

Comparison of Parameter-Identified Simulation Models with Different Detailing Level to Reproduce the Side Shaft Torque of an Automotive Powertrain with Automatic Transmission

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ABSTRACT

The underlying basic model represents a powertrain with automatic transmission including a torque converter. It is based on a greybox-modeling approach, which refers to ordinary differential equations with identified parameters and characteristic curves. The validated basic model is extended in order to reproduce the system behavior and especially the side shaft torque during a gear shift process. Therefore the model is extended by a transmission model with clutches for gear shifting in order to simulate specific powertrain dynamics additionally. The parameters have already been determined for the basic model using a method for isolated and structured parameter identification based on measurement data of an automotive powertrain test bench. A comparable structured parameter identification method is applied to obtain the parameters of the extended model.

The extended model will be compared with a state observer which is based on a low-order model and neglects the detailed consideration of submodels with nonlinear characteristics. The observer also reconstructs the side shaft torque during the entire maneuver and will be consulted as an alternative approach for estimating non-measurable states in the system. Comparing the extended model with the observer model will illustrate the limits and potentials of both approaches. The simulation results are compared and evaluated based on the measured side shaft torque.

Within this study the extended powertrain model is used as reference model that describes the vehicle dynamics for the considered maneuver precisely, whereas the observer is an alternative approach trying to reach similar results mainly obtained from the measured speed signals.

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INTRODUCTION

For simulating vehicle dynamics there are different possibilities to realize an accurate model approach. In this investigation two possibilities are used in order to simulate the states and side shaft torque of the powertrain, which are the nonlinear greybox model on the one hand and the linear observer model on the other hand. For basic comprehension [Figure 1](#) gives a structural overview of the presented modeling approaches.

The greybox model is a complete realization of all necessary elements for a successful simulation of the physical effects during a complex driving maneuver. The model contains mass inertias, elasticity and damping elements, driving resistance coefficients for modeling the powertrain loads on the drive side and diverse characteristic curves of the clutches and torque converter of the automatic transmission. Besides the listed elements for modeling it is also important which order or degrees of freedom the model has

respectively. For displaying the gear shift process it is necessary to implement two clutches which enable to simulate the torque transition from current gear to the upcoming gear. In fact the interaction between the disengaging clutch and the engaging clutch is deeply sensitive for an upshift process.

For the investigated driving maneuver both approaches (nonlinear greybox model vs. linear observer model) will be compared with each other. On the nonlinear greybox model side the reference model from [1] can simulate the entire maneuver except the shift process due to lack of modeling accuracy regarding shift related clutches. For this reason the reference or basic model is extended by the electrohydraulic model for the clutch operation and the model for both acting clutches. This extended model represents a full model that shows all necessary physical effects during this maneuver.

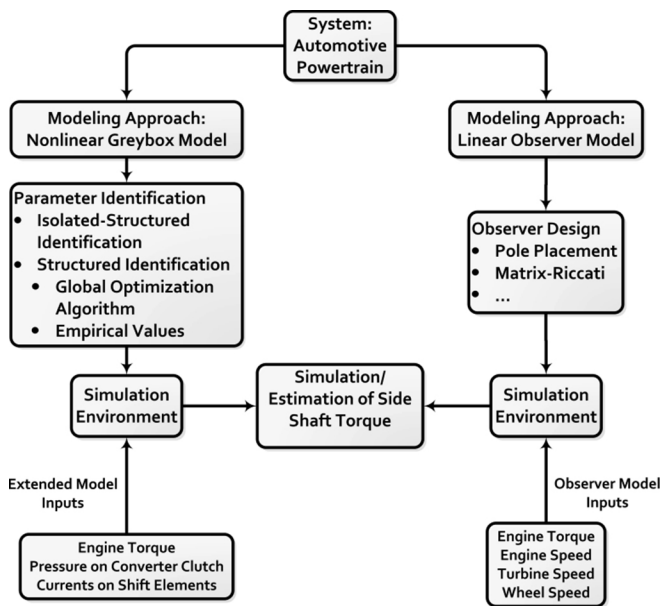


Figure 1. Overview of different modeling approaches

On the observer side, this investigation analyses how an observer model, which neglects the explicit modeling of characteristic curves, is able to estimate all main system states and the side shaft torque during the entire maneuver. The difference to the nonlinear powertrain model is founded by the simple model structure with less degrees of freedom and less model parameters. This structure can be applied by defining certain mass inertias which are only coupled with spring-damper elements, thus no characteristic curves are implemented.

