Modeling and Testing of the Hydro-Mechanical Synchronization System for a Double Clutch Transmission

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Abstract

Synchronization is a core component in the automotive powertrain. It uses friction and locking elements to synchronize the occurring speed difference during gear shifting. The optimization of this shifting process is of high interest in respect to fuel consumption and comfort considerations. Moreover, for the model-based calibration of automated transmissions, detailed simulation models of the synchronization system are also necessary. Highly accurate models allow simulation of nonlinear effects having a major influence on the shifting process. Currently, with less detailed models only rough estimations of the shifting process are possible, it has a reduced meaning for the precise calibration.

This paper uses a popular double clutch transmission (DCT) as the research object and presents the detailed hydro-mechanical synchronization model. Firstly, an introduction to the theory of the synchronization is given. Subsequently, a Modelica® based synchronization model consisting of hydro-mechanic actuators and gear shifting synchronizers is presented. Finally, these modules are discussed in detail and evaluated based on different simulations. A comparison with measurement data from a test bench is also included in the end.
Keywords: synchronization; hydraulic; gear shifting; double clutch transmission; physical modeling; automotive

1 Introduction

Due to the location of the synchronization in the automotive powertrain, this system has a crucial influence on the shifting quality. The shifting quality can be judged by:

- the duration of the shifting process
- the changes of vehicle longitudinal acceleration during shifting (shifting jerk)
- the oscillation to the powertrain
- the acoustic phenomena like shifting or impact noise

With conventional, less detailed models of the synchronization containing simple clutch elements as synchronization [1, 2], only three stages of the synchronization process is modeled:

- neutral position
- friction phase (synchronization)
- engaged position

In this paper a more complex simulation model of the synchronization is derived to describe certain detailed nonlinear phenomena during shifting (see section 2). Such a detailed modeling of synchronization is necessary for the model based calibration. The purpose of this calibration process is the adaption of control parameters to improve the shift quality between successive shifts. Furthermore an in-depth model provides the user with a fundamental understanding of the components composition principle and the system working function.

A 7-speed DCT with dry clutches is used here as the research object. For this transmission, a dynamic simulation model of the hydro-mechanical synchronization system is derived. This model could be used for the function development within the V-development process [3].

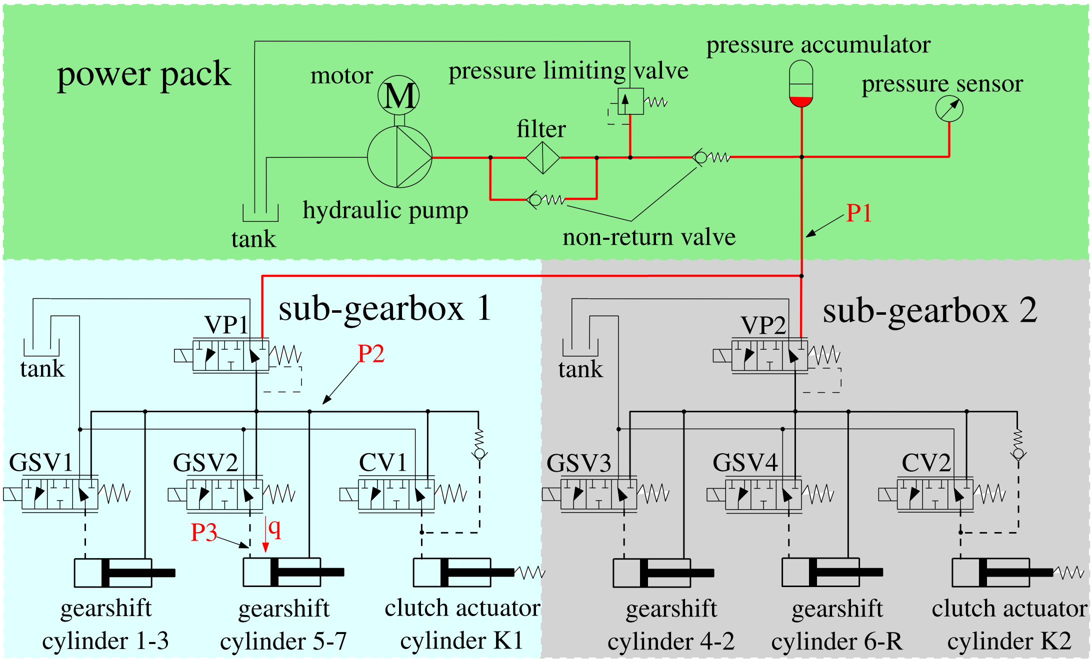

Fig. 1: Hydraulic system plan [4, 5]

