The Study on Accurate Modeling of Suspension Based on ADAMS

Li Xueying, Yu Zhuoping, and Xiong Lu

Abstract—**The torque vibration derived from in-wheel-motor transmits to body frame through suspension system without the absorption of mechanical transmission parts, which influenced the quality of the vehicle NVH. This paper aims to build an accurate suspension system model to analyze the vibration transmission property. A multi-rigid suspension model and a multi-flexible suspension model had been established respectively. The vibration characteristics of two models were simulated, furthermore the swept-sine exciting vertical force signal on wheel contact point were input on the simulation models to find the difference between rigid and flexible model. The simulation results show that: the multi-flexible model can more accurately reflect the vibration characteristics of the suspension system in the high frequency range, hence more applicable to the simulation analysis of in-wheel-motor electric vehicle suspension system vibration characteristics. Then the rubber bushing model was replaced with new empirical rubber bushing model, the inherent frequency and the frequency response functions were compared. The results show: The multi-flexible suspension model with new empirical rubber bushing model hasn't notable influence to inherent frequency. However, it can reflect more peak values of frequency response functions and the transmissibility at every peak frequency are higher than the original multi-flexible suspension model.**

Index Terms—**Multi-flexible suspension model, multi-rigid suspension model, new empirical model of rubber bushing, vibration characteristics.**

I. INTRODUCTION

The four in-wheel-motor driven electric vehicle type simplifies powertrain and vehicle system structure, greatly increases the performance of the electric vehicle. Meanwhile the torque vibration derived from in-wheel-motor transmits to bodyframe through suspension system without the absorbtion of mechanical transmission parts, then excites body panel to shape the vehicle interior noise, which influences the quality of the vehicle NVH.

However, the researches in domestic and outside rarely involved the vibration characteristics of suspension system in four in-wheel-motor driven electric vehicle, where the major source of interior noise is the vibration derived from wheel-hub motor. So it is important to build the suspension system simulation model including tire, suspension rods and rubber bushing components for analyzing the influence of suspension parts parameters to system vibration transmission property. It is helpful to effectively restrain the vibration arised from in-wheel-motor and reduce the vehicle interior noise.

In this paper, through accurate modeling and simulation, the difference between rigid and flexible model in the way of analysing vibration characteristics has been presented.

Manuscript received March 21, 2012; revised March 31, 2012. The Science and Technology Commission of Shanghai Municipality is truly appreciated for its big support. The project number is 10ZR1432400.

Authors are with the Automobile Engineering Department, University of Tongji, Shanghai, China (e-mail: sherry_lixueying@hotmail.com; yuzhuoping@mail.tongji.edu.cn; xiong_lu@mail.tongji.edu.cn).

II. MODELING OF MULTI-RIGID SUSPENSION SYSTEM

The independent dual lateral arm suspension researched in this paper is based on a four in-wheel-motor driven electric vehicle, and consist of upper control arm, lower control arm, knuckle, shock absorber assembly, wheel and tie rod.

Reasonable hypothesis and simplication is made as follows befor modeling:

1) Assume all of the components are rigid, i.e. ignore elasticity of them.
2) The model take the damping nonlinearity of shock absorber into consideration.
3) Ignore the influence of powertrain on vibration of suspension, only consider the influence of road irregularity.

Then, transfer the accurate 3D model into ADAMS and add the relative constraints.the final multi-rigid model of suspension system is shown in Fig.1.

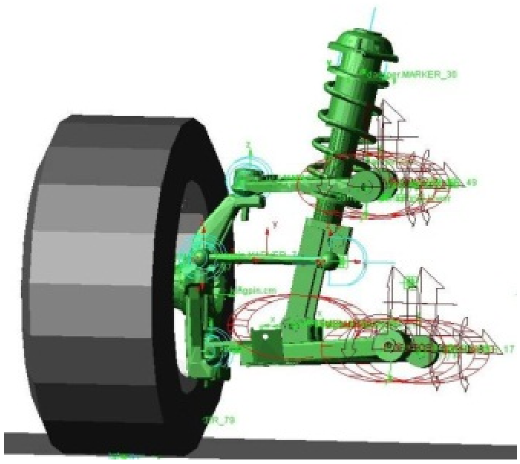


Fig. 1. Multi-rigid suspension system model

The kinematic model consist of 7 translational components, 3 spherical joints, 2 revolute joints, 1cylindrical joint and 3 universal joints. In elastodynamics model, 2 revolute joints at the end of two suspension arms are replaced by 4 rubber bushing and 1 universal joint at the lower end of shock absorber is replaced by 2 rubber bushing. The parameters of coil spring and shock absorber are acquired from test. Front wheel alignment parameters are provided by supplier. The model of wheel is rigid.

