

Improved Comfort Analysis and Drivability Assessment by the Use of an Extended Power Train Model for Automatic Transmissions

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ABSTRACT

The new generation of automatic transmissions is characterized by a compact and highly efficient design. By the use of a higher overall gear ratio and lightweight components combined with optimal gear set concepts it is possible to improve significantly fuel consumption and driving dynamics. Precise and efficient real time models of the whole power train including models for complex subsystems like the automatic transmission are needed to combine real hardware with virtual models on XiL test rigs. Thereby it's possible to achieve a more efficient product development process optimized towards low development costs by less needed prototypes and shorter development times by pushing front loading in the process.

In this paper a new real time model for automatic transmissions including approved models for the torque converter, the lock-up clutch and the torsional damper are introduced. At the current development stage the model can be used for comfort analysis and drivability assessment. Thereby various operating conditions including motored engine and an open, slipping or locked clutch condition can be simulated. For the torque converter a map based, an equivalent mechanical system model derived from the bond graph modeling method or a detailed physical model based on the energy and momentum equations can be used. The mechanical dampers can be varied in their positions and allow to model common used damper configurations including the conventional, TTD (Turbine Torsional Damper) and DTD (Double Torsional Damper) system arrangement.

Different simulation examples for various operation conditions and modeling approaches are analyzed to underline the broad range of application for comfort analysis and drivability assessment. Further an outlook is given how to include losses into the model to improve the efficiency of future power trains based on measurements and empirical equations.

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INTRODUCTION

In Europe automatic transmissions for passenger cars with hydraulic torque converter (TC) are mainly used in upper class vehicles demanding the highest driving comfort. The drive train of such cars is characterized by high engine

torques and high power output combined with excellent driving behavior. In 2009 in Western Europe about 16 percent of all cars were equipped with an automatic transmission [2]. For the near future (2015) the amount of automatic transmissions will slightly increase up to about 17% (see [figure 1](#)).

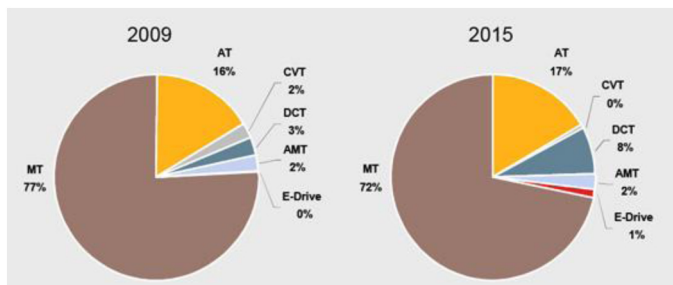


Figure 1. Market situation in Western Europe for passenger car transmissions [2]

Especially in the last years dual clutch transmissions became of higher interest, combining the positive effects of improved efficiency and good dynamic behavior, compared to common used 6 speed automatic transmissions [2]. The first dual clutch transmission arrived by Porsche in-house development for racing cars already in the 1980s and improvements in control strategies and actuator design allow today their broad use in passenger cars.

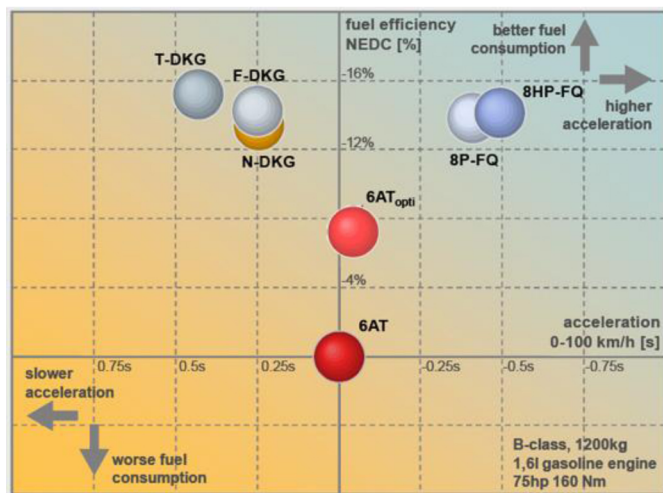


Figure 2. Comparison of fuel efficiency and acceleration behavior (NEDC) for different transmission concepts [1]

A new generation of automatic transmissions with torque converter was recently introduced by BMW/ZF [5,9] and is characterized by an improved efficiency, compact design with 8 gear stages and excellent driving dynamics.

