SENSITIVITY ANALYSIS OF A LIGHTWEIGHT REAR DIFFERENTIAL UNIT ON RIDE COMFORT OF A SPORT UTILITY VEHICLE

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Abstract

Lightweight is an important measure to the fuel efficiency and the cost reduction in development of contemporary vehicles. However, how the lightweight structure could affect the NVH performance or ride comfort of a vehicle needs to be carefully investigated. In this paper, the effect of a lightweight rear differential unit of a sport utility vehicle on ride comfort was studied. A multi-body dynamic model of the vehicle was developed which includes powertrain and driveline, tyres, suspensions, steering system, and a flexible car body. The baseline model was correlated with the accelerations measured at the seat rail in a field test. Using the correlated model, a factorial analysis of Design of Experiment was conducted to study the sensitivity of the masses and the moments of inertia of rear differential components on the vibration at the seat rails in terms of accelerations r.m.s. values. Results showed that the seat rail vibration energy was centred at two frequency ranges: 1-2 Hz and 6-15 Hz. The vibration in these two frequency bands in different directions had different sensitivities to the variation of the mass properties of the rear differential unit. It was found that the vertical vibration of the seat rail was most sensitive (in descending order) to: the mass of the rear differential housing, the moment of inertia of the inner constant velocity joint about the longitudinal direction, and the mass of the driveshaft. In the context of this study, they are three most significant contributors to the ride comfort that need to be considered when designing a lightweight structure for the rear differential unit.

1. Introduction

Power performance and fuel efficiency are important measures of a vehicle as well as the noise, vibration and harshness (NVH). To reach a greater fuel efficiency and a lower cost, lightweight structures are increasingly used by automobile manufactures. Nevertheless, the effect of weight reduction on NVH, which is always a strong influencing factor for customer decisions, needs to be well considered. Among various NVH factors, the ride comfort is at a high level of concern which has been highly advocated by relevant standards such as ISO 2631 (1997) and BS 6841 (1987).

Vehicle is such a complex and coupled dynamic system that any changes of its parts may have significant influence on the vibration transmitted to seats and occupants. The road irregularity and the engine unbalancing forces are the two main excitation sources. In idle condition, vibration of engine and driveline is the main input that causes seat vibration. When a vehicle is travelling on a road with the crankshaft exceeding the first critical speed, the road roughness will be the predominant input in relation to the seat vibration (Ahlin and Granlund, 2002). With a vehicle travelling on a road at speed (e.g., at 60 mph), the vibration excited by road roughness is transmitted to chassis including

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suspensions, to tsub-frames and car body through various bushings or mounts, finally to seats and occupants.

There are many transfer paths and contributing factors affecting the seat vibration. To find a purposeful optimization objective and evaluate the effect of weight reduction on ride comfort in a (vehicle) system level, multi-body dynamic (MBD) simulation is one of the most efficient approaches and has been proven reliable and useful (e.g., Yang *et al*., 2009). To utilise this method, the first step is to develop a suitable multi-body dynamic model of the vehicle. This model should include the main subsystems or substructures so as to properly reflect vehicle and seating dynamics (Kortüm, 1993). The structure or complexity of the vehicle model depends on types of the excitation, the vibration transmission path involved, and the scope of application. A representative model of a seat and occupant system may be sufficient for sensitivity analysis for identifying key parameters affecting the ride comfort and providing useful information for optimisation of seat comfort (Brogioli M *et al*., 2011; Qiu and Griffin, 2011). A model comprising wheels, front and rear suspensions, and car body may be suitable for analysing ride comfort when a car is travelling on a road (Goncalves and Ambrosio, 2005). A detailed car model focusing on the hydro-pneumatic spring-damper suspension system was used to adjust the suspension settings for optimising ride comfort (Uys *et al*., 2007). It is assumed in this study that a model consisting of tyres, suspensions, driveline system and flexible car body would be appropriate for analysing ride comfort and performing sensitivity analysis to identify significant factors among the mass properties of the driveline that affect the seat vibration.

Transmission driveline generally consists of transmission gearbox, prop shafts, front and rear differentials, and drive shafts as well as connecting joints and mounts. Changes of the components may affect the vibration transmitted to the seat, and the degree of effects may vary in different directions and frequency regions. It is desirable that a sensitivity analysis is carried out before modification of parts to check the contributions and the effects of the changing components (Chen *et al*., 2010). Sensitivity analysis is powerful in identifying contributions of design parameters to the objective performance of a system. Performing a sensitivity analysis by modifying one parameter per time is practical and relatively easy to implement (e.g., Braghin *et al*., 2005; Qiu and Griffin, 2011) but has limitations in computational efficiency and costing. Design of Experiment (DOE) can help find cause-and-effect relationships between factors and outputs efficiently and can be carried out in different methods, such as the factorial design (Plackett and Burman, 1946), the response surface design (Box and Draper, 1987), the mixture design (John, 1984), and the Taguchi design (Peace, 1993). The 2-level factorial design is used in most DOEs because it is simple, versatile and can be applied to cases with many factors (Xu *et al*., 2009). There may be two main advantages of the 2-level design: the size of experiments is much smaller than other designs, and the interactions between the factors can be detected (Bingham and Sitter, 2001).

