NVH Optimization of Vehicle Powertrain

Shouwei Lu, Huihua Feng, Zhengxing Zuo, Liyun Kang and Bo Yu

Abstract The noise and vibration performance of powertrain is main contributor for vehicle NVH (Noise, Vibration and Harshness) issue. To achieve better NVH performance, it is critical to conduct NVH optimization during the powertrain initial design stage. This paper presents an investigation of optimize vehicle powertrain NVH performance via modification of excitation–radiation system of powertrain. To minimize excitation force of the gearbox with special focus on gear pair dynamic characteristics via the gear profile modification, and to reduce transmission housing noise radiation via enhance its stiffness, are the main objective of optimization. The excitation forces are analyzed by Multi Body Dynamics (MBD) method, considering different excitation mechanisms of the powertrain. The vibro-acoustic behavior of powertrain is obtained by FEM/BEM coupled analysis. The acoustic transfer vector (ATV) calculation is used to predict the powertrain sound power level (SPL) and panel contributions. Based on the acquired NVH data of the powertrain, the optimization which couples the transmission gear profile modification for attenuating gear system excitation and the structure stiffness modification for reducing transmission housing noise radiation is proposed. Experiment validation is conducted in order to evaluate the modified results. The evaluation shows that the optimization can effectively reduce powertrain noise and vibration.

Keywords NVH · Powertrain · Transmission · Profile modification

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1 Introduction

The increasing safety, quality and comfort demands of customers have become important indicators for making market strategies of automotive manufacturers [1]. Excellent NVH performance of vehicles is essential for manufactures to gain the opportunities to win the competition and dominate the market in the global automotive industry. Particularly, the NVH performance of vehicle essentially depends on powertrain characteristics. Less noise and vibration, good sound quality powertrains, which can be applied to meet the customer demands, are needed in the vehicle design and development process.

In general, the powertrain noise and vibration optimization can be divided into three main solution process: reduce excitation, optimization of structural transfer behaviour and modified radiation surface [2, 3]. This paper presents a systematic analysis process to optimization powertrain NVH performance. Initially, the powertrain excitation forces are calculated through MBD method. The calculated excitation result is applied as input data for powertrain vibro-acoustic evaluation. The transmission gear profile modification for attenuating gear system excitation is performed. Transmission near-filed sound pressures distribution analysis is carried out to find out critical areas effecting the noise radiation. Based on the critical areas of weak stiffness identification, transmission housing stiffness is optimized by modify these critical areas. Finally the results of optimization have been verified with experimental result in this study.

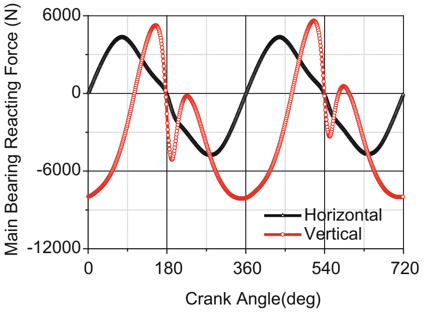
2 Excitation Force Analysis of Powertrain

Since the key parts of the powertrain, transmission and engine, are different in excitation mechanism, it is absolutely essential for the selection of different analysis tools to achieve accurate calculation. The detailed investigations for excitation mechanisms of the powertrain by analysing the excitation forces are conducted, depending on different analysis process. For powertrain dynamic characteristic evaluation, four main excitation forces are required as input data:

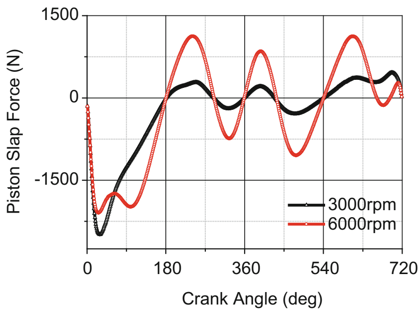
- boost pressure
- engine main bearing reacting force
- piston slap force
- transmission bearing reacting forces.

2.1 Engine Excitation Force

There are two main noise sources of ICE (internal combustion engine): combustion noise and mechanical noise. Combustion noise is induced by the firing pulses



Main Bearing 3 Reacting Force in Crank domain



Piston Slap Force in Crank Domain

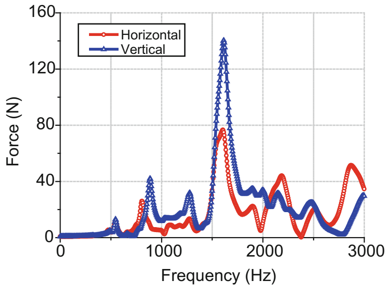
Fig. 1 Engine excitation force

due to the explosions of the fuel in the combustion chambers. The mechanical noise is induced by the various inertia forces and torque oscillations caused by the rotating and reciprocating parts. The major sources of engine mechanical noise are the main bearing reacting force, the piston slap force, timing gear rattle, fuel injection system, valve system and accessories operation. The main contributors among these sources are the piston slap force and the main bearing reacting force. The high excitation load caused by main bearing reacting force resulting from combustion gas pressure and inertia forces of reciprocating masses are transferred via the crank mechanism. The piston slap force is caused by the lateral motion of pistons across the cylinder clearances.