The development of such improved future products can be realized by an optimized development process based on virtual product development strategies introduced already at early development stages. At later development stages also the optimal combination of simulation models with real hardware, known as XiL methods, help to improve the calibration and integration of the new product into the whole drive train and full vehicle concept. Virtual methods in the product development process help to efficiently handle opposed requirements such as high comfort, light weight

design, high efficiency and low costs leading into an increased system complexity.

MOTIVATION

To simulate the low frequency dynamic behavior of drive trains with focus on automatic transmissions especially the torque converter in combination with the lockup clutch and the mechanical damper system is of high interest. Simulation models which can precisely predict the dynamic behavior up to about 25 Hz can be used for:

- Design of the over-lock-clutch and the mechanical damper system
- Pre-calibration of TCU by model in the loop strategies (control strategies for over-lock-clutch)
- Calibration of ECU and TCU by Hardware in the Loop (HiL) strategies on special test rigs
- Drivability assessment

So far different approaches to simulate the torque converter, the over-lock-clutch and the mechanical damper system are available. But so far a model including all subsystems working under various operational conditions and with high flexibility is not quoted in literature. To holistically optimize automatic transmission simulation models should allow simulating motored condition, operation under open, closed and slipping clutch and different damper configurations.

A NEW REAL TIME SIMULATION MODEL FOR AUTOMATIC TRANSMISSIONS

This new model consists of different submodels for the torque converter, the over-lock-clutch, the mechanical damper and the planetary gear sets, clutches and brakes. The modeling concept of each component is described subsequently:

HYDRAULIC TORQUE CONVERTER MODELING

In literature different modeling approaches for the torque converter, suitable for real time simulation and drivability assessment, are described. The different approaches can be integrated in Matlab/Simulink based on a signal flow approach (figure 3).

Nonlinear Differential Equations

Under the use of the unsteady equations of momentum (equation 1) and the energy conservation (equation 2) a nonlinear differential system can be derived to describe the dynamic behavior of the torque converter. The equations are derived by taking the momentum of flow along the middle flow line through the defined boundaries and by calculating

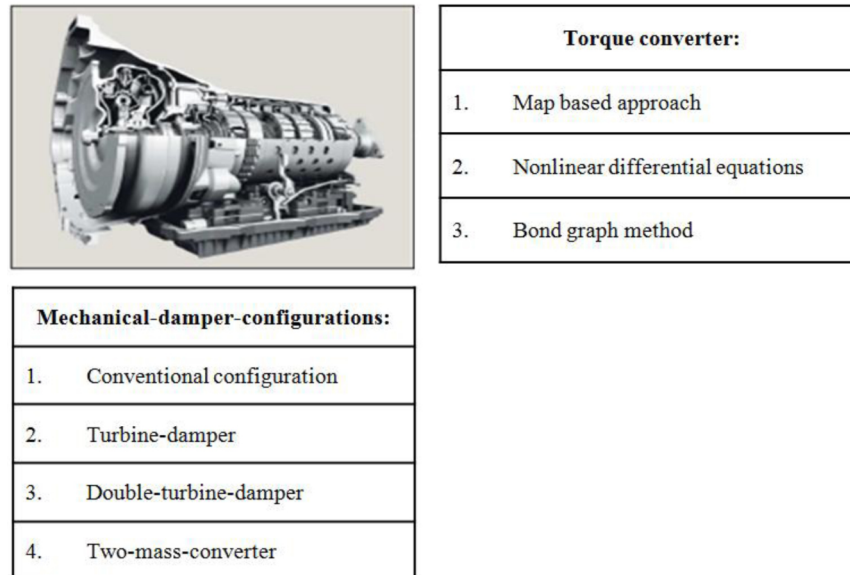


Figure 3. Overview of the analyzed torque converter modeling approaches and mechanical-damper-configurations

the energy of an infinitesimal fluid element along the middle flow line.

