

Application of CAE in Design Optimization of a Wet Dual Clutch Transmission and Driveline

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ABSTRACT

This paper describes the application of CAE tools in the design optimization of a DCT and driveline system of a passenger vehicle, with emphasis on NVH performance. The multi-body dynamics simulation tools are employed for driveline system analysis. The MBD model consists of the engine, transmission, clutch, drive shafts, tires and vehicle. The wheel slip effects are considered in the calculation of shuffle frequencies.

In the analysis of gear whine, the transmission housing, gears and shafts are modeled by detailed 3-D finite element models, so that the mesh stiffness of the gears and the housing support stiffness are described more accurately. The calculated velocity spectra of the housing are presented.

The prediction of gear rattle in the transmission is carried out. The loose gear acceleration index and the averaged impact power of free gears are calculated to assess the rattle generation potential and the level of rattle severity. The influence of the clutch spring rate and the gear backlash on rattle behavior is investigated.

The dynamics of a new parking brake of the DCT is simulated. The parking brake lever force and shift torque in relation to shift angle are predicted. The shift contour of adjustment lever is optimized for better shift feeling and shift comfort. The model was validated through a comparison between the simulated shift torque and the measured one.

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INTRODUCTION

To meet stringent emission and fuel consumption requirements, SAIC Motor has developed an advanced 6-speed wet dual clutch transmission with an input torque range of 360 N-m, which is now in mass production. The dual clutch transmission (DCT) is applied to ROEWE 550 passenger vehicle equipped with a 1.8L turbo or 1.8L natural aspirated gasoline engine as shown in Figure 1. The newly developed DCT has two independently operated clutches for odd and even gear sets. The transmission features compact layout of gears and shafts, wet dual clutch hydraulic system for actuation of wet clutch and forks, integrated parking system, integrated fork and synchronizer system. To achieve compact package target, a coaxial dual clutch with integrated damper system, two countershafts system, and optimized layout of gear system are adopted. The technologies for fuel consumption reduction include waved clutch plates, targeted cooling of wet clutch, optimized gear ratios and optimized control strategies. The adopted technologies for achieving improved driving performance include optimized control on

clutch pre-filling and clutch swap, close loop control on fork movement with anti-force and optimized control on different driving modes.



Figure 1. ROEWE 550.

To improve product quality and robustness, a virtual verification process is integrated into the design process of the DCT and the driveline. In the design stage, attention was paid to the dynamics analysis of the transmission and driveline systems to avoid potential NVH problems due to inadequate design. Through front loading of CAE, more design iterations were

studied for improved maturity level of the prototype. For example, with a MBD driveline model, the clutch stiffness was optimized to minimize the driveline vibration level and the stiffness of half-shafts was tuned to alleviate shuffle and clunk phenomena.

A NVH issue often is a system problem of the entire drive train, such as significant dynamic responses of the system or components. An analysis of the system dynamics can provide an insight into the low frequency behaviour of the driveline. Therefore the numerical simulation is one of the tools for root cause analysis and for planning of NVH improvement strategies.

So far most published papers in the area of NVH simulation focused on conventional manual box [1,2,3,4]. A detailed study of the dynamics behaviours of the driveline with a DCT is absent. As more car manufactures are developing advanced DCT and other innovative components in a drive train, it is important to fully understand the effect of each component in the system with a realistic mathematic model.

NVH issues of a manual transmission may also be found on a DCT. In Table 1, typical driveline NVH issues are listed. Click-clack occurs more frequently in DCT, which is caused by the impact of the structure of the clutch when the engine torque is changed abruptly and the gear is shifted or during creeping between first gear and reverse gear. The click-clack problem may be solved by using appropriate control strategy. At part loads and during acceleration at low gears, gear whine may occur. At low engine speed and high load conditions, gear rattle may occur in the DCT. Another potential NVH issue in the DCT is shudder, which is a kind of clutch vibration caused by the negative damper due to the friction characteristic of the clutch at high temperature, i.e. the decreasing friction coefficient with increasing slip speed.