The objective of this study was to conduct a sensitivity analysis to identify how mass properties of a rear differential unit and related driveline could affect the seat rail vibration and hence ride comfort. To this end, a vehicle model was developed in a multi-body dynamics environment. The vehicle model was correlated against the measured data in a field test. A sensitivity analysis was then conducted using Design of Experiment method to identify significant factors among the mass properties of the driveline and their contributions to the seat rail vibration. It is expected that the analysis results can provide a guidance on the mass reduction of the rear differential unit and adjacent parts and to understand the effect of the mass reduction on ride comfort.

2. Development of the multi-body dynamic model

2.1 Construction of the model

The MBD model of the vehicle (LHS drive) initially provided by a project collaborator was developed using SIMPACK software (Version 9.7). The model consists of tyres, suspensions, powertrain and driveline, and a flexible car body, as shown in Figure 1 (the car body is not shown). The tyres were established using the Flexible Structure Tire Model (FTire) suitable for relatively high-frequency and short-wave-length road excitation. Nonlinear air springs, hydraulic mounts and dynamic bushes were adopted for modelling the main suspension and the power unit and differential mountings. The driveline includes the front and rear differential units, prop shafts, drive shafts and the adjacent constant velocity (CV) joints. The flexible car body was built up in NASTRAN using the finite element (FE) method and its modal model was imported into SIMPACK to be combined with the MBD model of the car.

In the simulation, the road roughness and the air resistance were considered as external disturbance and forces applied on the vehicle. The road roughness defined with the model was measured from a smooth (B) road provided by the project collaborator and the air resistance was calculated as below,

$$F_{air} = -\frac{1}{2}\rho_{air}A_{veh}cv^2$$

$$M_{air} = -\frac{1}{2}\rho_{air}A_{veh}c_M v^2 l$$

where F_{air} and M_{air} are the air force and moment, ρ_{air} is the air density, A_{veh} is the characteristic area of the vehicle, c and c_M are the constant air resistance coefficients for force and moment, v is the vehicle velocity, and l is the vehicle wheelbase.

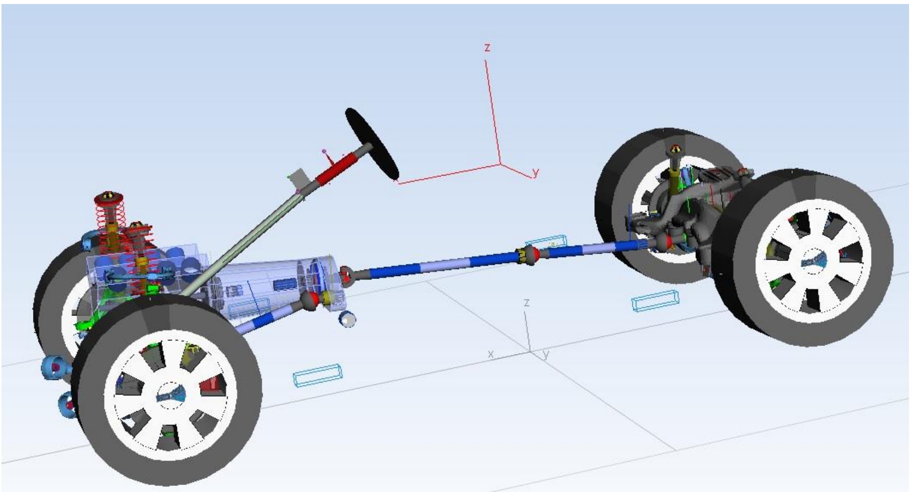

Figure 1 Multi-body dynamic model for the sport utility vehicle.

2.2 Calibration of the model

Before conducting the sensitivity analysis, the developed model was correlated with the experimental data measured from a field test. The accelerations at the seat rails on the driver side were measured with the vehicle travelling at 60 mph on the same (B) road as adopted in the model simulation. The power spectral density (PSD) of the seat rail acceleration in the vertical direction simulated with the correlated model is overlaid with the corresponding measured data, as shown in Figure 2. It can be seen that the model was correlated reasonably well.