In section 2 the basic components of the hydro-mechanical actuators are introduced and the synchronization process is described in detail. Then section 3 presents the simulation results of the physical model. The test bench measurements from an AMT with similar synchronization components are also compared. Finally, a summary and further research objectives are concluded.

2 Modeling

The whole synchronization system is divided into 2 parts: hydraulic and mechanical components. The hydraulic components are mainly supplying required oil pressure and flow while the remaining components are used to perform the mechanical actuator behavior and the synchronization process.

2.1 Hydraulic Components

The hydraulic subsystem consists of:

- a hydraulic pump
- magnetic valves
- gearshift cylinders

Hydraulic fluid is pumped from the tank to the pressure accumulator where it is stored under high pressure. The pump is controlled by a bang-bang controller which guarantees a pressure level between 40 and 60 bars [4]. When the oil circuit has got enough power to drive the gearshift cylinders, the magnetic valves will control pressure and flow of relevant branches.

There are mainly two types of magnetic valves included: pressure-control valves and flow-volume valves. The pressure-control valves are used to supply the corresponding sub-gearboxes under constant pressure levels. The flow-volume valves are used to control the movement of the gearshift and clutch actuator cylinders. The hydraulic plan is depicted in Figure 1, in which each flow-volume valve controls the left chamber of a gearshift cylinder while its right chamber is controlled directly by a pressure-control valve.

2.2 Mechanical Components

2.2.1 Synchronizer and Actuation Module

Synchronizers reduce speed difference through friction and locking elements during the gear shifting process. In this paper, a widely used single-taper synchronizer based on the "Borg-Warner" system (refer to [6]), shown in Figure 2, is used as a detailed example for the synchronization process.

The components of the synchronization are named (compare [7]):

① idler gears with needle bearings
② synchronizer hub with selector teeth and friction

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taper
③ synchronizer ring with counter-taper and locking toothing
④ synchronizer body
⑤ gearshift sleeve
⑥ transmission shaft

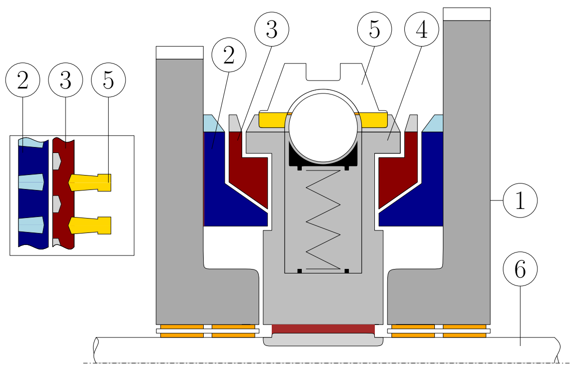

Fig. 2: Draft of the synchronization [6, 7]

During the synchronizing process, the selector fork supplies the gearshift force F_S for synchronization as the resultant of 4 forces exerted upon it: Shifting force F_C from the hydraulic cylinder, locking force F_{lml} from the detent pin, bearing friction F_{fl}, and acceleration force F_{al}, as expressed in Equation 1. The mechanic diagram of the shift actuator is presented in Figure 3.

