

Control and Integration Challenges for Future Automatic Transmissions

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ABSTRACT

The ever-increasing regulatory requirement on CO₂ emissions drives efficiency improvement of vehicle powertrain systems. In this context, three mega trends have been happening in the automotive transmission industry. First, future automatic transmissions will have more gear steps to offer a broader ratio spread and finer ratio steps, which may enable the engine to operate at its efficient regions more often. Second, engine downsizing with boosted power and flexible cylinder deactivation have become the technology trend to achieve better thermal efficiency. These engine technologies demand improved transmission dampers with greater isolation capabilities to drive future transmission dampers to be equipped with softer springs. Third, future transmissions will be more efficient due to new architectures and incremental subsystem improvements. We have discovered that all three trends can impose significant challenges to make the future transmission controls and powertrain integration more difficult. This paper starts with a fundamental study on the automotive drivetrain system. The powerful analytical modeling and sensitivity analysis method developed allows us to gain deep understandings of the drivetrain system dynamic nature. The results are general and can help answer some long-standing questions on transmission controls, such as, what is the physics behind the shift vibrations? why are some shifts intrinsically more difficult to control than others? Through the theoretical analysis, we can identify the latent shift vibration tendency and the dynamic structures behind. As the transmission system continues to evolve, it becomes clear that controls and integration will be more challenging for future transmissions. Simple adaption of the control systems from the existing transmissions today may no longer be adequate to the transmission systems tomorrow.

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INTRODUCTION

The ever-increasing regulatory requirement on CO₂ emissions drives efficiency improvement of the vehicle powertrain system. Although the powertrain electrification and alternative fuel technologies will continue to advance and penetrate with a growing market share, conventional powertrain will still dominate the global powertrain marketplace in the foreseeable future. The stringent 2025 fuel economy targets will present tremendous challenges to the conventional powertrain system on efficiency improvement, and technology advancement is necessary across the board from the engine, transmission, system integration, and controls, etc. Future engines will have smaller displacements with power boost technologies, and advanced technologies will also enable flexible cylinder deactivation to truly optimize the engine operation efficiency by minimizing the number of cylinders according to the instant power demand. On the other hand, transmission systems will continue to evolve, with a balanced distribution of various transmission types in different market regions. Among them, automatic transmission (AT) will continue to dominate the North American market and account for the bulk of the global market. With all the technology advancements, there are three notable trends for the future ATs. First of all, future ATs will have more gear steps to offer a broader ratio spread and finer ratio steps, which can better enable the engine to operate at its most efficient regions. Secondly, future engines (with further downsizing

and cylinder deactivation) demand advanced transmission dampers with improved torsional isolation capability. In the same time, the transmission torque converter clutch (TCC) will be operated in the lockup state more often for better fuel economy. Finally, future transmissions will have higher system efficiency due to new architectures used as well as component improvement.

Thanks to the desired launch performance, smooth shift quality, wide ratio spread, and relatively low cost with ever-improving efficiency, AT will remain popular and continue to dominate in the global marketplace in the foreseeable future. The speeds race for AT has accelerated in the recent past, and the market moved from 3/4-speeds to 5/6-speed transmissions not long ago, and will continue to grow into 8/9/10-speed transmission systems in the near future.

Transmissions with more gear steps can enable better fuel economy and vehicle performance. However, the trade-off comes with a higher system complexity and more components including gears, clutches and control devices, etc. The system overall mass and rotational inertias are likely to grow. Complexity for transmission controls will also increase as the shift scenarios grows exponentially with the gear steps. In practice, a future transmission will be shifted more frequently as the lines in a shift schedule become denser. However, maintaining the same or higher shift quality is critical because customers will associate the higher speed transmission with advanced technology and expect premium drive quality. Unfortunately, the

growth in the number of gear steps, improved transmission dampers as well as the ever increasing system efficiency can create additional challenges for the transmission shift controls. The control system know-hows built upon the transmissions of the past may no longer be sufficient to address all the control needs we will face in the future transmission systems.

The goal of the shift control is to deliver fast and smooth shift whenever the ratio change is needed. Clutch coordination and speed change are controlled closely to maintain the torque disturbance during the shift small and acceptable to the costumers. The inertia effect on the shift control has been well studied and documented [1]-[2]. However, the drivetrain structures, dynamic characteristics and their influences on the shift controls have been less studied, and there is little theoretical results in the literature on this subject. Through the years, rules of thumb have been generated on certain driveline structures and the control approaches that are associated with high risks of potential shift vibrations. However, questions like what is the physics behind a shift vibration issue and why some shifts are inherently more difficult to control than others remain unclear to a large degree. To answer the fundamental questions, we need a deeper understanding of the drivetrain system dynamic nature during gear shifting, which later leads to the findings on the control and integration challenges for the future transmission system.