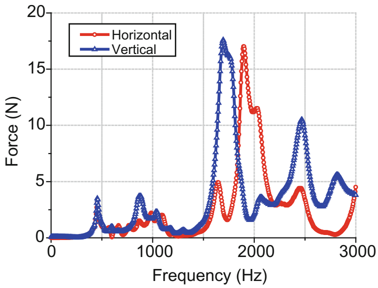
Figure 1 shows the bearing reaction force at main bearing #3 of a 4-cylinder inline engine, which is plotted as a curve of engine speed versus crank angle. The bearing reaction force depends on combustion force at certain operating condition with the engine firing order. Figure shows the piston slap force at different engine speed. To evaluate the engine excitation forces with respect to the relevant powertrain NVH issue, the data is transferred into frequency domain using Fast Fourier Transformation (FFT).

2.2 Excitation Force of Transmission

The excitation force of transmission is generated by transmission error (TE), rotary fluctuations of the gear shaft, and friction induced bearing forces, transmitting via transmission bearings and then radiating from transmission housing by means of vibration and noise [4]. The loads acting on bearings are not only influenced by the gear shift position changing, but also by the engine rotating speed variability. The calculation of the bearing reacting force is a difficult process, since it is the interaction for gear meshing, lubrication, transmission error and several mechanical effects. The detailed parameters of gears are required to establish a sophisticated analysis model. The transmission gear system model which is used to



Input shaft right bearing reacting force (1st gear)



Input shaft right bearing reacting force (2nd gear)

Fig. 2 Transmission excitation force

calculate the bearing reacting force is built by software ROMAX and the calculation results are shown in Fig. 2.

3 Forced Response Calculations

The main objective of this process is evaluation of powertrain structural transfer behaviour and surface velocities. The excitation forces are transmitted as structure-borne noise to the surfaces of the powertrain components where it is subsequently radiated as airborne noise [5]. In concept stage of powertrain development process, the forced response analysis provides basic dynamic characteristics of the powertrain to support designer making NVH optimized decisions. During the analysis process, a full powertrain model gradually develops as the design progresses. At the end of the process, the powertrain surface velocity is summarized over the frequency range 0–2,000 Hz associated with a specific load and engine speed. Using the principle of modal superposition, the response is calculated. The result is done in vector solution, which contains the response of a large set of outputs for different operating condition. Figure 3 shows the predicted vibration velocity distribution for powertrain.

4 Near-Filed Sound Pressure Distribution

In order to investigate the transmission housing panel acoustic characteristics at different operating condition, the near field sound pressure analysis is carried out, providing the structure stiffness information. Particularly for this powertrain, the differential cover has a significant large area, hence, is responsible for most of the noise. The near field point mesh is divided into four panels which are shown in Fig. 4.