$$F_S = F_C - F_{lml} - F_{fl} - F_{al} \quad (1)$$

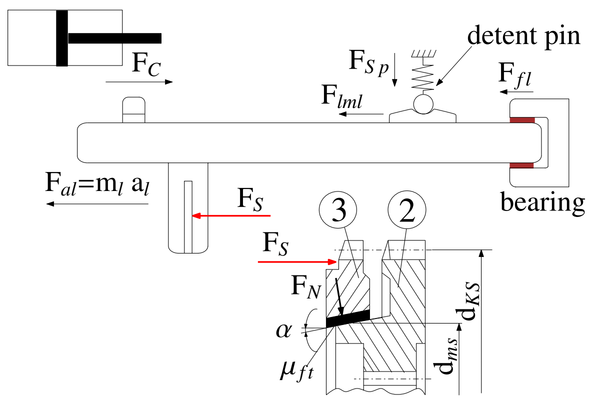

Fig. 3: Force diagram of shift actuator [7]

The detent pin showed in Figure 3 is designed to support the gearshift movement and guarantee determined positions. During the gearshift process from the neutral position to a shifted position, the detent pin introduces a counter force to the movement of the selector fork at the beginning and accelerates the fork after synchronization. This force characteristic can be calculated by Equation 2 and is depicted in Figure 4. The locking force depends on the spring force F_{Sp}, the ramp angle γ relative to initial basis and the friction angle δ_F acting against the movement direction [7, 8]

$$F_{lml} = F_{Sp}\, tan(\gamma + \delta_F) \quad (2)$$

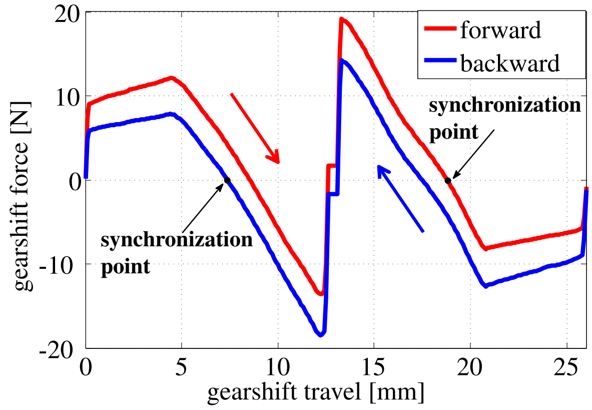

Fig. 4: Contour of ramp profile

The gearshifting process can be divided into five stages according to the gearshift position, speed difference, actuation forces and torques [6]. This classification is based on the assumption that at the beginning the gearshift sleeve is in the neutral position (see Figures 2, 3 and 5):

Stage 1: Gearshift force F_S causes an axial movement of the gearshift sleeve ⑤ and triggers the gearshifting process. The movement stops when the synchronizer ring blocks the gearshift sleeve.

Stage 2: The axial force is transmitted from the gearshift sleeve to the synchronizer ring ③, resulting in a friction torque T_R which is much larger than the gearing torque T_Z. At this stage the speed difference between the idler gear and transmission shaft will be reduced to zero.

Stage 3: When the speed difference is close to zero, the friction torque T_R vanishes. At this moment the synchronizer ring turns back to release the gearshift sleeve.

Stage 4: The gearshift sleeve begins to move until it encounters the synchronizer hub's ② external gearing. Speed difference increases again as the synchronizing torque diminishes.

Stage 5: The whole synchronization process is completed as soon as the gearshift sleeve toothing engages the synchronizer hub's gearing. The power flow is transmitted from the transmission shaft ⑥ to the gear ①.

Figure 5 shows the synchronization process with locking of the synchronizer ring and synchronizer hub.

