The available models mainly represent clutches with notches. Other shape variants are rarely considered by simulations in commercial programs.

These existing models can only be used to a limited extent for the prediction of profile shape variants of the existing torque-limiting clutch. It is not possible to adapt the available models for use in prediction and bench testing. Known and accessible models do not fully represent the relationship between design parameters and release behavior. The developtment of a simulation model of the existing torque-limiting clutch with variable design parameters is necessary.

Aim. The aim of this paper is to develop a simulation model for mapping the torque-limiting clutch release behavior. The relationship between design parameters and release behavior is analyzed on the basis of an existing torque-limiting clutch, which is designed as a locking element clutch. The determined relationships are transferred to a simulation model and the functionality is compared with experiments on a test bench. The modeling is exemplified by the torque-limiting clutch of a cordless drill/driver of the type GSR 10 8-2-LI Plus (Robert Bosch GmbH), see Figure 1 and 2.

implemented and parameterized as a grey box model.

The model building and simulation program used is Matlab Simulink and Simscape. The determined relationships are implemented in formulas and relationships within a new Simscape component or Simulink block.

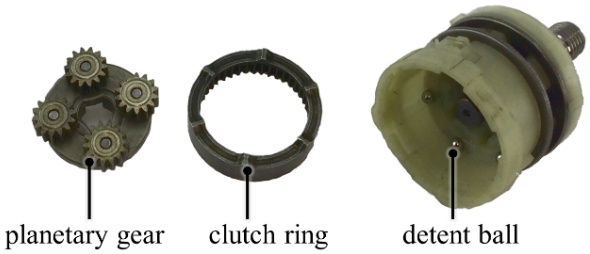

Figure 1: The exemplified system of the mechanical torque-limiting clutch consisting of a clutch ring with detents and the planetary gear. [7]

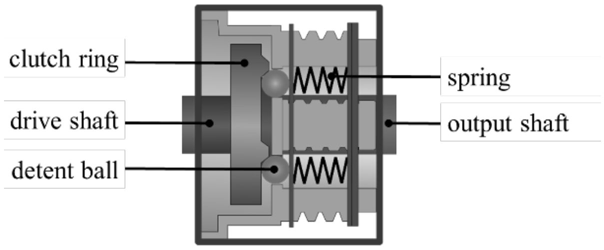

Figure 2: Representation of the components of the planetary gear with clutch [11]. The spring force forces the detent balls against the detent on the clutch ring to prevent them from rotating.

1 Materials and Methods

For model building, a three-step procedure based on [8–10] with system analysis, implementation and validation is used.

Clutch. The clutch (see Figure 2) limits the torque mechanically. The clutch consists of detents on a clutch ring, which is integrated into the third stage of the planetary gear unit. The detent balls are pressed onto the clutch ring by springs. The detent balls and springs are guided in the clutch housing, where they can support the applied torque. When the torque applied from outside (output shaft) increases, the springs are further compressed, causing the detent balls to move up along the detents. As soon as the detent balls move above the detents, no more torque can be supported. The clutch then releases and limits the maximum transferable torque of the output shaft to a defined value. This torque limit can be adjusted via the preload of the springs. The components of the clutch are shown in Figure 1 and Figure 2.

1.1 System Analysis

The basis for modeling is an analysis of the existing clutch. For this purpose, the clutch is broken down into individual subsystems and the relevant parameters are identified. In addition, the relationships between design parameters and behavior are derived in the form of equations. These relationships in the form of equations are based on a force equilibrium. The relationships and the defined parameters are then used in the model building.

Based on an analysis of the clutch, quantitative relationships are derived.

1.2 Implementation

Based on the analysis, the physical simulation model is implemented and parameterized as a grey box model.

For parameterization, the geometry and the relevant parameters of the clutch are recorded. The spring stiffness parameters is determined using a tensile testing machine. The profile geometry is recorded via a digital microscope with 3D measuring function. The geometry of the torque-limiting clutch is mapped via the profile shape. The occurring friction was summarized in two parameters. The friction was assumed to be sliding friction. The friction parameters are not precisely assigned to the friction contact. For the simulation, it is assumed that the friction conditions do not change significantly at different clutch stages and operating points.