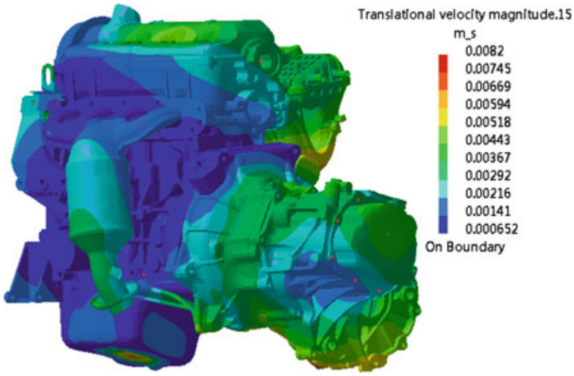


Fig. 3 Predicted surface velocity distribution

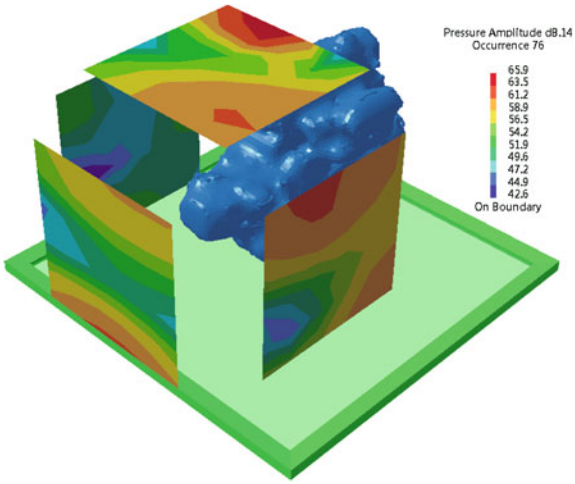


Fig. 4 Predicted sound pressure distribution

5 Transmission Housing Stifness Optimization

Based on the near field sound pressure distribution references, different modification strategies are carried out. In conventional optimizing process of the dynamic behaviour of structures, it is usually required to shift natural frequencies of system by making modifications to the mass, stiffness or damping characteristics of the structure. Rib stiffening is a frequently adopted method [6]. Therefore, in the transmission housing modification, reinforcement ribs are added on the transmission housing to enhance its stiffness. The layout of most effective position of stiffeners is designed by the previous analysis results of near field sound pressure distribution. Figure 5 shows the modified layout.

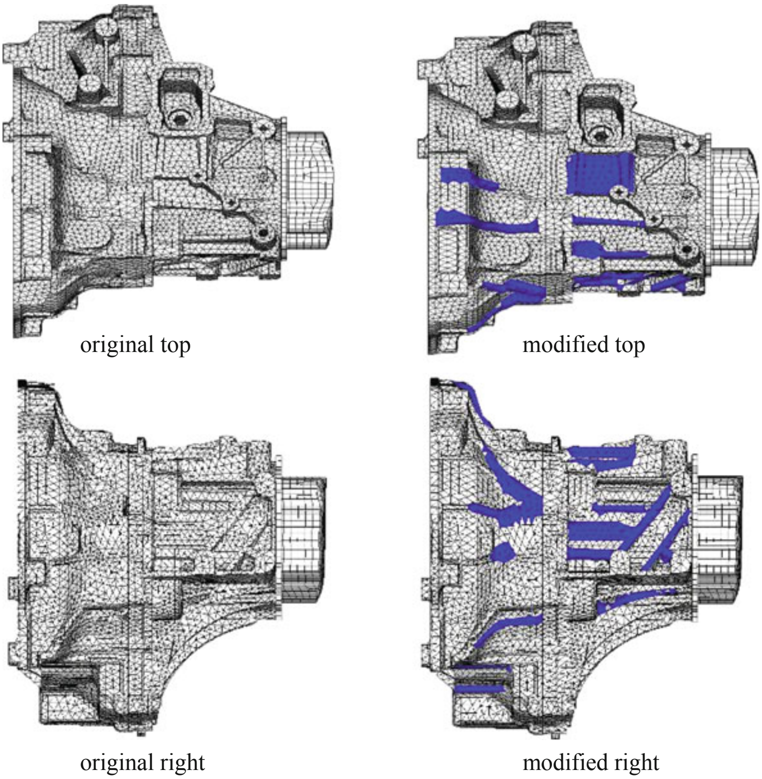


Fig. 5 Structure modified layout

6 Transmission Gear Profile Modification

Transmission gear systems have two types of noise sources: one known as the internal noise generating by gear mesh excitation, and the other known as the external noise generating by input torque fluctuation of engine. The most dominant noise source of gear system is from the gear mesh excitations of the gear teeth. More accurately, the gear mesh excitations are caused by transmission error fluctuation and fluctuation in the load transmitted by the gear mesh. Gear tooth profile modification is a widely used method to reduce dynamic excitation load for improving performance of transmission NVH performance. Different type of profile modifications has different effects on gear system performance [7]. The tooth profile modification technique, tooth crowning, was used to minimize bearing reacting force. The main aim of this methodology is to search for the optimum profiles of tooth crowning that eventually lead to optimum dynamic tooth load in the gear mesh. The 2nd gear pair modification is chosen for illustrating the gear profile modification process. The transmission error analysis of the 2nd gear

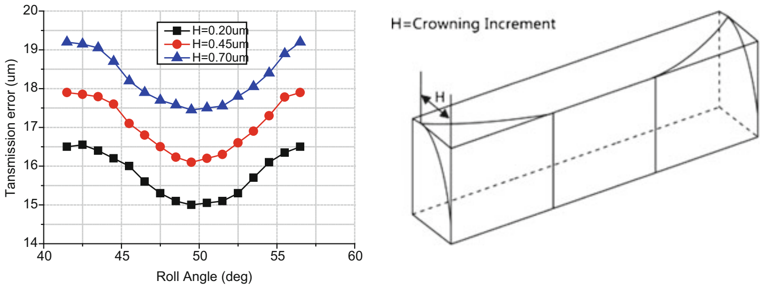


Fig. 6 The relationship between influence of tooth crowning increment and transmission error

was carried out first. The relationship between influence of tooth crowning increment and transmission error is given in Fig. 6.