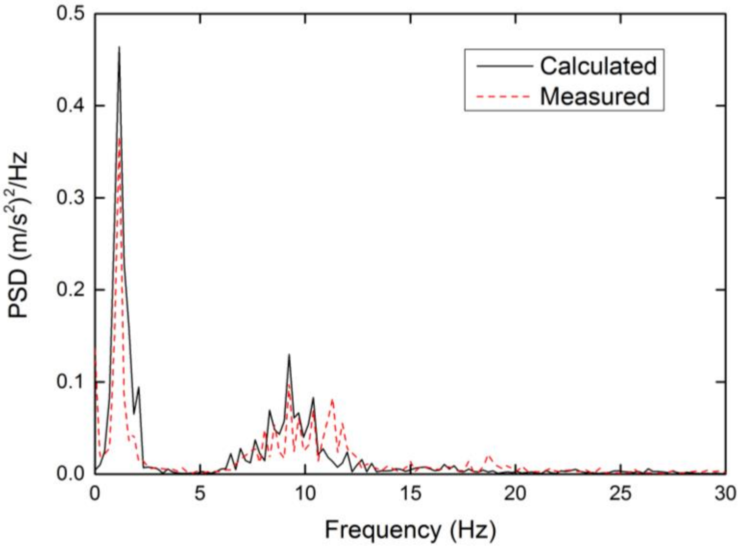

Figure 2 Comparison between calculated and measured acceleration PSDs (seat rail, driver side).

3. Sensitivity analysis

3.1 Method and design

The primary task of this sensitivity analysis is to identify the contributions of the mass and inertia of relevant rear differential unit (RDU) components to the vehicle seat vibration.

The RDU system consists of an input shaft, a rear differential gearbox (RDG), two CV joints, and two drive shafts (DS), as shown in Figure 3. The mass properties of the rotating components inside the RDU housing (such as input and output shafts, gears and bearings) were integrated with those of the rear differential housing at the centre of gravity (CoG) of the RDG assembly. This treatment may miss out high frequency vibration contents caused by gear coupling, oil film oscillation, and bearing misalignment, but contains the low frequency vibration excited by road (which is of importance to the ride analysis) and is beneficial to reducing computational costs (which is essential for a system level sensitivity analysis involving using a large scale vehicle model).

The design parameters considered in this analysis include the masses and the moments of inertial about three orthogonal (*x*-*y*-*z*) directions of the RDG, the CV joint, and the drive shaft (DS). In view of the symmetry of the geometry and layout, only one set of mass property (mass and moments of inertia about the axial (*y*) and radial (*x* and *z*) directions) of the drive shafts (left-hand side (LHS) or right-hand side (RHS)) are considered. Similar treatment was applied to the LHS and RHS CV joints. As a result, a total of 10 factors were considered in the sensitivity analysis, i.e.,

- M(RDG), I_x(RDG), I_y(RDG), I_z(RDG) - mass and moments of inertia about the longitudinal, lateral and vertical directions of the rear differential gearbox
- M(CVJ), I_x(CVJ), I_y(CVJ) – mass and moments of inertia about the radial (longitudinal) and axial (lateral and vertical) directions of the constant velocity joint
- M(DS), I_x(DS) and I_y(DS) - mass and moments of inertia about the radial (longitudinal) and axial (lateral and vertical) directions of the drive shaft

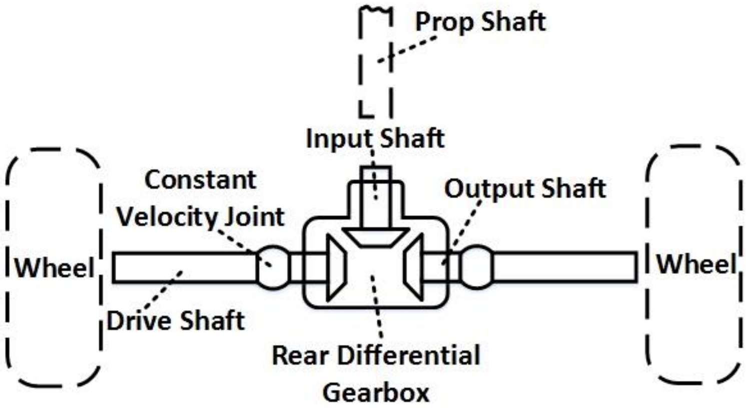

Figure 3 Schematics of the rear differential unit and adjacent components.

A DOE method was employed for the sensitivity analysis – a 10-factor and 2-level factorial analysis using MINITAB (version 17). The upper level and lower level values of each factor were calculated by increasing and decreasing 30% from its nominal value corresponding to the correlated baseline model.

The number of runs necessary for a 2-level full factorial design is 2^k where k is the number of factors. As the number of factors increases, the number of runs necessary to perform a full factorial design increases rapidly. A 2-level full factorial design with 10 factors requires 1024 runs which is computationally very expensive. To make the analysis more practical, a fractional factorial design analysis with 32 runs and resolution IV was adopted. Design resolution describes the extent to which effects in a fractional factorial design are aliased with other effects. The reduced number of runs was to some extent compromised by the reduced resolution. In the current study, however, the adopted method can still give satisfactory results as the main effects of the design factors which are of primary concern were not affected. With resolution IV, no main effects are aliased with any other main effect or 2-factor interactions.