III. Modeling of Multi- Flexible Suspension System

Through modeling, experimental modal analysis verification, and modification, the finite element model could be turned into flexible model. Meanwhile the constraints, parameters of the rubber bushing, coil spring and the damping parameters of shock absorber are all the same with the multi-rigid suspension model.

After material definition and meshing in Hypermesh, the first six orders modal frequencies and mode shapes of upper control arm, lower control arm, and shock absorber can be acquired through OptiStruct solver.

Then modal experiment is done using LMS to acquire the first five orders modal frequencies of suspension components. The process of data collection and analysis is strictly controlled to ensure the accuracy and reliability of the experimental results.

Finally, pick out experimental modal frequency and simulation calculated modal frequency at the same mode shape and do the error calculation. From Table I to Table IV show comparison results separaterly on upper control arm, lower control arm, knuckle, shock absorber.

TABLE I: Error Calculation on Upper Control Arm

order		Experimental modal frequency(Hz)	Simulation calculated modal frequency(Hz)	error
1		178.585	169.7	-4.98%
2	Upper	829.937	823.9	-0.73%
3	control	1414.21	1505.4	6.45%
4	arm	1688.82	1672.6	-0.96%
5		2623.74	2658.6	1.33%

TABLE II: Error Calculation on Lower Control Arm

order		Experimental modal frequency(Hz)	Simulation calculated modal frequency(Hz)	error
1		568.2	540.4	-4.89%
2	Lower	949.7	935.4	-1.51%
3	control	1141.8	1160	1.59%
4	arm	1632.6	1570	-3.83%
5		1733.7	1668	-3.79%

TABLE III: Error Calculation on Knuckle

order		Experimental modal frequency(Hz)	Simulation calculated modal frequency(Hz)	error
1		1474.9	1524.5	3.36%
2		2219.3	2408.1	8.51%
3	knuckle	2484.4	2558.0	2.96%
4		2922.2	2873.3	-1.67%
5		3378.3	3758.4	11.25%

TABLE IV: Error Calculation on Shock Absorber

order		Experimental modal frequency(Hz)	Simulation calculated modal frequency(Hz)	error
1	Shock	469.9	454.3	-3.32%
2	absorber	1065.0	1099	3.19%

The above data shows the multi-flexible model of the suspension components is mainly accurate. However, according to these error calculations further adjustments to the models are done.

All the error calculation results help modification to build a accurate multi-flexible suspension model, which is shown in Fig. 2.

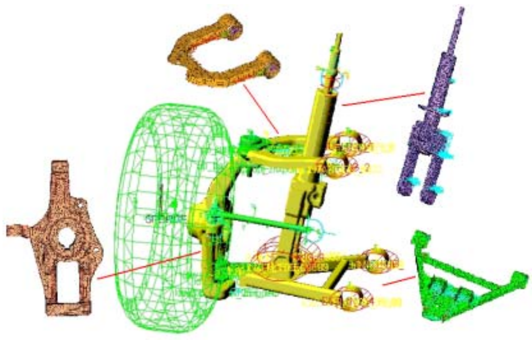
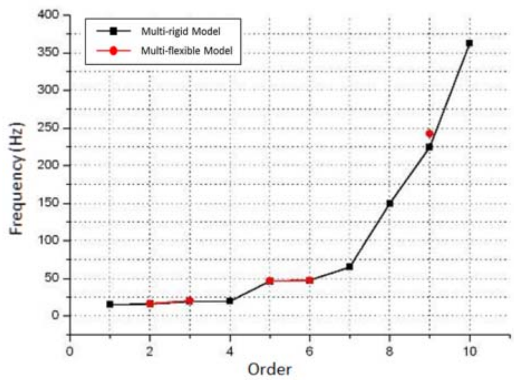


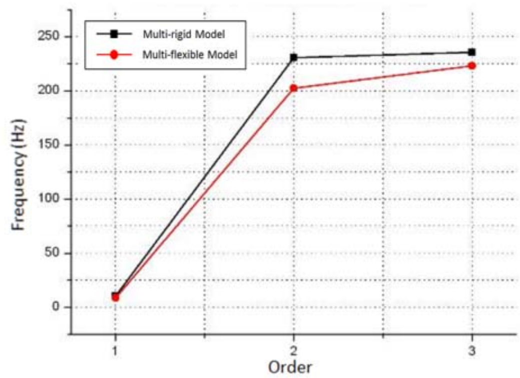
Fig. 2. Multi-flexible suspension model

IV. Comparison of Multi-Rigid and Multi-Flexible Model Based on Simulation Analysis of Vibration Characteristics

The inherent characteristics for multi-rigid and multi-flexible suspension model are simulated and calculated. The results are compared, as shown in Fig. 5.