1.3 Test Setup for Validation

The simulation model is validated with measurement data from a test rig, with actuators and measuring shafts (sensor 1 and sensor 2) on the input and output sides (see Figure 2). The validation of the virtual clutch is carried out in comparison with mechanical experiments at different operating points (defined by the preload levels 3, 5, 10, 15) on a test bench.

The verification is performed on the powertrain test bench (see Figure 3). For this purpose, the clutch between two load machines (actuator 1 and actuator 2) is investigated. In order to evaluate the performance of the clutch, the time signals are compared with the simulation results. The test setup is shown in Figure 3.

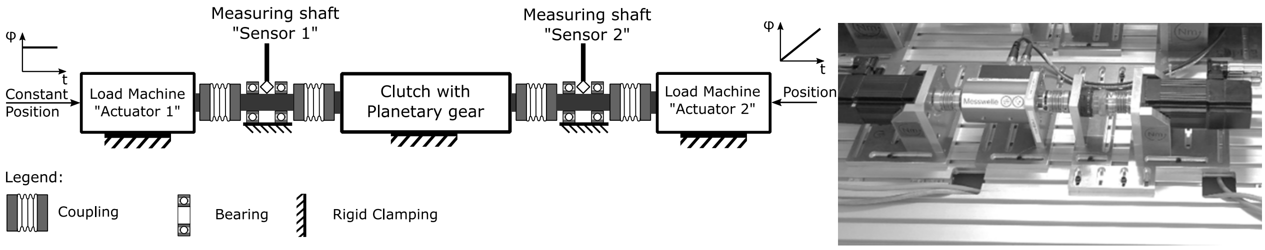

Figure 3: Test bench for comparison with simulation. The test bench contains the clutch between two load machines and additional measuring shafts.

In order to achieve a high degree of comparability, the relevant properties of the test bench are taken into account in the simulation. Thus, the inertias and stiffnesses of the test bench components such as measuring shafts and connecting elements are integrated into the simulation.

For this purpose, the mechanical properties can be implemented in the simulation based on data sheets. The evaluation is based on characteristic points in the time course of torque and speed. The maximum clutch torque and the negative torque are used as evaluation variables. The comparison is made by evaluating the torque and angular position signals at the interfaces of the clutch. In order to ensure simple conversion of the time curves, angle inputs were used in addition to speed inputs. For a speed-independent representation, the results are also plotted against the angular position of the clutch. For this purpose, the angular positions are used to infer the current position of the clutch. The time curve can then be converted into an angular curve. This allows the function of the clutch to be assigned to individual sections of the elevation. The relationship between design and function thus becomes notable.

2 Results

First, the system is presented and the analysis results are shown. Then, the implementation of the relationships of the model is shown and the simulation results are compared to experiments.

2.1System Analysis

The torque-limiting clutch of the cordless drill is integrated in a planetary gear. The clutch limits the torque mechanically and consists of six balls mounted in the plastic housing, which prevent the clutch from rotating via a profile. The design parameters (geometry, material) of the clutch ring, the springs and the detent balls define the functional properties of the clutch and thus the behavior with regard to the safety function. It is assumed that the detent balls are always in contact with the clutch ring.

In addition to the profile shape, the properties of the spring and the ball are important for the function. The torque depends on the position of the ball on the detent. The clutch torque was determined based on a balance of forces. The gradient at the contact point is determined by the position of the ball on the profile. The torque for the individual angular positions is determined via a following equation.

$$M_p(\varphi) = r_p F_c \left(\frac{\frac{\Delta z}{r_p \Delta\varphi} + \mu_k \, sign(\dot{\varphi})}{1 - \mu_k \, sign(\dot{\varphi}) \frac{\Delta z}{r_p \Delta\varphi}} \right) \tag{1}$$

$$M_s(\varphi) = r_p F_c \mu_s \, sign(\dot{\varphi}) \tag{2}$$

$$M_C(\varphi) = M_p + M_s \tag{3}$$

$M_C(\varphi)$ describes the torque of the entire clutch, $M_p(\varphi)$ the torque generated by the force redirection and the profile friction, $M_s(\varphi)$ the torque of the friction housung, r_p the radius of the profile track, $\frac{\Delta z}{r_p \Delta\varphi}$ the gradient at the contact point, μ_k the sliding friction between ball and profile, μ_s the sliding friction between the clutch ring and the housing and $sign(\dot{\varphi})$ the direction of rotation of the respective characteristic curve. The spring force F_c is determined by the spring stiffness and the compression due to the displacement of the ball and the preload biasing distance.