3.2 Sensors and solver

Six sensors were defined at: the seat rail on the driver side, the seat rail on the front passenger side, the base of the rear left passenger seat, the base of the rear centre passenger seat, the base of the rear right passenger seat, and the bottom face of the RDU housing. While the vibration of the RDU housing was included as an extra check or a reference, accelerations at the seat rail or base of the front and rear seats are acquired as the measure for analysing the vehicle ride comfort.

During the simulation, a backward differentiation formula named SODASRT2 (an implicit multistep integration scheme with a tolerance of 0.0001) was used to solve the equations of motion of the multi-body system. The simulation duration was 25 s, the vehicle speed was 60 mph, and the sampling rate was 512 samples per second.

4. Results and discussion

4.1 Ride vibration from the baseline model simulation

The acceleration time histories and the corresponding PSDs at the seat rail on the driver side in the longitudinal, lateral and vertical directions were calculated based on simulations using the correlated baseline model of the car travelling on the B-road at 60 mph. Taking the accelerations at the seat rail on the driver side as an example (Figure 4), it can be seen that the acceleration in the vertical direction (0.63 m/s^2 r.m.s.) has the highest value among the three directions, whereas the acceleration in the lateral direction (0.09 m/s^2 r.m.s.) is the lowest. The vibration energy in the vertical direction centred at two frequency regions (1-2 Hz and 6-12 Hz) with a prominent peak at 1.25 Hz. The vibration energy in the longitudinal direction centred at the frequency range 9-15 Hz. The vibration in lateral direction is very low over the frequency range of 0-30 Hz. Similar characteristics were observed for the accelerations at the seat rail of the front passenger seat.

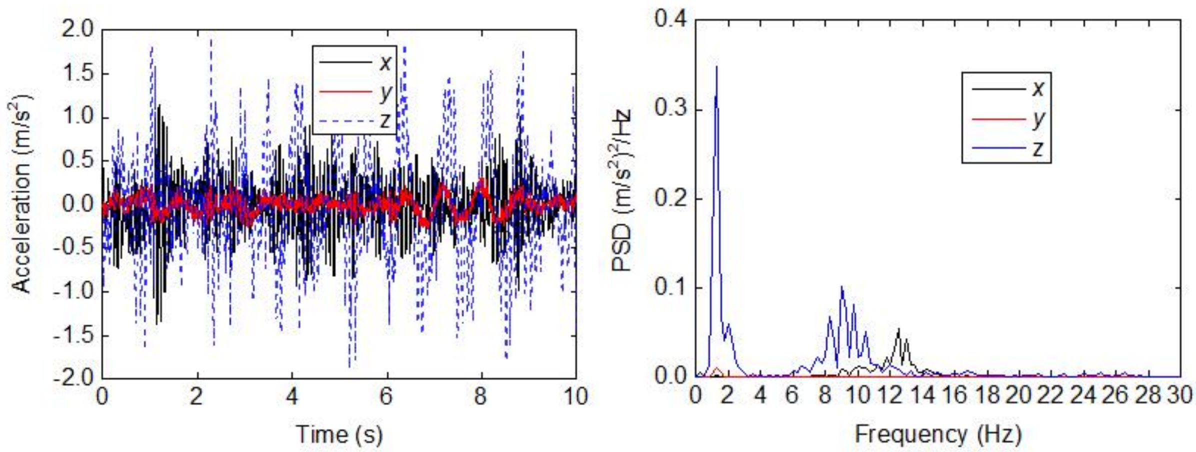

Figure 4 Accelerations and corresponding PSDs at seat rail on the driver side in three directions.

It is reasonable to believe that the vibration in the vertical direction contributes most to discomfort when the vehicle travels on the smooth B-road. The vibration energy in the longitudinal direction mainly distributed between 9 to 15 Hz, which along with the vertical vibration in the range of 8-12 Hz is likely associated with the powertrain and driveline excitation and should not be neglected. The vibration in the lateral direction is rather small and may be considered less important to discomfort compared to the vibration in the other two directions.

4.2 Responses of the factorial design

The acceleration r.m.s. values at the seat rails of the front seats and at the seat bases of the rear seat in three directions were calculated and used as the responses in the factorial design (Table 1).

Table 1 Acceleration r.m.s. values (m/s^2) at the RDU bottom, front seat rails and rear seat bases in different runs.