(a) Comparison of real number modal frequency



(b) Comparison of complex number modal frequency

Fig. 3. Comparison of the inherent characteristics of multi-rigid and multi-flexible suspension model

As displayed in Fig. 3, under the same mode shape, the real number modal frequency of multi-rigid and multi-flexible model are similar. But the first 3 orders complex number modal frequency of multi-flexible model are all lower than multi-rigid model, which agrees with the theory that the modal frequency of flexible-body shall be lower than that of rigid-body because of its lower rigidity.

Table V shows that simulation of multi-flexible suspension model can present more inherent frequency characteristics than multi-rigid model. Particularly within 300Hz, the mode mainly reflects the deformation of lower control arm.

TABLE V: INCREMENTAL INHERENT FREQUENCY OF MULTI-FLEXIBLE MODEL

Inherent frequency(Hz)	Description of Mode shape
227.91	deformation of upper control arm
262.72	
90.77	deformation of lower control arm
102.81	
187.31	
227.91	

Based on multi-rigid and multi-flexible suspension model, the swept-sine test signal is vertically added to the wheel-ground contact point.Through simulation the frequency response functions (FRF) of different positions are acquired, the comparison results are shown in Fig. 4 to Fig. 8.

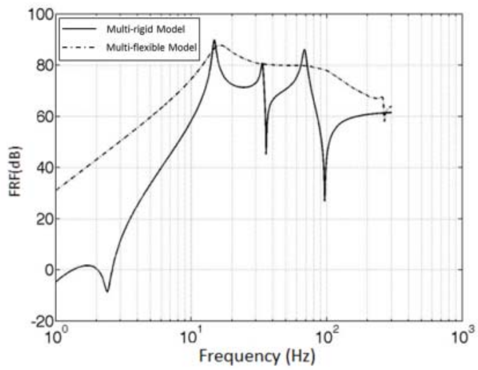


Fig. 4. FRF from wheel-ground contact point to the body-shock absorber connection point

