

Optimization of Front Wheel Drive Engine Mounting System for Third Order Shudder Improvement

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ABSTRACT

Nowadays, the vehicle design is highly ruled by the increasing customer demands and expectations. In addition to ride comfort and vehicle handling, the Noise, Vibration and Harshness (NVH) behavior of the powertrain is also a critical factor that has a big impact on the customer experience. To evaluate the powertrain NVH characteristics, the NVH error states should be studied. A typical NVH event could be decoupled into 3 parts: source, path, and receiver. Take-off shudder, which evaluates the NVH severity level during vehicle take-off, is one of the most important NVH error states. The main sources of Front Wheel Drive (FWD) take-off shudder are the plunging Constant Velocity Joints (CVJ) on the left and right half shafts. This is because a plunging CVJ generates a third order plunging force with half shaft Revolution Per Minute (RPM), which is along the slip of the plunging CVJ. The primary path of take-off shudder is the Engine Mounting System (EMS), which isolates the vibration inputs from the vehicle body. A typical receiver of shudder is the passenger seat, so seat track acceleration and velocity are usually chosen to be the design objective for vehicle NVH optimization. This paper presents the optimization of FWD engine mounts for third order shudder improvement. Pointer automatic optimizer is used to perform the optimization with respect to a large number of design variables.

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1. INTRODUCTION

Nowadays, vehicle design is highly ruled by the increasing customer demands and expectations. In addition to ride comfort and vehicle handling, the NVH behavior of the powertrain is also a critical factor that has a big impact on the customer experience. There are a large number of NVH error states which need to be studied, such as take-off shudder, tip-in shuffle, shift boom, moan, combustion instability, and so on. A typical NVH error state could be decoupled into of 3 parts: source, path, and receiver. Currently one of the most popular research areas in vehicle design is the optimization of the path and source for the NVH response improvement.

There is a large body of literature for vehicle NVH path and source optimization. Bernard, 1983, showed a design procedure to be effective in selecting stiffness, orientation, and location of engine mounts which removes engine natural frequencies from the range excited at idle [1]. Geck, P, 1984, presented a guide for FWD transverse engine mount optimization using MONTE CARLO procedure [2]. Bretl, 1993, presented a new simulation-based method for design of a EMS using optimization based on response sensitivities [3]. Marco La Civita, 1999, developed an optimization

procedure based on genetic algorithms for a EMS to provide good comfort conditions [4]. JS TAO, 1999, presented design optimization of marine engine mount system to minimize the vertical force transmitted from the engine to the floor to control the structure-borne noise [5]. T Naruse, 2000, described a method for optimizing the performance for vehicle handling low frequency vibration in a EMS supporting on a torque roll axis [6]. D Blanchet, 2000, presented an efficient and straightforward approach to develop and optimize the engine suspension for the improvement of several design objectives, such as transmitted forces, power flow injected in the elastic base structure, vibration levels at critical locations, and driver ears acoustic pressure levels [7]. A Farshidianfar, 2002, presented a complete dynamic model of a vehicle driveline for the optimization of high-frequency torsional vibration by the Distributed-Lumped Modelling Technique (DLMT) [8]. Ma, Zheng-Dong, 2004, developed a systematic approach for the multidisciplinary design optimization of a EMS and the stability analysis and optimization [9]. Chang Yong Song, 2010, performed design optimization of engine mount for passenger vehicle using MSC.ADAMS based engine mount system analysis program EMTTOOLS [10]. Yaddan Rasekhipour, 2011, presented the hydraulic engine mounts are

optimization to improve the vehicle ride comfort using a global optimization method called Directed Tabu Search (DTS) method [11]. Findeisen, B. 2013, presented the coupling of Multi Body Simulation (MBS) with mathematical optimization tools exemplary for a powertrain mounting at a passenger car vehicle [12]. Alkhatib, 2013, presented techniques for the design of a EMS that takes into account modeling of the mount characteristics, the excitation load present in the system, and a determination of the final geometrical shape of the engine mount [13]. Eric Courteille, 2013, introduced a new method to support designers to find optimal and robust solutions of a EMS, which is a compromise between isolation of the vehicle from engine vibration and constraining the motion of the powertrain within vehicle packaging [14]. Xia Hua, [15, 16, 17], creatively developed cutting-edge effective mathematical models and advanced analytical approaches of a vehicle driveline for the design optimization to minimize driveline steady-state dynamic responses by applying the enhanced lumped parameter and finite element modeling techniques.