The gear profile modifications that are often required to maintain a good contact for durability are not always conducive to low TE (good NVH performance) and some sort of balance must be considered. Thus, to achieve good NVH performance, the low TE was set as target of optimization of gear profile modification [8]. The changes of panel contribution and SPL comparison between original and modified are used to evaluating the validity of the transmission gear profile modification.

7 Experimental Validation

To evaluate the validity of the optimization performed in this study, an experimental validation is carried out. Figure 7 shows the test setup. The experiment is performed in semi-anechoic chamber under different torques and rotary speed. Three microphones are adapted to measure sound pressure of transmission. All of the microphones are perpendicular to longitudinal centerline of transmission, and the measurement distance is 1 m from the transmission housing to each of microphones respectively. The average sound pressure calculation is based on the equation given below:

$$\bar{L}_p = 10 \cdot \log \frac{1}{N} \cdot \sum_{i=1}^{N} 10^{0.1 \cdot L_{pi}} \tag{1}$$

Where $\bar{L}_p$ is the A-weighting average sound pressure level of the measured points; L_{pi} is the A-weighting sound pressure level of i measured point; N is the total number of measured points.

Figure 8 show the experimental results under different running conditions. The results shows that the optimization, which couples the transmission gear profile modification and the transmission housing stiffness modification, can reduce the vibration and noise of transmission to achieving the improvement of NVH performance of the powertrain. However, comparison from the Fig. 8 shows that this

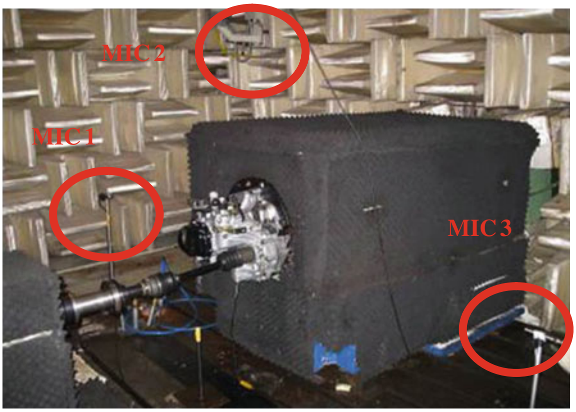


Fig. 7 Experiment setup

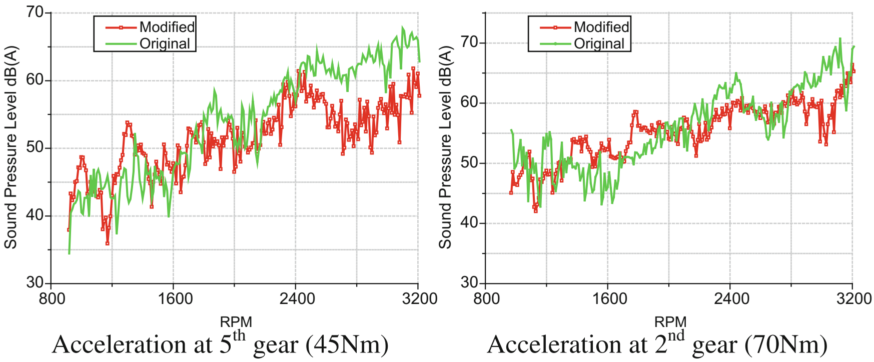


Fig. 8 Sound pressure level comparison between original and modified

hybrid approach is not validity enough during the rpm range (1400–1800 rpm). This is due to excitation force order coincide with the natural frequency of transmission housing, generating high level of vibration and noise. A precise investigation of dynamic behaviour for gear system is necessary in further research.

8 Conclusion

NVH optimization process of a powertrain is proposed. From the beginning of excitation analysis to the end of panel radiated noise contribution analysis, a detailed analysis process is present. Based on the difference of excitation mechanism, engine excitation analysis model and transmission gear system model, are developed. The excitation models are the most critical models for whole research process. The calculated results of excitations show that all the excitations are related to the harmonic order which can easily draw from the excitation force results in frequency domain. Through extensive dynamic and acoustic analysis, it is found that not only the dynamic characteristic which is reflected by the powertrain surface vibration velocity, but also the acoustic characteristic which is reflected by the powertrain sound pressure are all involves with the excitation force. That means excitation force optimization can achieve dramatic noise and vibration reduction. Both the excitation force and radiated surface are modified, thus making the NVH improvement more achievable. For reasonable evaluation of validity of modification, validation experiment is conducted under various running condition in the semi-anechoic chamber. The average sound pressure level is calculated using measurement data by Eq. (1). Despite a little increase of sound pressure level during certain rpm range, the overall sound pressure level decreased dramatically. In this case, the NVH optimization of the powertrain, combining transmission gear profile modification and transmission housing modification, can significantly reduce powertrain noise and vibration to a sufficiently desired level. In further research work, investigations should focus on the transmission gear system dynamic behaviour.

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