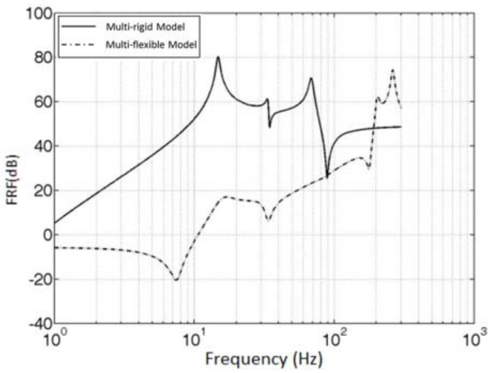


Fig. 5. FRF from wheel-ground contact point to rubber bushing1

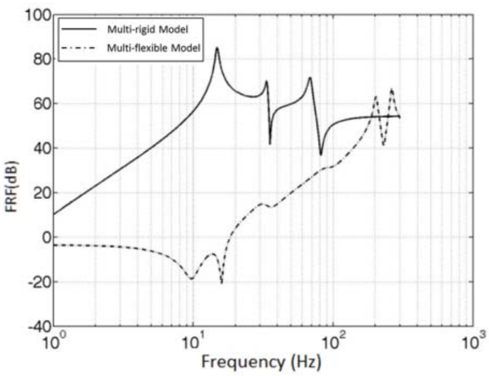


Fig. 6. FRF from wheel-ground contact point to rubber bushing 2

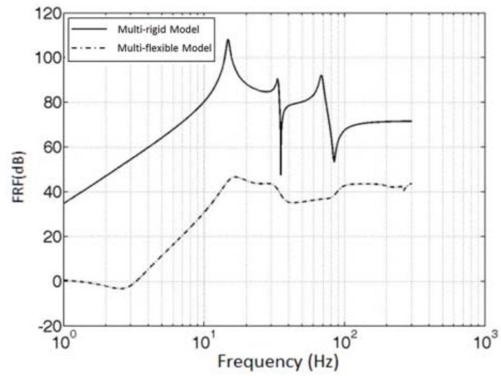


Fig. 7. FRF from wheel-ground contact point to rubber bushing 3

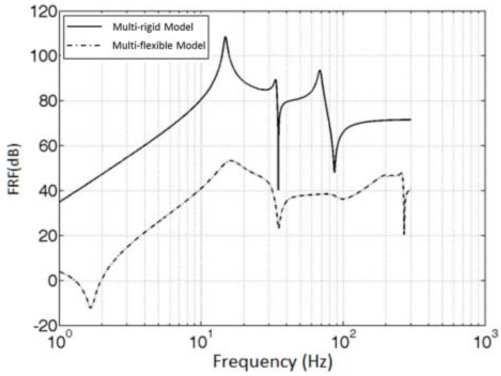


Fig. 8. FRF from wheel-ground contact point to rubber bushing 4

It is observed through Fig. 4 to Fig. 8 that:

Except for the body-shock absorber connection point, the frequency response characteristics at the connection points between all the rubber bushings and body in the multi-flexible suspension system model is less than the relative results in multi-rigid suspension system model.

By adding 0-300Hz swept-sine test signal vertically on the wheel-ground contact point, the peak frequencies of the multi-rigid suspension system model turn out to be mainly limited within 70 Hz, meanwhile most of peak frequencies of the multi-flexible suspension system model distribute in the range higher than 100Hz, which means the multi-flexible suspension system model can more accurately reflect the vibration characteristics in medium and high frequency range.

V. INFLUENCE OF THE RUBBER BUSHING MODEL TO VIBRATION CHARACTERISTICS OF SUSPENSION

A new empirical model of rubber bushing [1], [2] was raised by Yu Zengliang and others, which is shown in Fig. 9.

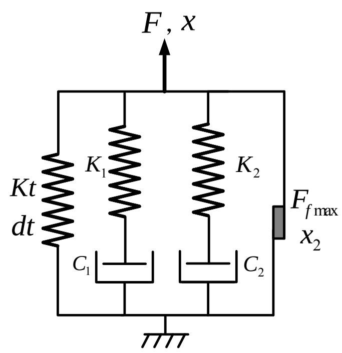


Fig. 9. Sketch map for new empirical model of rubber bushing

This model can preferably reflect the change law of frequence-and-amplitude dependent dynamic stiffness compared to the others.

Replace the rubber bushing model of the above multi-flexible suspension model with this model, then calculates and compare the inherent frequency.

TABLE VI: COMPARISON OF THE INHERENT CHARACTERISTICS OF SUSPENSION MODEL WITH ORIGINAL AND NEW RUBBER BUSHING MODEL

Normal Mode	Inherent frequency (Hz)		Complex mode	Inherent frequency (Hz)	
	Original model	New model		Original model	New model
1	16.05	15.98	1	15.02	15.15
2	22.02	21.90	2	31.53	32.19
3	23.49		3	81.73	83.13
4	32.47	31.59	4	91.44	98.53
5	45.67	45.79	5	209.05	196.21
6	47.40	47.27	6	204.38	201.24
7	47.89	47.87	7	291.10	248.07
8	49.47		8	261.07	258.12
9	62.75	52.72	9		232.73
10	95.67	95.82			
11	150.91				
12	166.29				
13	177.22				
14	235.30				
15	241.97				
16	250.20	251.46			
17	260.65	261.47			

Compared to original suspension model, the model with new empirical rubber bushing model has less normal mode and equal complex mode within 300Hz, the inherent frequencies are very much alike. Therefore we can see, that the new model hasn't notable influence to inherent frequency.

Separately with the original and the new multi-flexible suspension model, swept-sine test signal is vertically added to the wheel-ground contact point. Through simulation the frequency response functions (FRF) of different positions are acquired, the comparison results are shown in Fig. 10 to Fig. 14.