Measurements of the clutch using a digital microscope with a 3D measurement function show that the profile is more rounded than the generic trapezoidal shape presented in [12].

The friction occurring in the components was summarized in two parameters. There is no exact assignment of the friction parameters to the friction contact. For the simulation, it is assumed that the friction conditions do not change significantly at different clutch stages and operating points, and therefore parameterization via a test is expedient. The friction behavior was determined on the basis of a measurement of the clutch on a test bench.

2.2Implementation of the Clutch Model

The behavior of the torque-limiting clutch is embedded as a table of characteristic curves (as lookup-table) in simulation. These curves can be implemented as a Simulink block or as a Simscape component. The structure of the simulation is shown in Figure 4.

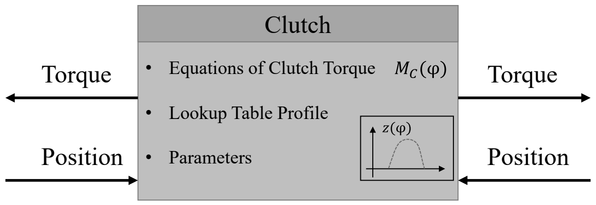

Figure 4: Implementation of the clutch in the simulation model.

Based on the determined relationships, the transmittable torque is converted into a characteristic curve as a function of the clutch position. These characteristic curves represent the torque of the clutch via the angular position of the clutch. Within the simulation, separate characteristic curves are used for the two directions of rotation of the torque-limiting clutch.

Other necessary properties for modeling the clutch are added in separate elements of the simulation. These include the inertia of the clutch ring and the planetary gear, as well as the backlash for the gear and clutch.

A pre-calculation of the characteristic curves is performed for the profile at hand before the simulation is carried out. First, the pre-calculation of the displacement of the ball, through the profile and the point where the profile and the ball contact, is performed. Using the formula, the maximum torque can be determined from the position of the ball.

These pre-calculations are prepared in characteristic curves as a lookup table for the simulation and thus allow a very performant calculation.

In this modeling, the two sliding frictions μ_s (clutch ring / housing) and μ_k (ball / clutch detent) are assumed. The sliding friction is assumed to be independent of the velocity additionally static friction effects were neglected here. Furthermore, small gear losses were considered in the modeling of the gearbox. Damping is also assumed within the planetary gear. It represents the contacts between the components and primarily serves to improve the simulation stability. The behavior of the clutch is shown in Figure 5.

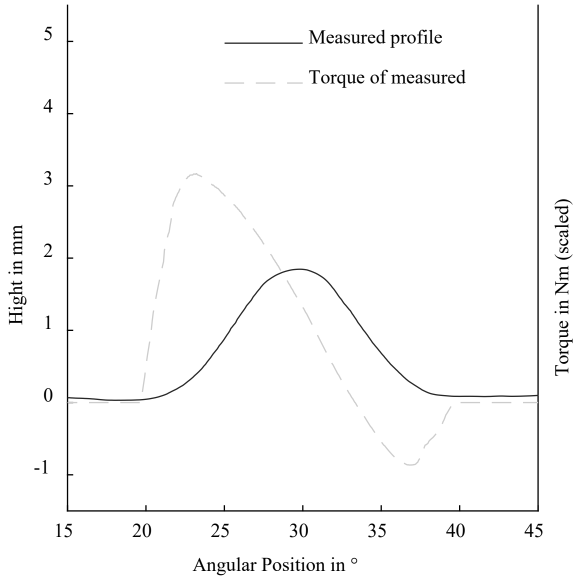

Figure 5: Measured profile of the clutch detents and torque of the clutch calculated via the simulation as a function of the angular position.

2.3Clutch Behaviour in the Simulation

The simulation of the torque-limiting clutch allows the analysis and initial estimation of the influencing parameters on the clutch behavior (see Figure 5). The asymmetry of the torque-on-angle curve is remarkable. This can be explained by the friction and the backlash. The frictions act against the drive direction and are additively superimposed with the redirected spring force.