This comparison shall demonstrate which benefits and limitations such approaches can deliver. Speaking from the observer's point of view a lower model order, which means a more simple structure regarding mathematical complexity leads to a minor modeling effort. Furthermore less parameters for the linear observer model indicates less determination of values and less dependence from system knowledge. But the downside aspect is the necessity of an appropriate observer design, where stability and certain robustness evidences must be done for a proper usability through the entire operating range of the investigated driving maneuvers. Certain physical effects in the system states are not computable without observer extensions. That is the reason why the model order needs to be increased. Particularly the existence of many nonlinearities of the torque converter (TC), the torque converter clutch (CC) and the clutches (C1, C2) involved during a shift operation require a minimal observer model order. Even the amount of obligatory speed signals depicts a disadvantage, but besides the main system input signal, those signals are the only information source to the observer. The three speed signals for the observer coming from engine, torque converter turbine and wheel contain the dynamics and characteristics of the powertrain during the maneuvers. These signals also illustrate the influence of the characteristic curves within the named powertrain components. So the observer's task is to extract that information and reconstruct the system states and furthermore the side shaft torque as good as possible without considering any nonlinear correlations.

The comparison will give insight about the difference between the approaches and what kind of benefit each implementation can provide. Both approaches have their own advantage and disadvantage, but the presented procedures show different possibilities of modeling and estimating powertrain dynamics in its own way. Also the application can differ from each other but the intention of system modeling and simulation remains the same.

MODELING

In this publication the investigated system represents an automotive powertrain with automatic transmission and standard drive on a test bench. Both rear driven side shafts are connected with electric machines which provide driving and dynamic resistances depending on the equivalent vehicle speed and vehicle mass. Amongst others the powertrain is additionally equipped with a pressure device to measure the pressure of the electrohydraulic system inside the gear box and a torque sensor to measure the side shafts torque.

Basic Model

The basic model represents an automotive powertrain which is presented in [1]. This model contains the submodels combustion engine, torque converter, gearbox (automatic transmission with a fix gear ratio), combined side shaft and vehicle (equivalent vehicle body mass inertia). The submodels are built up on physically based differential equations. The overall basic model has three rotational degrees of freedom which are $\varphi_{E/P}$ (engine/pump), φ_T (turbine) and φ_{Veh} (vehicle). A schematic illustration of the model is depicted in Figure 2. The dashed line in Figure 2 shows the detailed gear box and output shaft (further degree of freedom φ_{OS} submodels of the extended basic model, which will be explained in the course of this chapter. The basic model has a simplified gear box with a constant gear ratio i_{GB} without a separate mass inertia of the output shaft.

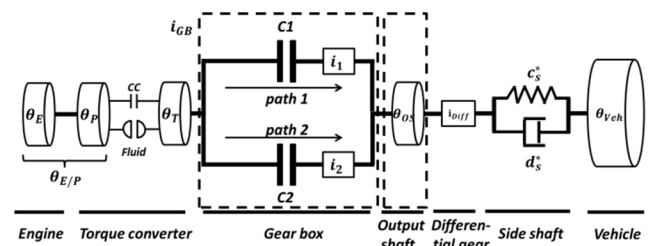


Figure 2. Basic and extended model of an automotive powertrain with automatic transmission

The measured signal of the engine torque T_E is used as input signal for the basic model. For the simplification of the model complexity, also the measured cylinder pressure $p_{CC,Cy}$ is used as model input instead of the current acting on the electro hydraulic system of the converter clutch. The effective pressure p_{CC} acting on the converter clutch disks is calculated by subtracting a constant empirically estimated offset $p_{CC,offset}$ from the cylinder pressure. The engine torque, the torque of the pump T_p and the torque of the converter clutch T_{CC} act on the rotating mass inertia of the engine and pump

unit $\theta_{E/P}$. It should be noted that T_E represents the net torque value and therefore no losses were modeled explicitly. The differential equation of the engine submodel is

$$\theta_{E/P} \cdot \ddot{\phi}_{E/P} = T_E - T_P - T_{CC}. \quad (1)$$

The pump torque is determined by

$$T_P = \lambda(v) \cdot \rho_{TC} \cdot D_{TC}^5 \cdot \dot{\phi}_{E/P}^2. \quad (2)$$

It depends on the power coefficient $\lambda(v)$, the density of the operating fluid ρ_{TC} and the profile diameter of the pump impeller D_{TC} which represent specific parameters of the torque converter. The quantity $v = \frac{\dot{\phi}_T}{\dot{\phi}_P}$ represents the ratio between turbine speed $\dot{\phi}_T$ and pump speed $\dot{\phi}_P$ or $\dot{\phi}_{E/P}$ respectively. Furthermore the converter clutch is generally never completely closed and operates continuously in slip or micro slip state. Therefore T_{CC} is calculated by

$$T_{CC} = A_{CC} \cdot r_{m,CC} \cdot N_{CC} \cdot \mu_{CC}(\Delta\dot{\phi}_{P/T}) \cdot p_{CC}, \quad (3)$$

where A_{CC} and $r_{m,CC}$ represent the surface and the mean radius of the converter clutch disks respectively. N_{CC} is the number of the rubbing disk surfaces. The quantity μ_{CC} is the friction coefficient which is a function of speed difference at the converter clutch $\Delta\dot{\phi}_{P/T} = \dot{\phi}_P - \dot{\phi}_T$.