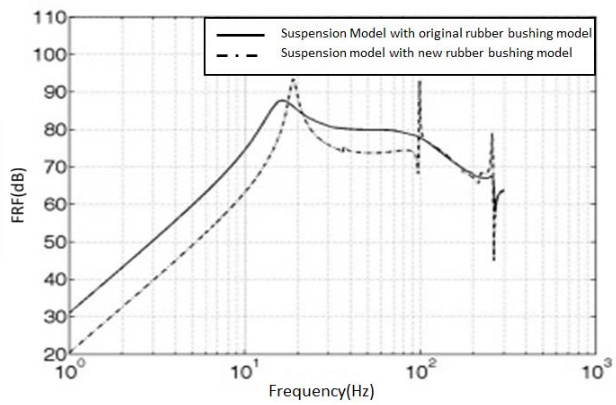


Fig. 10. FRF of body-shock absorber contact point

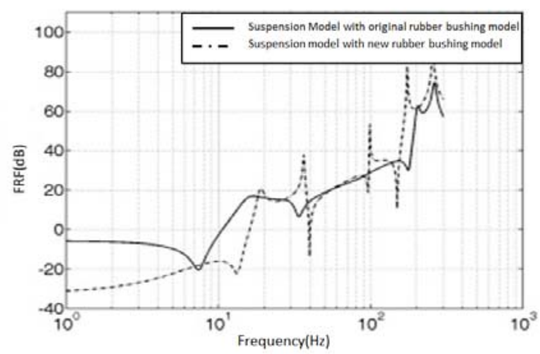


Fig. 11. FRF of rubber bushing 1-body contact point

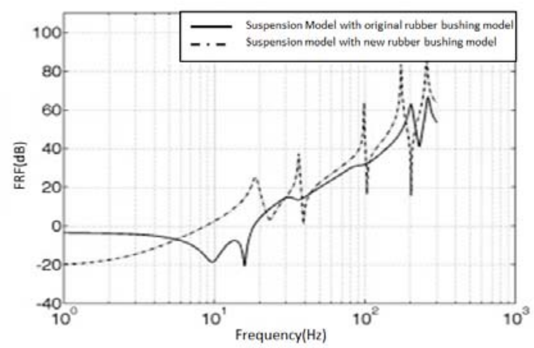


Fig. 12. FRF of rubber bushing 2-body contact point

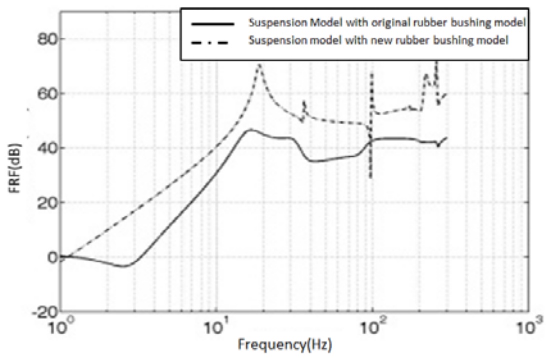


Fig. 13. FRF of rubber bushing 3-body contact point

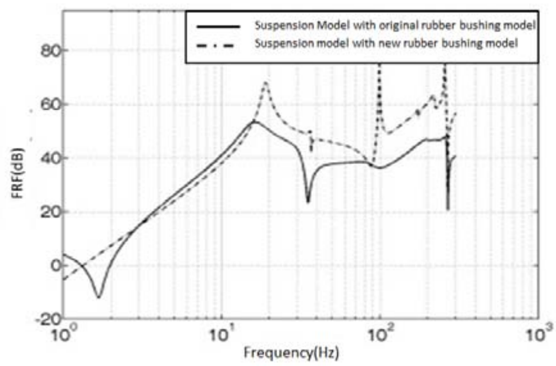


Fig. 14. FRF of rubber bushing 3-body contact point

The above comparation results show that, the multi-flexible suspension model with new empirical rubber bushing model can reflect more peak values of frequency response functions and the transmissibilities at every peak frequence are higher than the original multi-flexible suspension model.

VI. SUMMARIES

In this paper, based on the precise geometrical model of suspension system, accurate multi-rigid and multi-flexible dynamics simulation model for the suspension system are builded, the vibration characteristics are researched by both calculated and experimental way. The comparative results are:

Except for the body-shock absorber connection point, the frequency response characteristics at the connection points between all the rubber bushings and body in the multi-flexible suspension system model is less than the relative results in multi-rigid suspension system model.

By adding 0-300Hz swept-sine test signal vertically on the wheel-ground contact point, the peak frequencies of the multi-rigid suspension system model turn out to be mainly limited within 70 Hz, meanwhile most of peak frequencies of the multi-flexible suspension system model distribute in the range higher than 100Hz, which means the multi-flexible suspension system model can more accurately reflect the vibration characteristics in medium and high frequency range and is more proper to simulation analysis for vibration characteristics of the suspension system in the in-wheel-motor driven electric vehicle.