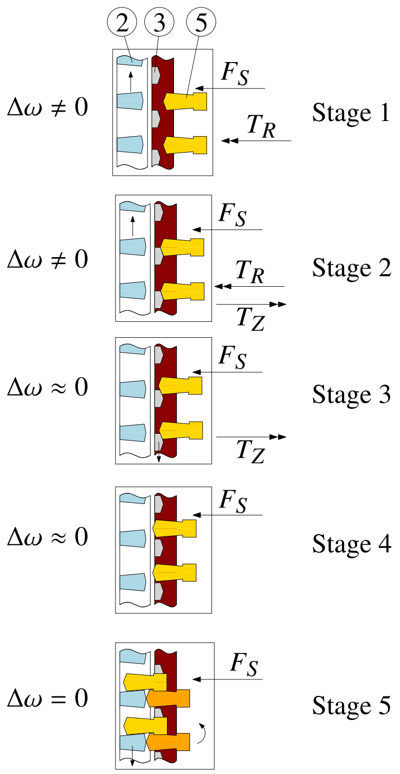

Fig. 5: Synchronizing process

2.2.2 Status Determination Module

This module is created based on Modelica®, it uses these 3 factors as mentioned above: the gearshifting position, difference speeds, actuation force and torque, to determine the synchronization process (Figure 6 shows the flowchart of status determination). The appropriate calculations of the friction torque T_R and gearing torque T_Z are also realized here.

The detailed torque values are changed according to the synchronization stages: The friction torque T_R, given by Equation 3 (applied to stages 1 and 2), is calculated through the gearshift force F_S, the number of friction surfaces j and some other geometric values. The gearing torque T_Z, expressed as Equation 4 (used in stages 2 and 3), is calculated by gearshift force F_S, clutch diameter d_{KS}, teeth angle β and friction μ_{lt} between gearshift sleeve and synchronization ring [7, 9, 10].

$$T_R = jF_S \frac{d_{ms}}{2} \frac{\mu}{sin\alpha} \tag{3}$$

$$T_Z = \frac{F_S d_{KS}}{2} \left(\frac{cos\frac{\beta}{2} - \mu_{lt} sin\frac{\beta}{2}}{sin\frac{\beta}{2} + \mu_{lt} cos\frac{\beta}{2}} \right) \tag{4}$$

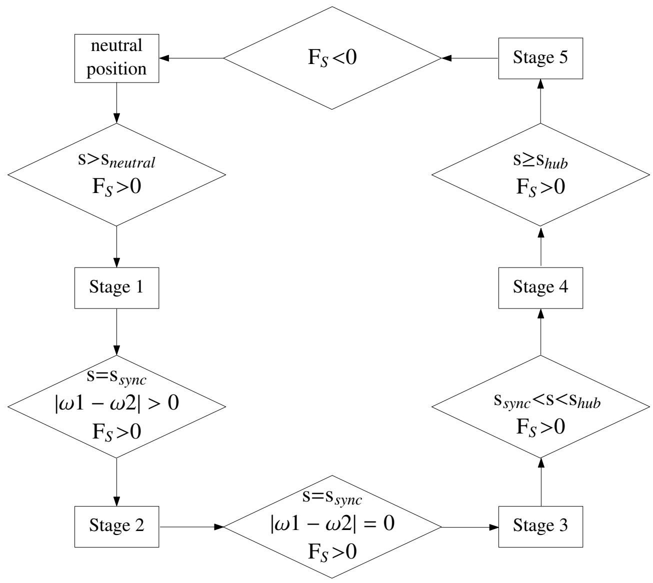

Fig. 6: Flowchart for status determination

2.2.3 Assembly of mechanical submodules

The mechanical subsystem consists of the 3 parts described above (compare Figure 7).

1) The gearshifting displacement part, used to simulate the movement of the selector fork
2) The synchronization part, functioning to simulate the synchronization process between synchronizer ring and synchronizer hub
3) The synchronization status determination and torques calculation part, working to determine the synchronization stages, calculate corresponding friction forces, and coordinate the gearshifting displacement part with the synchronization part

2.3 Modeling Result

Figure 8 shows the relevant physical model. The hydraulic components are modeled with hydraulic library HyLib® [11], the mechanical components with Modelica Standard Library (MSL) [12] and some new created blocks based on Modelica®. In order to simplify the model structure and improve the model portability, subsystems are built here. For example, *Gear_Selector* is used as a subsystem block, which stands for all mechanical components (see Figure 7).