The turbine torque is calculated using the pump torque and the torque factor $\mu_{TC}(v)$

$$T_T = T_P \cdot \mu_{TC}(v). \quad (4)$$

The nonlinear quantities $\lambda(v)$, $\mu_{CC}(\Delta\dot{\phi}_{P/T})$ and $\mu_{TC}(v)$ are represented by characteristic curves.

Both driven side shafts are combined to one side shaft. The combined side shaft torque is calculated as

$$T_{SS} = c_{SS}^* \cdot \left(\frac{\phi_{OS}}{i_{Diff}} - \phi_{veh} \right) + d_{SS}^* \cdot \left(\frac{\dot{\phi}_{OS}}{i_{Diff}} - \dot{\phi}_{veh} \right), \quad (5)$$

where i_{Diff} is the ratio of the differential gear and c_{SS}^* and d_{SS}^* represent the stiffness and damping coefficients of the merged side shaft. Moreover, the entire vehicle mass m_{veh} is reduced to a rotational vehicle mass inertia θ_{veh} and concentrated at the driven wheel

$$\theta_{veh} = m_{veh} \cdot r_{dyn}^2, \quad (6)$$

where r_{dyn} is the dynamic radius of the tire. There is no tire model implemented on the test bench. Thus, effects of the tire were neglected and the torque of the combined side shaft and the drive resistances T_{Res} act directly on the vehicles mass inertia

$$\theta_{veh} \cdot \ddot{\phi}_{veh} = T_{SS} - T_{Res}, \quad (7)$$

with

$$T_{Res} = w_0 + w_1 \cdot \dot{\phi}_{veh} + w_2 \cdot \dot{\phi}_{veh}^2. \quad (8)$$

The coefficients w describe the static (w_0), roll (w_1) and air drag (w_2) driving resistances. The resistance coefficients, the vehicle mass and the dynamic radius of tire are adjustable parameters on the test bench for different investigations.

Model Extension

In this investigation, the basic model has been extended into a model with four degrees of freedom in order to simulate a gear shift operation additionally. The extension comprises a comparatively detailed gear box submodel containing two clutches which are actuated during a gear shift process. The extension allows to simulate the torque transfer from the disengaging clutch C1 (path 1) to the engaging clutch C2 (path 2) via the clutch pressure p_c and thus to consider characteristic effects of a gear shift process. Both clutches can pass into the stick and the slip state. A logic module decides on the current state of each clutch. The condition for the stick state is fulfilled, if the maximum transmittable torque of the sticking clutch T_{max} is equal or higher than the amount of the current transmitted torque T_c

$$|T_c| \leq T_{max} = A_c \cdot r_{m,c} \cdot N_c \cdot \mu_0 \cdot p_c, \quad (9)$$

where μ_0 represents the static friction coefficient, A_c and $r_{m,c}$ are the surface and the mean radius of the clutch disks respectively and N_c is the number of the rubbing disk surfaces. The transmitted torque for the stick state of the disengaging clutch $T_{C1.stick}$ and the engaging clutch $T_{C2.stick}$ is calculated by

$$T_{C1.stick} = \frac{\theta_{OS}^* \cdot ((T_T + T_{CC}) - T_{C2}) + \theta_T^* \cdot \left(\frac{T_{SS}}{i_{Diff}} \cdot \frac{1}{i_2} - T_{C2} \cdot \frac{i_2}{i_1} \right)}{\theta_{OS}^* + \theta_T^*}, \quad (10)$$

where θ_{OS}^* and θ_T^* represent the reduced mass inertias on path 1 and

$$T_{C2.stick} = \frac{\theta_{OS}^* \cdot ((T_T + T_{CC}) - T_{C1}) + \theta_T^* \cdot \left(\frac{T_{SS}}{i_{Diff}} \cdot \frac{1}{i_2} - T_{C1} \cdot \frac{i_1}{i_2} \right)}{\theta_{OS}^* + \theta_T^*}, \quad (11)$$

where θ_{OS}^{**} and θ_T^{**} are the reduced mass inertias on path 2. The quantity θ_{OS} represents the mass inertia of the output shaft. The values i_1 and i_2 represent the gear ratios of the current and the upcoming gear. If $|T_c| > T_{max}$, the clutch begins to slip, then the torque of each slipping clutch is calculated according to Eq. (3).