Run	RDU bottom			Driver			Front passenger			Left rear passenger			Centre rear passenger			Right rear passenger		
	x	*y*	*z*	*x*	*y*	*z*	*x*	*y*	*z*	*x*	*y*	*z*	*x*	*y*	*z*	*x*	*y*	*z*
1	1.66	0.38	1.36	0.34	0.10	0.64	0.33	0.10	0.54	0.33	0.11	0.67	0.33	0.11	0.62	0.33	0.11	0.59
2	1.71	0.38	1.35	0.34	0.11	0.65	0.33	0.11	0.54	0.33	0.11	0.67	0.33	0.11	0.62	0.33	0.11	0.59
3	1.66	0.44	1.45	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.34	0.11	0.61	0.33	0.11	0.59
4	1.68	0.43	1.53	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.67	0.33	0.11	0.62	0.33	0.11	0.59
5	1.69	0.39	1.38	0.35	0.11	0.65	0.33	0.11	0.54	0.34	0.12	0.67	0.33	0.12	0.62	0.33	0.12	0.59
6	1.66	0.45	1.44	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.34	0.11	0.61	0.34	0.11	0.59
7	1.67	0.43	1.51	0.34	0.11	0.65	0.33	0.11	0.54	0.34	0.11	0.66	0.33	0.11	0.62	0.33	0.11	0.59
8	1.68	0.40	1.49	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.33	0.11	0.62	0.33	0.11	0.59
9	1.74	0.43	1.56	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.67	0.33	0.11	0.62	0.33	0.11	0.59
10	1.66	0.39	1.34	0.34	0.11	0.65	0.33	0.11	0.54	0.33	0.11	0.67	0.33	0.11	0.62	0.33	0.11	0.59
11	1.72	0.42	1.55	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.33	0.11	0.62	0.33	0.11	0.59
12	1.71	0.38	1.38	0.34	0.11	0.65	0.33	0.11	0.54	0.33	0.11	0.67	0.33	0.11	0.62	0.32	0.11	0.59
13	1.72	0.42	1.52	0.34	0.11	0.65	0.33	0.11	0.54	0.33	0.11	0.67	0.33	0.11	0.62	0.32	0.11	0.59
14	1.65	0.44	1.41	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.34	0.11	0.62	0.34	0.11	0.59
15	1.71	0.41	1.52	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.33	0.11	0.62	0.33	0.11	0.59
16	1.60	0.41	1.29	0.34	0.11	0.65	0.33	0.11	0.54	0.34	0.11	0.66	0.33	0.11	0.61	0.33	0.11	0.59
17	1.63	0.40	1.29	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.33	0.11	0.61	0.33	0.11	0.59
18	1.65	0.41	1.32	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.34	0.11	0.61	0.33	0.11	0.59
19	1.68	0.42	1.41	0.35	0.11	0.65	0.35	0.11	0.54	0.35	0.11	0.66	0.34	0.11	0.61	0.34	0.11	0.59
20	1.66	0.42	1.41	0.36	0.11	0.65	0.35	0.11	0.54	0.35	0.11	0.66	0.34	0.11	0.61	0.34	0.11	0.59
21	1.75	0.42	1.59	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.33	0.11	0.62	0.33	0.11	0.59
22	1.66	0.42	1.32	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.34	0.11	0.61	0.33	0.11	0.59
23	1.63	0.41	1.31	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.33	0.11	0.61	0.33	0.11	0.59
24	1.61	0.39	1.29	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.34	0.11	0.62	0.33	0.11	0.59
25	1.65	0.37	1.33	0.34	0.11	0.65	0.33	0.11	0.54	0.33	0.11	0.67	0.32	0.11	0.62	0.32	0.11	0.59
26	1.63	0.41	1.30	0.35	0.11	0.65	0.34	0.11	0.54	0.34	0.11	0.66	0.33	0.11	0.61	0.33	0.11	0.59
27	1.72	0.39	1.39	0.34	0.11	0.65	0.33	0.11	0.54	0.33	0.11	0.66	0.33	0.11	0.62	0.32	0.11	0.59
28	1.69	0.37	1.35	0.34	0.11	0.65	0.33	0.11	0.54	0.33	0.11	0.67	0.33	0.11	0.62	0.32	0.11	0.59
29	1.71	0.44	1.46	0.36	0.11	0.65	0.35	0.11	0.54	0.35	0.11	0.66	0.34	0.11	0.61	0.34	0.11	0.59
30	1.69	0.45	1.45	0.35	0.11	0.65	0.34	0.11	0.54	0.35	0.11	0.66	0.34	0.11	0.61	0.34	0.11	0.59
31	1.69	0.45	1.43	0.36	0.11	0.65	0.35	0.11	0.54	0.35	0.11	0.66	0.34	0.11	0.61	0.34	0.11	0.59
32	1.69	0.45	1.43	0.36	0.11	0.65	0.34	0.11	0.54	0.35	0.11	0.66	0.34	0.11	0.61	0.34	0.11	0.59