Most of these papers focus on the study of steady state error states. In contrast, the transient error states are rarely studied. Take-off shudder, as one of the typical transient error states, evaluates the vehicle NVH severity level during take-off. There are two methods to optimize the NVH response on the receivers: modifying the sources and modifying the paths. For third order shudder, the sources of shudder are the plunging forces on the plunging CVJs; the path is the EMS. As shown in equation (1) and figure 6, the amplitude of the plunging force is dependent on the CVJ angle. However, modifying the CVJ angle might have negative effects on other NVH error states. Thus, modifying CVJ angles is not feasible. The optimization of the path is

also not easy since there are a large number of design variables in the path, so it's necessary to find a tool that is able to perform optimization with respect to multiple design parameters efficiently. This paper presents the optimization of EMS on a FWD vehicle for third order shudder improvement. The paper is organized as follows. Section 2 is the problem statement, which describes the source, path, and receiver for third order take-off shudder. In section 3, the modeling of a FWD vehicle is described. In section 4, the dynamic analysis for shudder is performed first and the pointer automatic optimizer is then used to perform the optimization of third order shudder. Section 5 is the conclusion.

2. TAKE-OFF SHUDDER DESCRIPTION

Take-off shudder evaluates the NVH severity level during take-off. As one of the typical transient NVH event, it's composed of three parts: source, path, and receiver. As shown in figure 1, the shudder is generated by the source and transferred to the receiver through the path. To minimize the dynamic response on the receiver, the optimization of path becomes critical. Thus, thorough understanding of the source, path, and receiver for a NVH event is necessary. In this study, the source of FWD take-off shudder is the plunging CVJs on the left and right half shafts. This is due to the fact that the plunging CVJ generates a third order plunging force along the slip of the CVJ. The path of take-off shudder is the EMS, which isolates the vibration inputs from the vehicle body. Passengers are the receivers of shudder, so seat track acceleration and velocity are usually chosen to be the design objectives for vehicle NVH optimization.

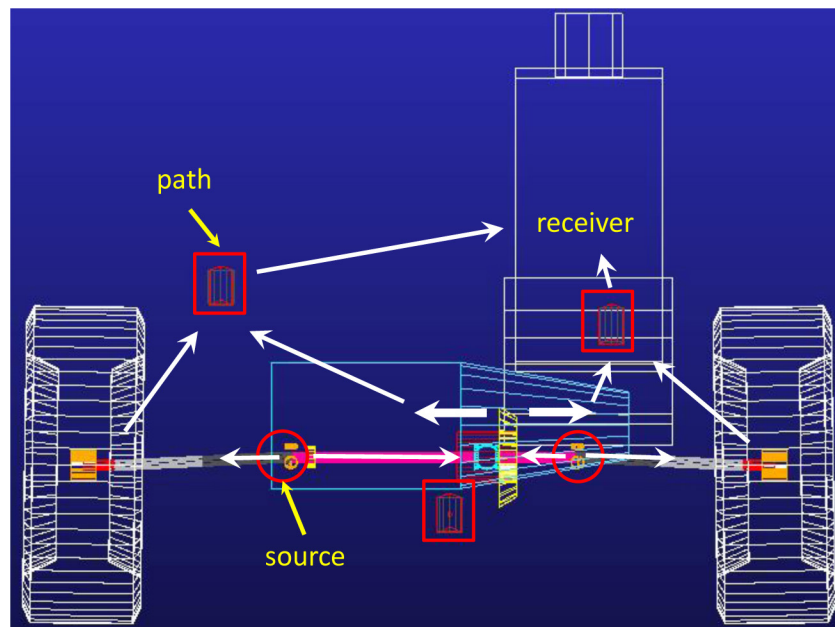


Fig. 1. Shudder transmission from source to receiver

3. FWD VEHICLE MODEL

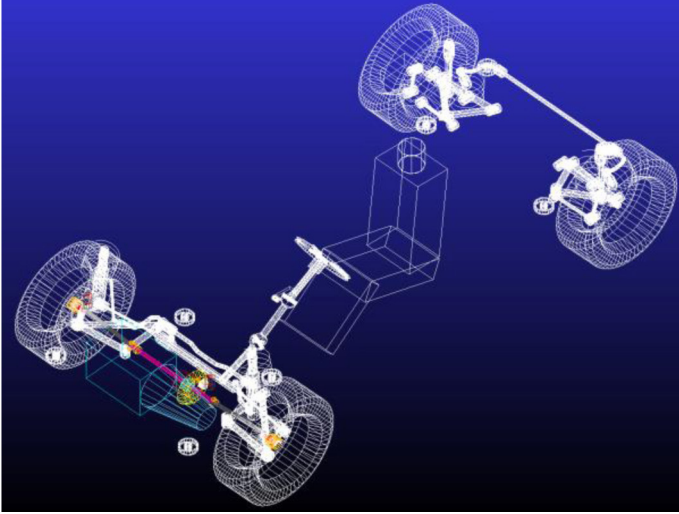


Fig. 2. FWD vehicle with a transverse engine

As shown in figure 2, an Adams model of FWD vehicle with a transverse engine is used as a case study. In the next section, the vehicle topology is introduced and the modeling of the plunging CVJ is described.