At the rising flank, both the redirected force and the friction force point in the opposite direction to the direction of motion, which increases the total amount of force. At the falling flank of the detent, the two forces point in opposite directions, reducing the overall applied force. At the top of the detent, the slope is very shallow, and the friction force is high, making the friction force dominant over the redirection force.

As a result, we see a flattening torque curve in the area of the peak, instead of the zero crossing in a curve without friction. This zero crossing is shifted further into the range of the falling edge. The backlash (sometimes called lash, is caused by gaps between the parts, with result in lost motion) generates a negative torque pulse directly after the zero crossing (see Figure 6), since the direction of force changes there, the clutch ring is accelerated until it touches the other side of the backlash again.

As the moment of inertia of the clutch ring is very low, this occurs in a very short period of time and is therefore difficult to measure by torque measurement. In the simulation model, damping is applied to the backlash (with a translational hard stop). Since the profile continues to rotate in the backlash, the section of the profile is not visible in the torque curve.

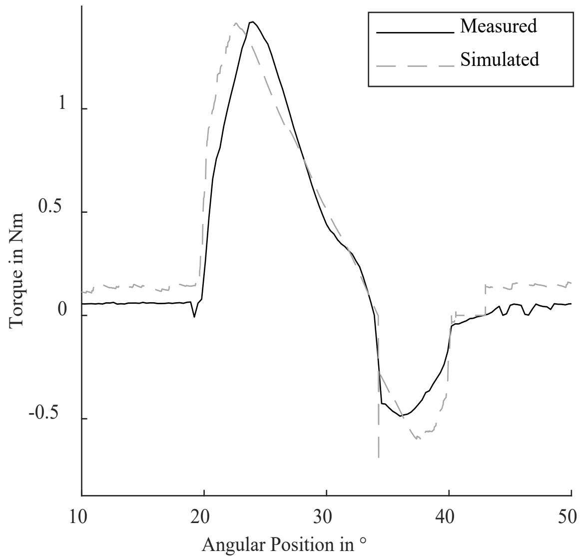

Figure 6: Torque of Simulation and Measurement on Ring Angle

2.4Validation and Model Abilitys

The simulation model replicates the relationship of the presented torque-limiting clutch between design parameters and functional behavior. A comparison between experiment and simulation is shown in Figure 7. The simulation calculates the transmittable clutch torque as a function of the angular position. The simulated clutch torque is compared with experiments on the test rig. The preload leves 5, 10 and 15 of the clutch were used for this purpose both on the test rig and in the simulation. Comparison of simulation with mechanical experiments shows good correlation for the present torque-limiting clutch. Figure 7 shows that the peak torque of the clutch can be predicted at different clutch levels with the simulation.

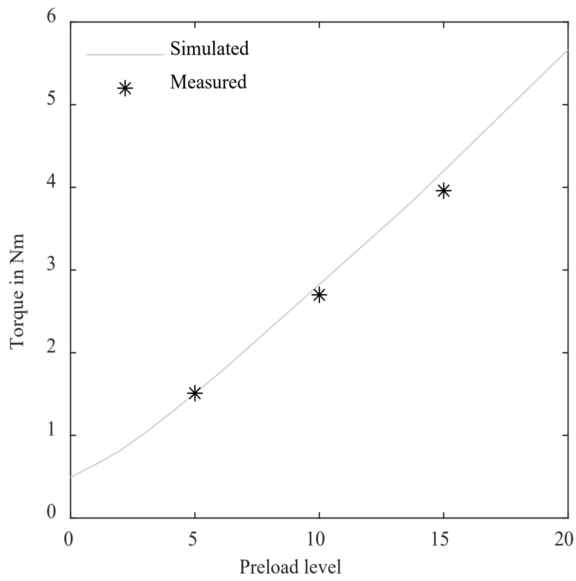

Figure 7: Relationship between preload level and peak torque of the clutch and simulation at different clutch levels

The simulation model allows the maximum clutch torques to be predicted as a function of the clutch level with a maximum deviation of 6%. The deviation is considered almost constant over the clutch stages. In the model, different design variants of the clutch can be simulated via different profile curves.

Customization. The model of the torque-limiting clutch nevertheless allows the customization of individual design parameters to be estimated by variation.