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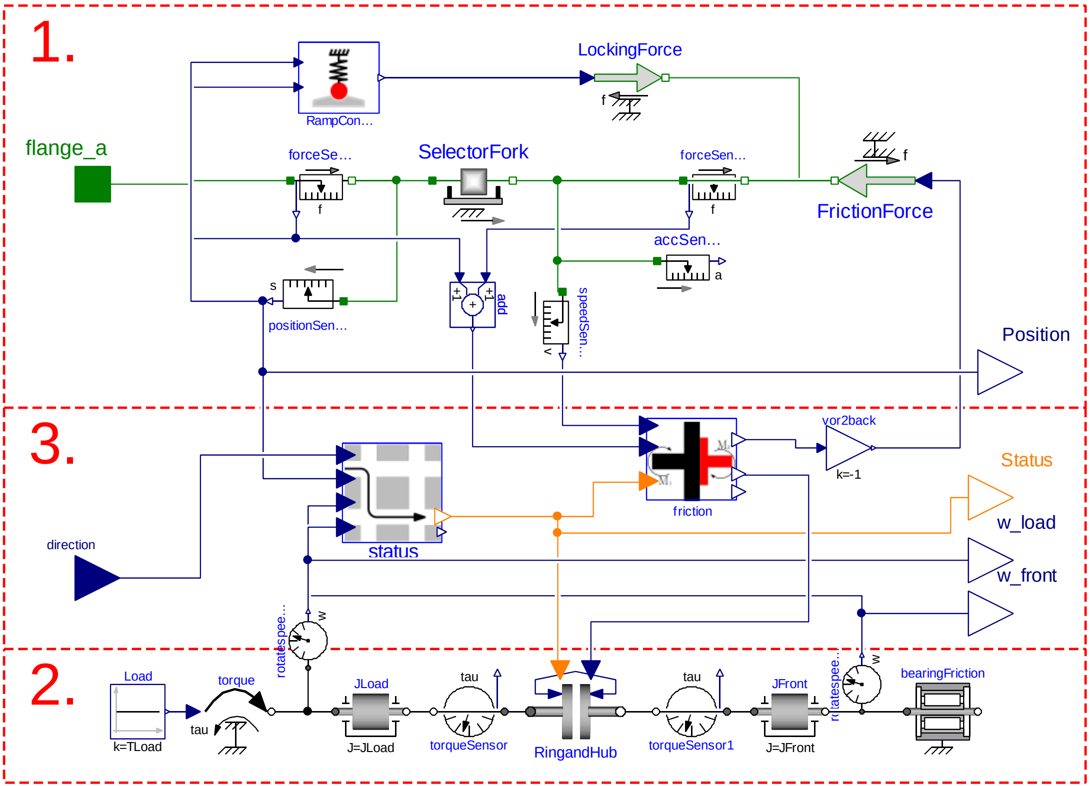

Fig. 7: Mechanical model

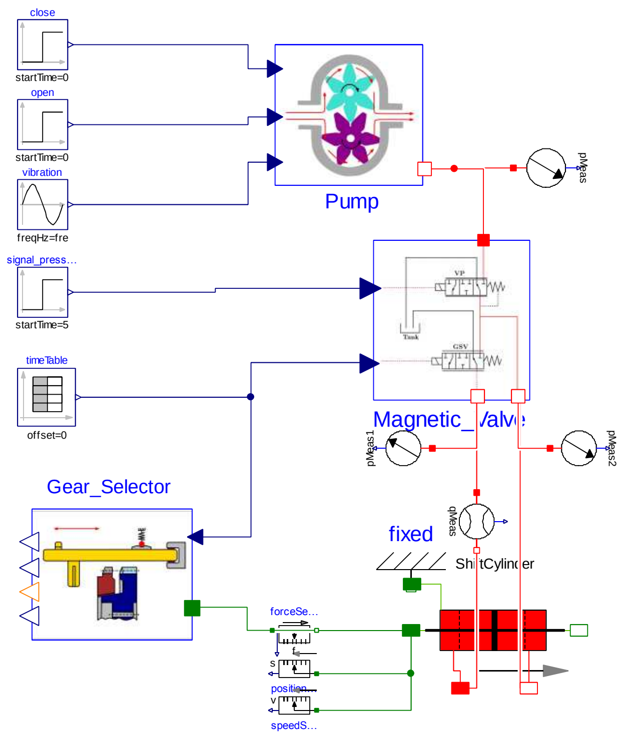

Fig. 8: Synchronization model

3 Testing

In order to verify this dynamic model's rationality and effectiveness, the following testing steps are carried out:

1) testing of the hydraulic model
2) testing of the mechanical model
3) testing of the whole hydro-mechanical model
4) comparison of the simulation results with real AMT test bench measurements

During testing, the dynamic model is driven under an open-loop control. Step- and constant-signals are used for stimulations (see Figure 8).

3.1 Hydraulic Model

The hydraulic supply circuit is first examined against measurement data from real DCT. In this process all magnetic valves are closed, only the oil pump is working. Simulation result, shown in Figure 9 depicts a small model error in comparison to the measurement data, the normalized root mean square error (NRMSE) of $e_{NRMS} = 4.9\%$. From beginning the pump is kept working until hydraulic pressure reaches the required value. Then the pump stops to wait for restart when pressure level drops, as the result of leakage in the whole hydraulic system, below a predefined threshold value.

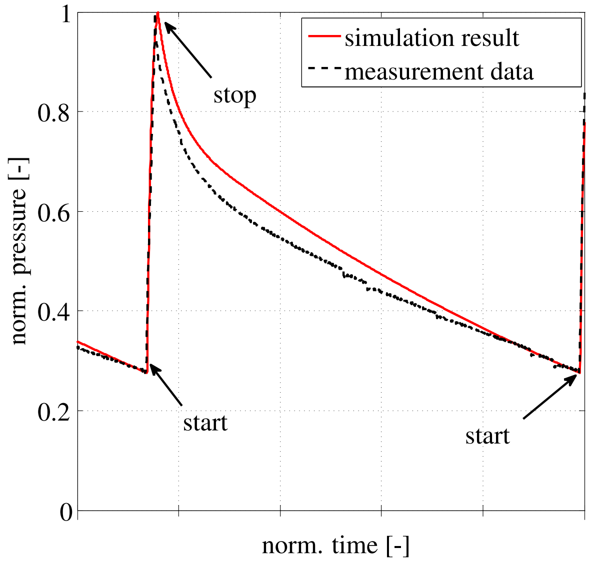

Fig. 9: Comparison of oil pump

Figure 10 shows the movement simulation of the hydraulic gearshift cylinder. In this simulation, the pressure-control valve (*VP1* in Figure 1) is controlled by a constant value while the flow-volume valve (*GSV2* in Figure 1) is controlled by a stimulation signal, as shown in Figure 10 (b). Figure 10 (a) shows change of oil pressures during this process, in which *P1* denotes the oil pressure from the hydraulic pump, *P2* the hydraulic pressure in the right cylinder chamber and *P3* the pressure in the left chamber. *P2* is controlled by *VP1* and the control current is constant; hence *P2* keeps a almost constant pressure value during this process. Figure 10 (b) shows the flow rate into the left cylinder chamber (denoted by *q*, see Figure 1) and the control signal for the flow-volume valve. The constant control signal of *GSV2* (from *0.5* to *1s*, from *1.5* to *2.1s*) leads to a constant flow rate during the movement of the gearshift cylinder. The displacement process of the cylinder from the middle to right end and reverse is displayed in Figure 10 (c).

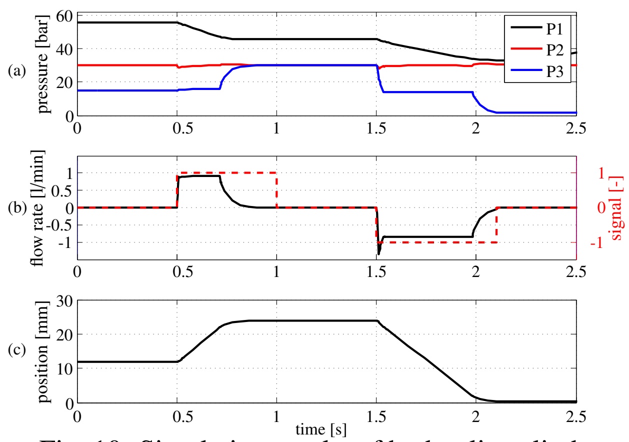

Fig. 10: Simulation results of hydraulic cylinder

3.2 Mechanical Model

This subsection describes the testing of the mechanical model and states that a correct synchronization process can be achieved. Therefore, the typical movement behavior (fast-slow-fast) and the results of the synchronization state determination are examined both.