3.1. Model Topology

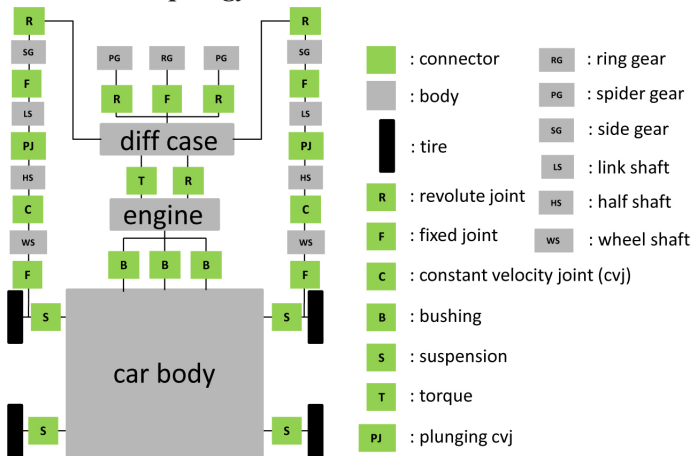


Fig. 3. Topology of the FWD vehicle model

The vehicle topology is represented in figure 3, which includes the car body, suspensions, tires, and the powertrain. This model contains 389 Degrees Of Freedom (DOF), 131 moving parts, and 102 connectors. For the sake of the space and simplicity, only the parts and connectors that are related to the take-off shudder event are listed in figure 3. Since the design of the car body is not of interest in this study, it is simply modeled as a single rigid body. Macpherson suspensions and trailing arm suspensions are used, respectively, in the front and rear. The engine, which is also modeled as a single rigid body, is connected to the car body using two engine bushings and a

roll restrictor bushing. Since the transfer case and transmission are not modeled here, the differential case is directly connected to the engine body by a revolute joint on which the torque curve is applied. Two spider gears and two side gears are connected to the differential case by four revolute joints. The link shaft is fixed to the side gear. The plunging CVJ on the inboard side of the half shaft is referred to as inboard CVJ. The inboard CVJ is used to connect the link shaft and half shaft. The outboard CVJ, which is a fixed CVJ, is used to connect the half shaft and wheel shaft. Finally, the wheel shaft is fixed to the wheel hub, which is then fixed to the tire.

3.2. Plunging CVJ Modeling

The source of take-off shudder for this study is the plunging CVJ. The plunging CVJ is simply modeled using a CVJ, a translational joint, and a plunging part. The topology of the plunging CVJ is shown in figure 4 as follows:

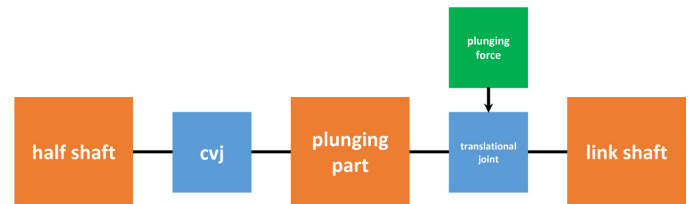


Fig. 4. Topology of the plunging CVJ

As shown above, the half shaft is connected to the plunging part by a CVJ. The link shaft is connected to the plunging part by a translational joint. The plunging force is modeled and applied directly on the translational joint. This plunging force is a third order harmonic excitation along the plunging direction. The reason this force is called a third order force is that the driving frequency ω^* is three times of the half shaft angular velocity ω , as shown in equation (1) and (2). The root cause of this third order force is the three-roller structure of the CVJ [18].

The amplitude of the plunging force is dependent on the half shaft torque and the inboard CVJ angle. The inboard CVJ angle is the angle between link shaft and drive shaft as shown in figure 5. The half shaft torque is the total torque lumped on the drive shaft.

Figure 6 is the color map that gives the amplitude of the plunging force for given inboard CVJ angle and half shaft torque. The plunging force function is shown as follows:

$$F = A(\theta, T_q) \sin(\omega^* t) \quad (1)$$

$$\omega^* = 3\omega \quad (2)$$

where F is the plunging force, $A(\theta, T_q)$ is the amplitude, θ is the CVJ angle, T_q is the half shaft torque, ω^* is the driving frequency, and ω is the half shaft angular velocity.