To simulate the pressure acting on the clutch, it is necessary to model the components which influence the progress of the clutch pressure. The model which has been implemented represents a simplified electrohydraulic system and the piston, which actuates the disks of the clutch. The model is based on the approach “incompressible cylinder model” [2]. A schematic view of this subsystem is showed in Figure 3.

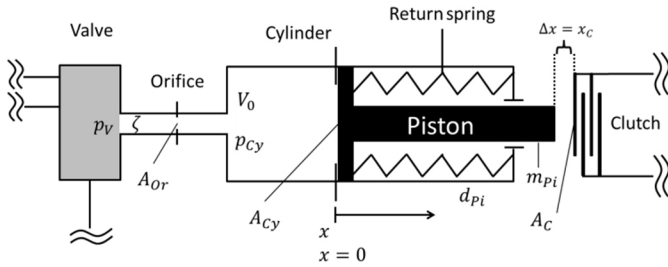


Figure 3. Schematic view of the simplified electrohydraulic system with piston for clutch actuation

The transmission control unit (TCU) provides a current I in dependence on the operating point and the driving maneuver respectively, which is an additional input for the extended model for each clutch. A valve inside the electrohydraulic system sets the valve pressure p_v depending on the current. The correlation between the current and the valve pressure is described by a characteristic curve. The dynamics of the valve are considered using a second order transfer function [3]

$$G(s) = \frac{1}{1 + g_1 \cdot s + g_2 \cdot s^2} \quad (12)$$

with the coefficients g_1 and g_2 . The pressure acting on the clutch p_c depends on the piston stroke x , which can be calculated by

$$x(t) = x(t_0) + \frac{1}{A_{cy}} \int_0^t \dot{V}(t) \cdot dt, \quad (13)$$

where $x(t_0)$ is the initial stroke of the piston, A_{cy} is the cross-section surface of the cylinder and $\dot{V}$ is the volumetric flow through the orifice. The volumetric flow is primarily influenced by the pressure difference between the valve and the cylinder $\Delta p = p_v - p_{cy}$. It is calculated by

$$\dot{V} = A_{or} \cdot \sqrt{\left| \frac{2}{\rho_o \zeta} \cdot \Delta p \right|} \cdot \text{sign}(\Delta p), \quad (14)$$

where A_{or} is the cross-section surface of the orifice, ρ_o is the density of the fluid and ζ is the resistance coefficient of the hydraulic path.

The calculation of the cylinder pressure is carried out by solving the differential equation of the piston

$$m_{pi} \cdot \ddot{x} = F_{cy} - F_D - F_S(x) = p_{cy} \cdot A_{cy} - \dot{x} \cdot d_{pi} - F_S(x). \quad (15)$$

The quantity m_{pi} is the piston mass, d_{pi} represents the damping coefficient of the piston and $F_S(x)$ is the spring force, which comprises the forces of the return spring F_R and the clutch disks F_C which also contains the force of a limit stop spring. The spring force depends on the piston stroke and thus it has nonlinear characteristics

$$F_S(x) = \begin{cases} F_R(x) & 0 \leq x < x_c \\ F_R(x) + F_C(x) & x_c \leq x \end{cases}. \quad (16)$$

During the gear shift process, the piston passes through two sections (see Eq. (16)). In the first section the return spring forces against the piston but there is still no pressure acting on the clutch. Finally in the second section, the piston generates a pressure which acts on the clutch disks. The force or the stiffness of the clutch disks respectively is huge so the raise of the piston stroke is comparatively small in this section. The additional clutch pressure caused by centrifugal force of the rotating clutch has been neglected.

Parameter Identification

The parameters of the basic model have already been identified using a method for isolated and structured parameter identification based on individual driving maneuvers [1]. In this publication, the objective is to estimate the parameters of the output shaft, electrohydraulic and clutch model. The challenge of estimation of electrohydraulic and clutch model parameters is in fact, that the electrohydraulic system and the clutches operate simultaneously during a gear shift process and there is no possibility to control the systems independently in a series automatic transmission. Furthermore, there are only few measured signals available, which can be used to describe the system states. This restricts an isolated parameter identification approach according to [1]. Hence, a comparable method for parameter identification will be introduced with the objective to estimate the parameters of the electrohydraulic and clutch model structurally. The method is based on a three-stage identification process using measured data, a global optimization algorithm [4] and empirical values.

In Table 1 all parameters or characteristic curves of the extended model and their used identification method are listed.