In Figure 11 the upshifting simulation results are depicted, and its state shows that the model works as expected. Even the *speed difference increases* due to the missing connection between the toothing of the synchronizer hub and ring in stage 4 is also reproduced.

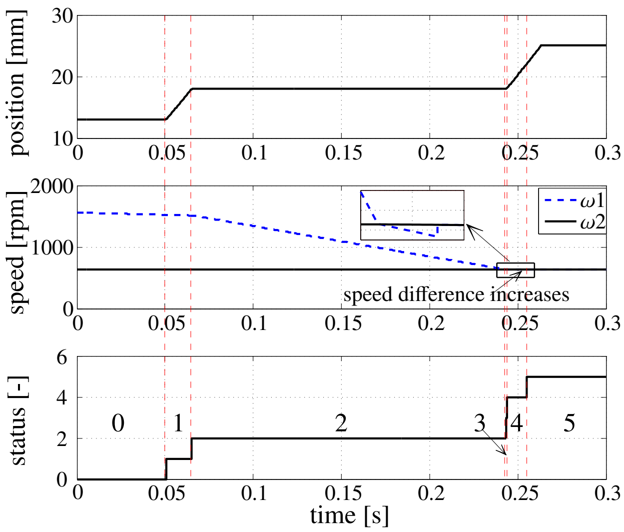

Fig. 11: Simulation results of synchronization

Self-return, an important characteristic of the detent pin (refer to Figure 4), is also tested, see Figure 12. The behavior when shifting force F_C vanishes behind the synchronization point (24mm, upshifting synchronization point is 18mm) is shown on the left-hand side, and the right-hand side shows the behavior of self-return in front of the synchronization point (15mm).

3.3 Hydro-Mechanical Model

Figure 13 shows different synchronizing processes under different working pressures. Synchronization time is reduced as expected when oil pressure increases.

3.4 Comparison with Measurements

Finally, the simulated synchronization process is compared with test bench measurement data from an automated manual transmission (AMT) system (compare [13]) having similar synchronization components. The AMT shifting valves are driven by constant currents

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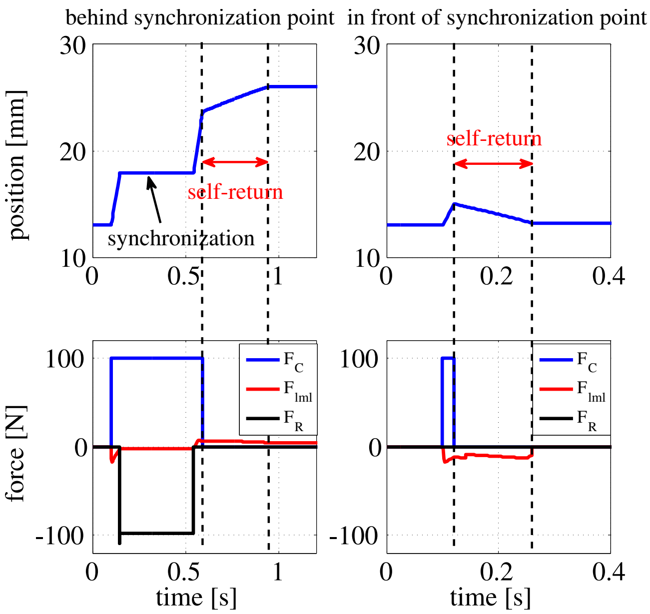

Fig. 12: Simulation results of self-return during up-shifting

and the DCT model shifting valves are driven by step signals. Figure 14 shows the comparison between the representative simulated and the measured shifting processes. The simulation result has a normalized root mean square error (NRMSE) of $e_{NRMS} = 1.5\%$. It can be stated that the presented model reproduces the characteristic details of the shifting process (pre-sync, locking, unlocking, turning hub and engagement).