In this paper, we approach the shift control problems from the drivetrain dynamic characteristics and NVH perspective. The novel analytical modeling and sensitivity analysis method we developed [8] are used to study the drivetrain system during gear shifting. The tool is powerful and the results can clearly reveal the key components contributing to a specific vibration phenomenon among the highly coupled drivetrain structures. The insights can greatly simplify the related NVH analysis and troubleshooting process, which can be very difficult otherwise through inefficient simulation study. The core drivetrain structure identified through this analysis reveals the potential interactions among the key drivetrain components that can be the root cause to the vibration during gear shifting. It will become clear that how the key parameters can influence the shift vibration sensitivity, and why the three system trends in today's transmission systems can make the controls and integration more difficult in the future. The method used and the findings presented in this study are useful for the NVH related analysis, troubleshooting, damper design, model reduction as well as hardware and controls design.

In what follows, we will start with the novel modelling and analysis method. In the subsequent section, the analytical driveline models are created for different transmission systems. Through the sensitivity analysis, two typical drivetrain structures are identified that can lead to possible shift vibration tendencies. It is followed by the discussions on the system dynamic structure and the parameter influences in the context of future transmissions. Summary and concluding remarks will be drawn in the end.

TRANSMISSION & DRIVELINE SYSTEM EQUATIONS

For reader's convenience, we include a brief introduction of the analytical modeling and sensitivity analysis method developed previously. Interested readers may refer to [8] for more details.

Transmission & Driveline Model Structure

An automotive driveline is usually modelled as a series of inertia-spring systems with lumped parameters. Figure 1 shows an example of a typical driveline model, where the entire transmission is simplified as a lumped inertia. Although it is often used to study the driveline dynamics, when it comes to transmission related NVH analysis, such model is over simplified, where possible dynamics driven by the transmission internal components are neglected.

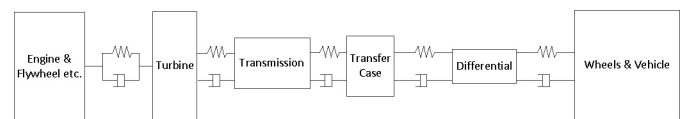


Figure 1. Lumped Parameter Driveline Model Structure

To study transmission behavior and control characterization, an inertia-based transmission model can also be used [1]-[2]. Using the basic free body diagram of an automatic transmission presented in [2] shown in Figure 2, the transmission can be broken into two lumped inertias before and after the oncoming clutch. The equivalent inertias can be calculated with respect to the input acceleration. However, when it comes to the detailed dynamics and the NVH nature involving the transmission, this simplified inertia-based model is no longer sufficient because the dynamic components like compliance and damping are not considered.

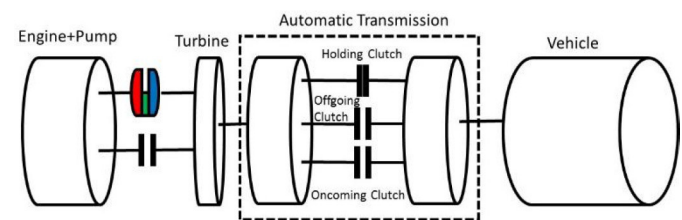


Figure 2. Free Body Diagram of an Automatic Transmission [2]

To understand the transmission and driveline dynamics during a shift from the NVH perspective, a more generic model framework for the transmission is needed. In fact, the model structures shown in Figure 1 and 2 can be combined and extended to include all the transmission internal inertias, stiffness, and damping elements. In this way, a new and generic transmission and drivetrain model can be created so that the dynamics involving transmission internal parts and interactions with other drivetrain parts can be studied, which often can reveal insights on drivetrain dynamic nature during gear shifting. Figure 3 illustrates a generic transmission and drivetrain model framework.

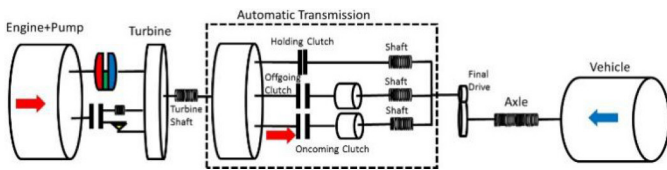


Figure 3. Detailed Transmission Model Structure in a Driveline Environment

State Space Model Equation

Based on the model structure in Figure 3, a system equation can be formulated using the standard procedure from Lagrangian Mechanics [3]. First, the desired nodes need to be selected, and based on which the state variables can be defined. The nodes can usually be selected according to the rotational speeds, and the components that rotate at the same speed may be lumped together as one node. With all the nodes defined, the state variable φ can be denoted by an $n \times 1$ vector, where each entry φ_i represents the rotational angle of the i^{th} node. The speed and acceleration vector are written as $\dot{\varphi}$ and $\ddot{\varphi}$ respectively. The system equations can be formulated based on Newton's second law of motion with the constraints on the motion considered relevant to each node. Such system constraints are written in the following form.