4.3 Standardized effects

The degree of importance of the design parameters on ride comfort is calculated based on the data given in Table 1 along with the design chart (not shown) generated within MINITAB and displayed using Pareto charts as shown in Figure 5. In the Pareto charts, a bar extending past the reference limit (dashed line in the figure) means that the effect of corresponding factor on the acceleration response at a specific position and direction is statistically significant. It can be seen that the mass of the RDG is a significant influencing factor to the seat rail vibration on the driver side in the longitudinal and vertical directions. The moment of inertia of the CV joint about the longitudinal direction is another important factor to the seat rail vibration on the driver side in the longitudinal and lateral directions. As to the seat rail vibration on the front passenger side, the mass of the RDG and the moment of inertia of the CV joint about the longitudinal direction are the top two sensitive factors. For the vibration at the base of three rear seats, in addition to the above two factors, the mass of the drive shaft also affects the vibration significantly, especially in the lateral direction. The effect of the mass of CV joint on the lateral vibration at the base of the rear left passenger seat just exceeds the significant level. In terms of the vibration at the RDU housing, the number of the significant factors increases. In addition to the mass of RDG and the moment of inertia of the CV joint about the longitudinal direction, the moments of inertia of the RDG with respect to the longitudinal and lateral directions and the masses of the drive shaft and the CV joint have considerable influences as well. Besides, instead of the mass of the RDG, the moment of inertia of the CV joint about the longitudinal direction has the most significant effect on the vibration at the RDU housing.