Different profile shapes can be considered in the simulation. When the geometry is changed, release behavior can be predicted. The behavior of an alternative clutch is shown in Figure 8. A comparison of the simulation results with generic descriptions [12], shows a high functional behavior of the simulation model. The simulation must be parameterized with multiple parameters. These parameters can be varied to estimate the behaviour of different clutches. Some of these parameters are simple values such as the profile radius, ball radius, preload distance, spring resistance, ball to profile friction and disk friction. The profile, on the other hand, is parameterized via the height as a function of the angle.

Profile Variation. Any reasonable profile design can be implemented. Figure 8 compares two different profile shapes with regard to their release behavior. Thus,

custom designs like trapezoids or notches and hight profiles from measurements can be used. Profiles with self-locking designs can create nonsensical simulation results. These designs would turn the physical clutch nonfunctional. Simulations are used to vary the profile shape for prediction to represent different behaviors.
The model is not validated for design changes in the clutch profiles. As long as there are no undercuts or longer vertical profile components in the geometry, a prediction of the behavior is possible.

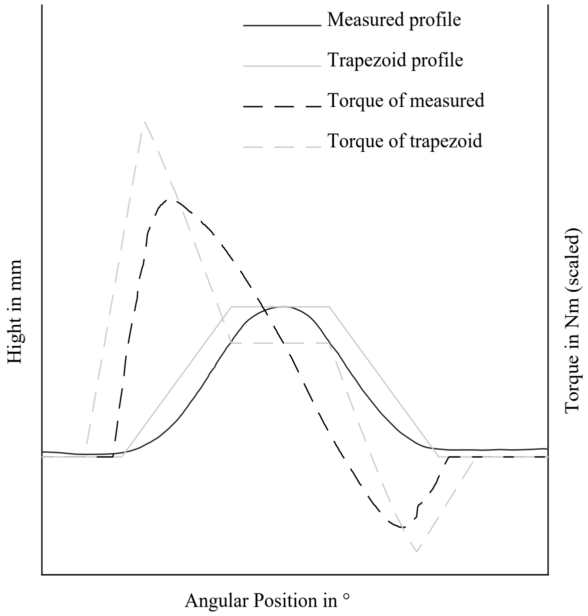

Figure 8: Relationship between the shape of the profile and the torque of the clutch when the geometry of the profile is customized.

3 Discussion

The simulation model is able to reproduce the behavior of the torque-limiting clutch. The model can be used for different clutchs and adapted profile shapes to predict the behavior.

The pre-calculation of the ball center of gravity leads to a fast execution of the simulation. The representation in a Simulink model enables real-time execution. Thus, the simulation can also be used as a virtual subsystem in mixed physical virtual studies.

One approach to validate development variants on test benches and to integrate them into the overall system is the X-in-the-loop (abbr. XiL) approach [13]. The XiL approach supports this integration and enables the functional testing of a component or a subsystem (System-in-Development) in test benches by mapping the remaining subsystems through suitable physical or virtual models [13] (as Connected Systems). Such test benches can be used to examine subsystems in a system context.

Limitations. The friction effects are implemented as simplified substitute effects, and do not fully represent the physical relationship. Furthermore, possible frictions are not considered at all surface pairs. The concrete behavior of the balls and the related friction behavior of the torque-limiting clutch can be modeled in more detail. It is still unclear how the balls behave in dynamic conditions. For example, the balls could lift off from the clutch ring. In this way, any interactions of the friction parameters with the functional behavior could be fully modeled and thus enable a more accurate prediction even in the case of changing material properties and the associated changes in the friction parameters. Static friction effects are also not part of the simulation model. Since the friction is not completely determined from the shape, a design of a clutch without a reference clutch, is not possible. If the parameters differ too much from the reference shape, the simulation model becomes increasingly invalid. A high quality of the analysis of the reference clutch is also necessary, since even small deviations in some parameters, such as the slope in the profile, have a large effect on the resulting torque.

4 Outlook

The tests have shown that the simulation model is suitable for mapping the relationship between the design parameters and the limited clutch torque.

If a higher reproduction quality is desired, further analysis and improvements to the models are necessary. Optimization is possible with regard to the behavior of the ball. Due to its real-time capability, the model is already suitable for use as a virtual subsystem for investigations in the context of XiL, for instance [14]. The use of the simulation model of the torque limiting clutch as part of a digital twin [15] can also be considered.

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