$$\Phi \varphi = 0 \text{ (or } \Phi \ddot{\varphi} = 0) \quad (1)$$

Here, Φ is an $m \times n$ matrix with m representing the total number of the constraints considered in the system. Constraint matrix Φ can also reveal the interaction forces among the coupling nodes. Such forces on each of the nodes can be calculated as $\Phi^T \Lambda$, where the $m \times 1$ vector Λ is referred to as the Lagrangian and needs to be resolved [3].

Using the second law of motion, the system dynamic equations can be formulated in a matrix form as

$$\bar{I} \ddot{\varphi} + C \dot{\varphi} + K \varphi = \Sigma T + T \quad (2)$$

In eq. (2), φ is an $n \times 1$ vector representing the angular position of the nodes; n is the number of nodes considered; $\bar{I}$ is an $n \times n$ diagonal matrix, where each entry on the diagonal represents the rotational moment of inertia associated with the corresponding node; C is an $n \times n$ matrix with the diagonal entries presenting the associated damping elements; K is the $n \times n$ matrix with the diagonal entries presenting the effective stiffness; ΣT is an $n \times 1$ vector with each entry representing the sum of the reaction forces acting on the corresponding node; T is an $n \times 1$ vector to denote the external forces acting on relevant nodes. The external forces T may be considered as the system input and usually come from the engine, clutches and brakes etc. Note that through Lagrangian Mechanics, the internal forces ΣT can be written as

$$\Sigma T = -\Phi^T \Lambda$$

Here, the minus sign is chosen arbitrarily for simplicity.

Combine eq. (1) and (2), one can obtain the system equation.

$$Mx + \bar{C} \dot{\varphi} + \bar{K} \varphi = \bar{T} \quad (3)$$

The terms in eq. (3) are defined as

$$M = \begin{bmatrix} \bar{I} & \Phi^T \\ \Phi & 0 \end{bmatrix}, \bar{C} = \begin{bmatrix} C & 0 \\ 0 & 0 \end{bmatrix}, \bar{K} = \begin{bmatrix} K & 0 \\ 0 & 0 \end{bmatrix}, \bar{T} = \begin{bmatrix} T \\ 0 \end{bmatrix},$$

$$x = \begin{bmatrix} \ddot{\varphi} \\ \Lambda \end{bmatrix} \text{ and } \bar{\varphi} = \begin{bmatrix} \varphi \\ 0 \end{bmatrix}. \quad (4)$$

where $\bar{I}$, C , K and T are defined in eq. (2); $\bar{\varphi}$ and $\bar{T}$ are defined in such a way that the dimensions of the relevant terms remain consistent; x is an $(m+n) \times 1$ vector where the bottom m entries, represented by vector Λ , are the Lagrangians associated with the constraints Φ .

The system equation in (3) represents a general form of the equation established for a transmission and drivetrain system with dynamic elements (e.g., inertia, stiffness, damping) taken into account. Generally speaking, this is a model equation of a linear system that is useful for dynamic analysis and controls design. The equation is generic and with the proper constraint matrix Φ , it can represent all possible transmission and drivetrain system structures in different transmission conditions, gear states as well as the drivetrain structures including those during gear shifting. It can also handle different the system input and output scenarios for different problems of interest.

System Sensitivity Analysis

In a vibrational mechanical system represented by eq. (3), component inertias $\bar{I}$ and the interacting stiffness K determine fundamental vibration modes including the associated natural frequencies and relative strength of vibrations among the relevant nodes, i.e., the mode shape. To identify possible system vibration tendencies and the corresponding underlying structures, it is usually sufficient to conduct analysis based on the undamped system model with possible parasitic losses ignored for simplification. In reality, the behavior of an undamped system is usually very close to the real system, unless the proper damping is carefully introduced, and in which case, the vibration tendency at a certain frequency is still present and the vibration energy is dissipated through the damping. For the analysis and system identification purpose, we can focus on the undamped version of the system equation. If we set $C = 0$ in eq. (3), it becomes

$$Mx + \bar{K} \bar{\varphi} = \bar{T}. \quad (5)$$

With eq. (5), the corresponding system characteristic equation can be derived and written as the homogenous equation of free motion in eq. (6) [4]. This question is useful in characterizing the system dynamic nature.

$$\text{Det}(\bar{K} - M\omega^2) = 0. \quad (6)$$