$$M_p, M_T, M_R = f(a_i, \omega_p, \dot{\omega}_p, \omega_T, \dot{\omega}_T, \dot{V}, \ddot{V}, \dot{V}^2) \quad (1)$$

$$0 = f(a_i, \omega_p, \dot{\omega}_p, \omega_T, \dot{\omega}_T, \dot{V}, \ddot{V}, \dot{V}^2) \quad (2)$$

M_p, M_T, M_R are the reaction torques of the pump-, turbine- and the idler wheel. $\omega_p, \omega_T, \dot{V}$ are the speeds of the pump-, turbine wheel and the volume flow through the torque converter. The coefficients a_i are functions of the flow parameters including different flow angles and cross sectional areas at the in- and outflow between the wheels. Further they include the fluid density and integrated flow parameters along specific segments of the middle flow line between the pump, turbine and guide wheel. To derive all the parameters for a specific torque converter very detailed data on geometry and flow properties are needed. For parametric studies and pre design investigations such parameters are usually not available, so that the approach based on the momentum and energy of flow based equations is not applicable. Besides the model is limited to speed ratios of the turbine to pump smaller than one and it is not possible to simulate reverse flow, which is required under motored condition.

Bond Graph Method

To derive a substituted mechanical system the Bond graph method uses the already quoted equations of momentum (equation 1) and energy conservation (equation 2). The bond graph method is a method to abstractly represent a technical system. The system is not fixed to a certain domain and can consist of electrical, mechanical etc. components. The method is domain independent and uses the energy

conservation and energy flow through the system [4]. The substituted mechanical system for a torque converter is shown in figure 4 and consists of the inertias of the turbine and pump, two continuously variable ratios, an ideal slipping clutch (no loss through friction torque) and a linear torsional spring. The speed ratios (SR) on pump (index i) and turbine side (index t) depend on the speeds of turbine and pump, the volume flow, the cross sectional area of the flow and blade angles. The speed difference ω_c calculated out of the pressure loss term in the energy equation represents losses caused by impacts and friction and depends on the speeds and volume flow. The spring stiffness k represents the fluid inertia term.

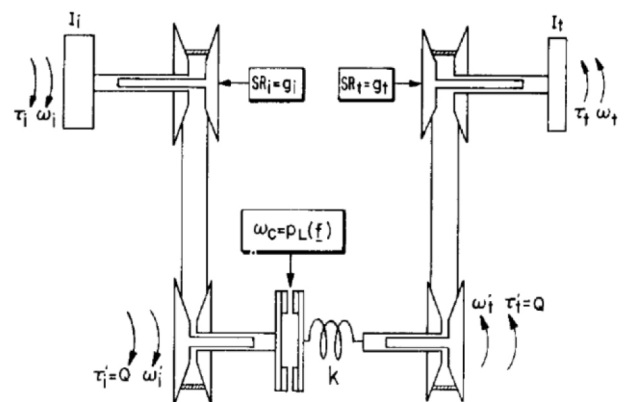


Figure 4. The substituted mechanical system for a torque converter [3]

The modeling approach with the bond graph method has the same disadvantages as the one based on the momentum and energy equations. To calculate the speed ratios and the speed difference for the clutch, again detailed converter data and flow properties are needed.