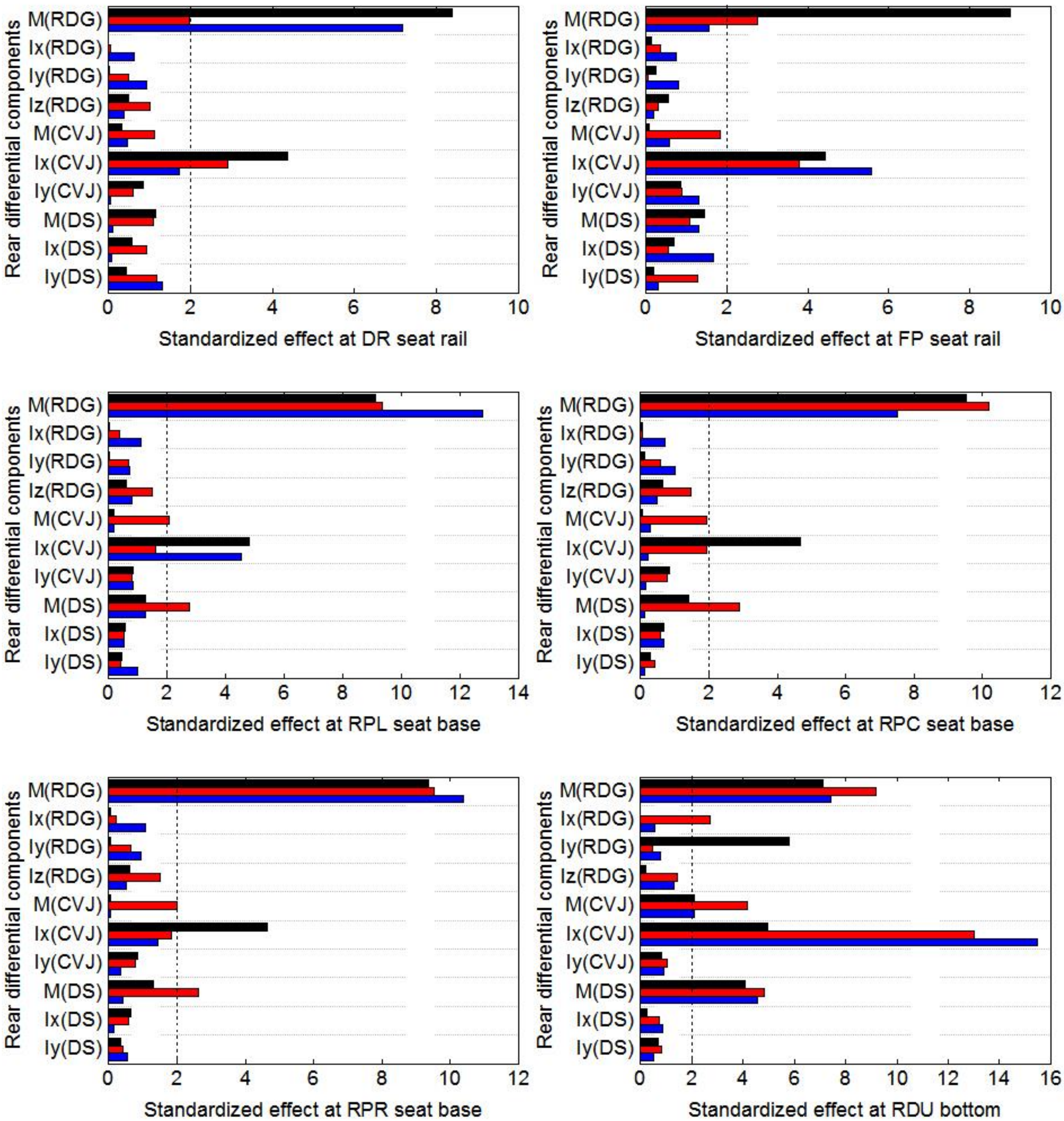

Figure 5 Standardized effect of design parameters on vehicle vibration at different positons and different directions (black: x direction, red: y direction, and blue: z direction).

Since the RDG is the heaviest part and its mass is far bigger than other components of the RDU considered in the sensitivity analysis, any changes in the mass of the RDG may introduce variations in the modal frequencies of the RDU system which can affect the vibration transmitted to the RDU bushes. Indeed, according to the sensitivity analysis, the mass of the RDG and the moment of inertia of the CV joint about the longitudinal direction are the two most important factors affecting the vibration at the seat rails (bases), and the latter is the most significant factor affecting the vibration of the RDG in the lateral and vertical directions.

In summary, there are six factors (mass properties of the RDU) having significant effect on the vibration at the seat rails and the RDU housing in different directions: the mass of RDG, the mass and the longitudinal moment of inertia of the CV joint, the mass of the drive shaft, and the moments of inertia of the RDG about the longitudinal and lateral directions. However, as discussed above, the acceleration in the lateral direction is much smaller comparing to the vibration in the vertical and longitudinal directions. Consequently, the vertical and longitudinal vibration at the seat rails and seat bases was further considered in the analysis of main effects below.

4.4 <u>Main effects</u>

Based on the calculated standardized effects of the design parameters on the seat rail vibration, the mass of the RDG, the moment of inertia of the CV joint about the longitudinal direction, and the mass of the drive shaft are the three most significant factors (in descending order) among the 10 factors examined. How the above three factors affect the ride vibration was further analysed and the results are shown in Figure 6. The vertical axis of this figure gives the change of the acceleration r.m.s. value caused by the change of a factor or parameter.

At the front seat rails, the vibration in the longitudinal direction decreases with decreasing the mass of the RDG and the moment of inertial of the CV joint about the longitudinal direction. The vibration in the vertical direction is decreased with decreasing the moment of inertia of the CV joint with respect to the longitudinal direction and with increasing the masses of the RDG and the drive shaft. Among these three factors, the mass of the RDG contributes most to the vibration at the front seat rails in the longitudinal direction and to the seat rail vibration on the driver side in the vertical direction. The moment of inertia of the CV joint about the longitudinal direction is the dominant factor affecting the vibration at the front passenger seat rail in the vertical direction.

At the rear seat base, the effect of the mass of the RDG and the moment of inertia of the CV joint with respect to the longitudinal direction on the vibration is similar in the longitudinal direction and in the vertical direction, respectively. Increase of the value of these two factors is beneficial to the vibration reduction in the vertical directions, but has adverse effects on the vibration in the longitudinal direction. Mass reduction of the drive shaft increases the vibration at the rear seat bases in the longitudinal direction but has very small effect on the vibration at the rear seat bases in the vertical direction.

Although mass reduction of the RDG increases the vibration at all seat rail (base) positions in the vertical direction, the absolute variation in the acceleration r.m.s. value (about 0.0006-0.0085 ms^{-2}) is rather small which may be hardly to be detected by drivers or passengers. In the longitudinal direction, mass reduction of the RDG reduces the vibration at all seat rail (base) positions, and the level of the vibration reduction in the longitudinal direction is about double of the level of the vibration increase in the vertical direction. Reduction of the moment of inertia of the CV joint about the longitudinal direction is generally beneficial to vibration reduction at the seat rail (base) in the longitudinal and vertical directions. Lightweight drive shaft has much less effect on ride vibration compared with the mass reduction of the RDG and the CV joint.

5. Conclusions

Ride analysis of a correlated baseline model of a vehicle travelling on a B-road shows that the vibration at seat rails in the lateral direction is much smaller than that in the longitudinal and vertical directions. The sensitivity analysis shows that the mass of RDG, the moment of inertia of the CV joint about the longitudinal direction, and the mass of the drive shaft are the top three factors (in descending order) affecting the vibration at seat rails or seat bases, which needs to be taken into consideration when designing a lightweight structure for the rear differential unit.