The purpose of this paper is to present the use of analysis tools to investigate NVH behaviors of the DCT and the driveline system. Efforts are made to consider the effects of wheel slip and the influence of tire stiffness in a simple way. Detailed MBD predictions of whine and rattle of the transmission system are presented. The influence of the non-linear factors such as the clutch spring rate and the gear backlash on rattle behavior is studied through simulation examples.

In addition, the dynamic behavior of a new parking brake of the DCT is simulated, which is, to some extent, related to the driveline through a differential.

DRIVELINE SIMULATION

Shuffle phenomena occur when engine torque changes suddenly. This dynamic response of the first mode torsional vibration may be excited under a tip-in or tip-out condition or gear shifting conditions.

Linear Model

In a comprehensive MBD model, the rigid powertrain model, crank train torque/speed fluctuation, components in driveline path (including clearances in the system model, non-linear contacts), tires, mounts, suspensions and vehicle are included. The level of model complexity depends on the type of problems and specific need. Using a complicated approach does not mean that a better result can be obtained, compared with using a simplified approach. The classic linear three-inertia vibration model consisting of the engine, transmission, drive shaft and vehicle was proven valid for comparative evaluation of vibration isolation options such as clutch torsion dampers. In this work, a modified multi-inertia model is employed for shuffle simulation. As shown in Figure 2, the simplified lumped parameter model of a FWD driveline is built with Ricardo's VALDYN software, which consists of the engine, transmission, tires and vehicle. The tires are included to consider the damping effect, and the asymmetric half-shafts are modeled. Here is I_{En} the equivalent moment of inertia of the engine. I_{TR} is the gear box equivalent moment of inertia. K_{Clutch} is the clutch working stiffness. $K_{Drive\ shift_L}$ and $K_{Drive\ shift_R}$ are the equivalent stiffness of the left and right drive shaft, respectively. I_{Tire_L} and I_{Tire_R} are the equivalent moment of inertia of the left and right tire, respectively. K_{Tire_L} and K_{Tire_R} are the equivalent stiffness of the left and right front tire, respectively. When the equivalent stiffness of tire approaches infinite, this model reduces to the classic three-inertia vibration model.

Table 1. Typical driveline NVH issues

NVH issue	General frequency range	Key influence parameters	Note
Shuffle	2~10 Hz(vibration)	Stiffness of half-shaft	
Shudder	90~14 Hz(vibration)	Inertial of clutch disk, friction characteristic, damper of the components of driveline	MT, DCT
Low frequency clunk	10~50 Hz (vibration)	Inertial of gear, stiffness of gear shaft, stiffness of half-shaft	MT, DCT
Booming	80~120 Hz (vibration)	Inertial of differential, stiffness of parallel shaft, stiffness of half-shaft	RD, AWD
High frequency clunk	400~1000 Hz (vibration)	Inertial of gear, stiffness of gear shft, meshing stiffness of gears	MT, DCT
Whine	>200 Hz (broadband noise)	Micro geometry of gear, mode of gearbox case, tooth ratio of gears, shaft deflection	MT, DCT
Rattle	30~60 Hz (vibration), 1000~7000 Hz broadband noise	Stiffness and hysteresis damper of clutch, gear inertial, layout of gear shaft, drag torque of loose gears, tooth backlash	MT, DCT
Click-clack	5~10 Hz (vibration)	Stiffness of gear shft, stiffness of half-shaft, control strategy of the clutch	MT, DCT

I_{Vehicle} is the equivalent moment of inertia of the vehicle,

$$I_{\text{Vehicle}} = M_V \cdot \frac{R^2}{SR^2} \quad (1)$$

Here M_V is the vehicle mass. R and SR are the tire radius and the effective speed ratio, respectively.

The equivalent moment of inertia of each component equals to the moment of inertia divided by the square of speed ratio. The equivalent stiffness of each component equals to the torsional stiffness divided by the square of speed ratio.