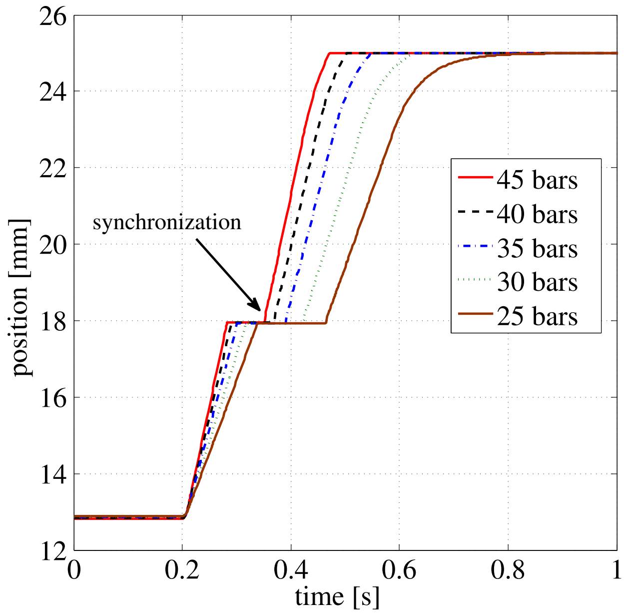

Fig. 13: Synchronization with different pressure

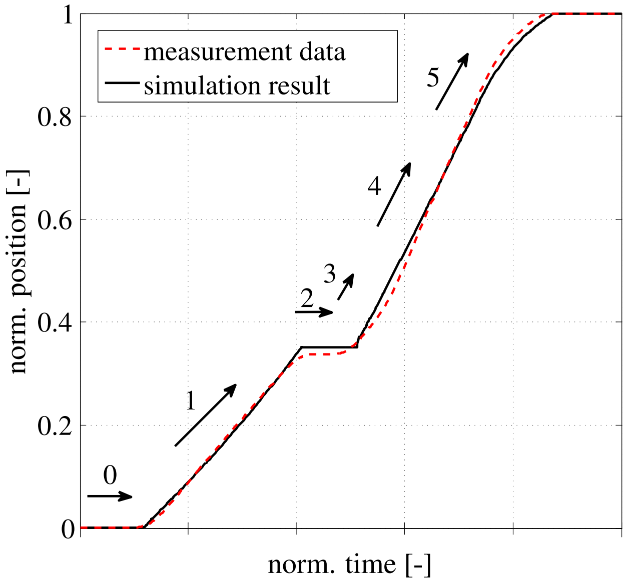

Fig. 14: Synchronization: Comparison of simulation results with measurements

4 Summary and Outlook

This paper gives a detailed introduction to the synchronization process and presents a dynamic Modelica® model for the hydro-mechanical actuation and synchronization system based on a popular DCT. This model has following features:

1) Gives a detailed representation of the synchronization process with 5 stages instead of simple 3 stages. Additionally in-depth reflection of the nonlinear dynamic system is also presented. This could provide a good reference for shifting quality optimization and more reliable standard for the model-based calibration.
2) Reveals the phenomenon that speed difference increases after the synchronization process because of power interruption in this stage. This is important to judge shift quality control strategies because during this phase serious problems as tooth breaking and shifting noise may occur.
3) Presents the user a fundamental understanding of the components composition principle and the system working function.
4) Shows that the tested hydraulic and mechanical modules have a good modularity for other similar system setups only through parameters changes.
5) Provides a good platform for the model-based calibration and function development.

Based on this dynamic simulation model, follow-up researches become possible: such as the integration of a clutch system (refer to [14]) and an appropriate control algorithms into a complete transmission model. The further important research field of model-based calibration on AMTs and DCTs in order to optimize shifting quality can also be identified.

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