The effects of the design factors (mass properties of the RDU system) on the vibration at seat rails and bases vary in different locations and directions. In the longitudinal direction, reduction of the mass of RDG and the moment of inertia of the CV joint about the longitudinal direction is beneficial to reducing the vibration at seat rails and bases. The weight reduction of the drive shaft will slightly increase the vibration at seat rails (bases) but the amount is rather small. In the vertical direction, the trend of the mass reduction of the RDG and the drive shaft on the seat rail vibration is similar: decrease of the masses results in increase of seat rail vibration. Reduction of the moment of inertia of the CV joint with respect to the longitudinal direction is beneficial to reducing the vibration at the front seat rails in both longitudinal and vertical directions and at the rear seat bases in the longitudinal direction but has adverse effect on the vibration at the rear seat bases in the vertical direction.

6. Acknowledgement

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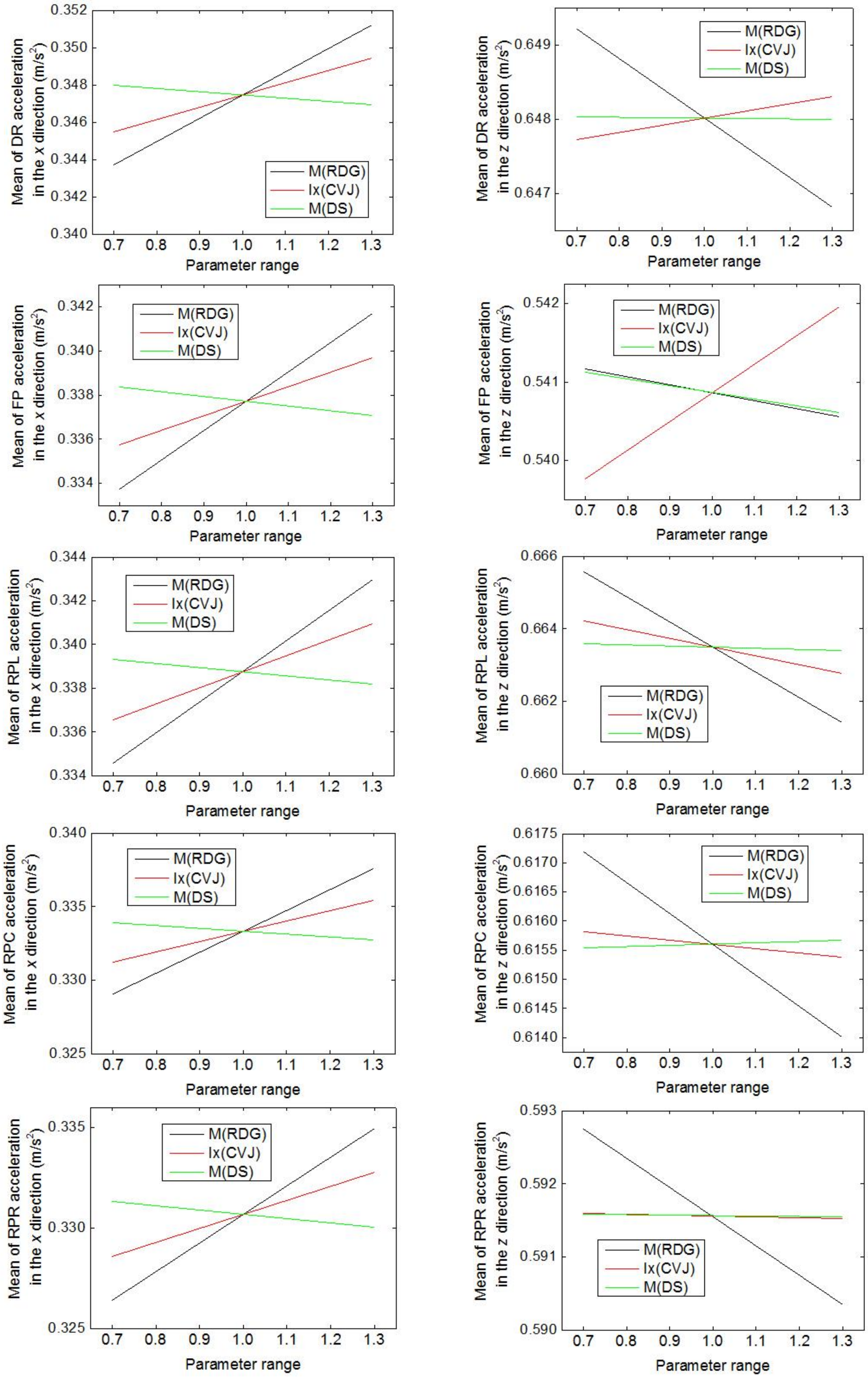

Figure 6 Main effects of the significant parameters on the seat rail or base vibration in the longitudinal and vertical directions.

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