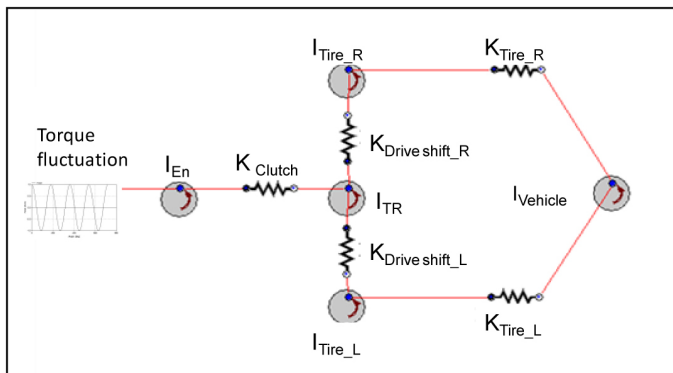


Figure 2. The linear multi-inertia vibration model.

A constant torque fluctuation is applied as the excitation input to the model. From the model parameter study, it is found that the drive shaft torsional stiffness influences the primary mode of the driveline, which is referred to as shuffle. When the drive shaft stiffness increases from 80 /88 Nm/deg (left /right) to 100/110 Nm/deg (left /right), the shuffle frequency is increased from 2.18 Hz to 2.38 Hz and the magnitude of the vehicle acceleration is reduced by 35.88%. It is concluded that the modified multi-inertia model is useful to compare the effects of different design parameters on the driveline nature frequencies and acceleration response of the vehicle.

Non-Linear Model

Non-linear factors are gear clearance, spline clearance, clutch multiple stiffness and hysteresis. To consider these non-linear effects, a detailed FWD system model is created with AMESim software to simulate the driveline system. Figure 3 shows the vehicle response to the change of engine's throttle body opening. In the tip-out phase (at first gear), the vehicle's speed and acceleration show transient behaviors and the decay of fluctuations. The FFT analysis of the acceleration curve in the relevant phase indicates that the resonant frequency is very close to the result from the multi-inertia vibration model. From the modal shapes, it is seen that the vibration amplitude and the phase of the engine and the transmission are very close. The vibration amplitude of the vehicle is the lowest, and the phase difference between the vehicle and the engine is 180 degree.

The detailed driveline analysis provides an insight into the low frequency behaviour of the driveline and engine mounting system. Beside hardware design, the shuffle problem can be addressed with controller algorithms. There are two engine control strategies to dampen the longitudinal vehicle oscillations. In the torque gradient control concept, decreasing the gradient of engine torque through engine control algorithms is used to reduce the amplitude of driveshaft torque. In the anti-phase shuffle control concept, modulation of vehicle response is done through engine torque phasing or active engine control.

Wheel Slip Effects

It is known that the shuffle characteristic of a driveline on a low friction (low- μ) surface is different as that on a high friction surface [5]. Because of the low levels of available traction on the low- μ surface, the effective equivalent moment of inertia of the vehicle drops as function of traction. Thus, it is necessary to derive an equation which relates the effective equivalent moment of inertia of the vehicle to the slip ratio.

The slip ratio of a wheel is defined as

$$\lambda = \frac{\omega^* - V_{\text{veh}}/R}{V_{\text{veh}}/R} = \frac{\omega^* - \omega}{\omega} \quad (2)$$

where ω is the angular speed of the wheel without any sleep, ω^* is the effective angular speed of the wheel when slip occurs. V_{veh} is the vehicle speed, which is assumed fixed. According Equation (1) and Equation (2), the following equation is obtained,

$$I_{\text{Vehicle}}^* = \frac{1}{(1+\lambda)^2} I_{\text{Vehicle}} \quad (3)$$

where I_{Vehicle}^* is the effective equivalent moment of inertia of the vehicle when the wheel is slipping, and I_{Vehicle} is the equivalent moment of inertia of the vehicle without wheel slip.

The analysis model for the high- μ surface case can be reused for the low- μ surface case, in which the model properties of the system are changed accordingly. Figure 4 shows the calculated shuffle frequencies as function of the slip ratio. Here Equation 3 is employed to calculate the equivalent moment of inertia of the vehicle for different values of slip ratio. It shows that the shuffle frequency is increased with the increasing of the slip ratio, and it is increased with the increasing of the equivalent stiffness of tires. It is concluded that on the low- μ surface, as wheel slip occurs during vehicle acceleration at high engine torque conditions, the shuffle frequency is higher compared with that of the high- μ surface case.

In the proposed model the damping of the tire, either positive damping for low slip velocity or negative damping for high slip velocity, is not considered. The influence of the damping on