Map Based Approach

With a torque converter map including the characteristic curves the torques of turbine and pump based on available speeds can be calculated. The approach is based on the Euler turbine equation for single torques saying that the reaction torque of the flow is equivalent to the volume flow multiplied by the density of the fluid and the swirl difference between income and outcome. The swirl difference depends on the difference between radius and the speed in circumferential direction between income and outcome [6].

$$M = Q \cdot \rho \cdot \Delta(r \cdot c_u) \quad (3)$$

The torque ratio between turbine and pump μ is a function of the pump and turbine speeds.

$$\mu = \frac{M_T}{M_P} = f(v) \quad (4)$$

$$v = \frac{\omega_T}{\omega_P} \quad (5)$$

Based on the law of similarity [4] the torque for the pump can be calculated as

$$M_P = \lambda(v) \cdot \rho \cdot \omega_P^2 \cdot D_{Wandler}^2 \quad (6)$$

The coefficient of performance $\lambda(v)$ and the torque ratio $\mu(v)$ is measured under steady state operation. The resulting curves are shown in the converter map (figure 5). Today the classical torque converter is combined with a hydrodynamic clutch (Trilok torque converter) to combine the advantages of the converter and clutch. This is realized over a guide wheel which is locked in one rotation direction and free in the other. At a certain speeds ratio (clutch point) the flow changes so that the fluid flows on the backside of the guide wheel resulting in a free rotation of the wheel. Now the torque converter is operated in clutch mode.

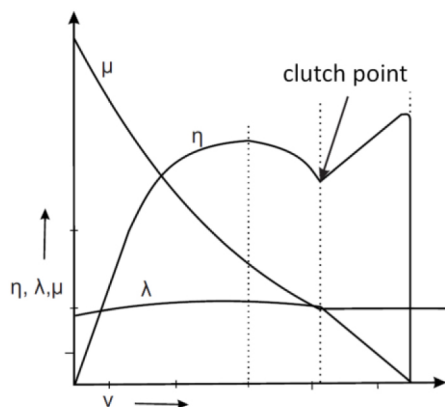


Figure 5. Characteristic map of a trilok-torque-converter showing the coefficient of performance, the torque ratio and the efficiency factor over the speed ratio [4]

The Matlab/Simulink model of the different torque converter and over-lock-clutch configurations consists of four modules representing the

1. Locked clutch under fired condition
2. Open or slipping clutch and torque converter under fired condition
3. Locked clutch under motored condition
4. Open or slipping clutch and torque converter under motored condition

The decision which model path is chosen is done based on the turbine and pump speed and the input torques on pump (engine torque) and turbine side (input torque from drive train/transmission). To avoid permanent and fast shifting between the modules the shift points have a small overlap. Under motored condition the inverse of v is taken to avoid an unstable model behavior caused by division through zero. The converter map for motored condition shows a nearly linear decrease of μ .

MECHANICAL DAMPER

The mechanical damper is modeled as a simple linear spring damper system with damping parameter d and stiffness parameter c based on following equation

$$T_{\text{Spring+Damper}} = c \cdot \Delta\phi + d \cdot \Delta\dot{\phi} \quad (7)$$

Where $T_{\text{Spring+Damper}}$ represents the reaction torque, “delta phi” and “delta phi-dot” the relative angle and angular velocity of the spring damper element. For the position of the damper different configurations are possible depending on the layout of the whole power train and its characteristic eigenmodes (mainly the second and third eigenmode):

In the Matlab/Simulink model the different damper configurations (figure 6) are realized in different sub models which can be included into the whole transmission model by drag and drop.

OVERLOCK CLUTCH

The over-lock-clutch is actuated by a normal F_N force that is either controlled over the shift table or over a PID algorithm in case of a slipping clutch. The clutch limit torque is calculated by

$$M_{\text{Kupplung,max}} = \frac{2}{3} \cdot \frac{R_a^3 - R_i^3}{R_a^2 - R_i^2} \mu \cdot F_N \quad (7)$$

The torque on the clutch output side is defined by the maximum between the engine input torque and the highest possible clutch torque. The decision between the slip and closed condition can be easily found by analyzing the delta speed between disk 1 and 2 of the clutch. Additionally the maximum clutch torque has to be compared with the reaction