The multi-flexible suspension model with new empirical rubber bushing model hasn’t notable influence to inherent frequency. However,it can reflect more peak values of frequency response functions and the transmissibilities at every peak frequence are higher than the original multi-flexible suspension model.

VII. APPENDIX THE MODAL EXPERIMENT OF SUSPENSION COMPONENTS

A. Experimental Measurement And Analysis System

Experimental measurement and analysis system composed of three parts: the experimental excitation system response-acquisition system, modal analysis and processing system:

1) experimental excitation system including the hammer excitation;
2) response-acquisition system consists of acceleration sensors, force sensors, and LMS SCADAS III SC316W signal amplification and intelligent acquisition system;
3) modal analysis and processing system is mainly LMS modal analysis software Test.lab. The specific composition is shown in Fig. 15 and Fig. 16.

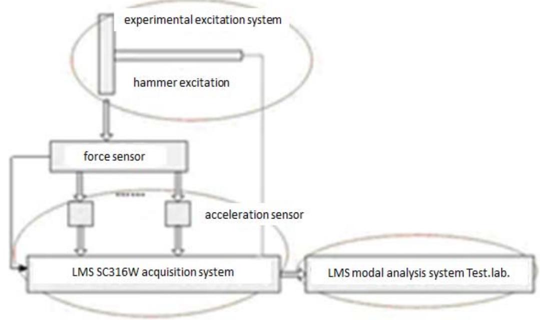


Fig. 15. Connection relationship chart of Experimental measurement and analysis system

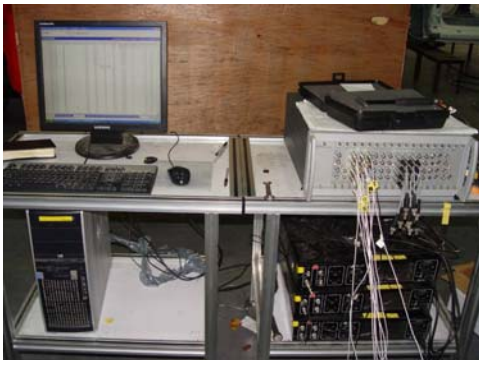


Fig. 16. Experimental measurement and analysis system

B. Experimental Scheme

1) Excitation way

The excitation way used is single-point excitation and multi-point pick-up method, which is proceeded in the Z direction of test specimen according to the coordinate. We set excitation to be impulse signal, the signal acquisition frequency to be twice of the frequency we concern, the times of signal acquisition to be 5 to 6.

2) Sensor layout

The acceleration sensors are arranged at the measuring points. The PCB accelerometers are selected. This kind of sensor has little mass and volume with no need of the magnet installation device. By using it can greatly reduce the impact caused by the additional mass and finally improved test accuracy. The mass of each sensor is only 6g.

3) Measuring points layout

The signal obtained at measuring points requires high signal to noise ratio, therefore the measuring points should not set close to the node. Meanwhile the measuring points should try to reflect the geometry of the measured object for easier observe of mode. Measuring points arrangement according above rules are shown in Fig. 17 to Fig. 20:

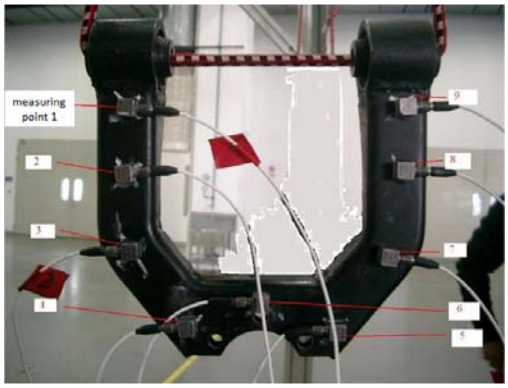


Fig. 17. Measuring points at upper control arm

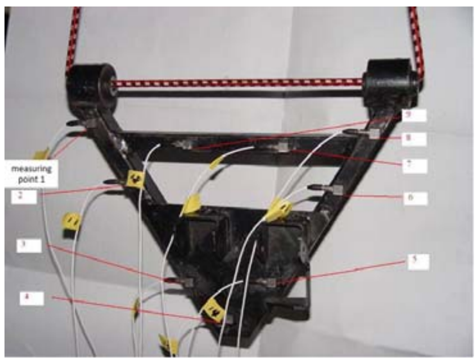


Fig. 18. Measuring points at lower control arm

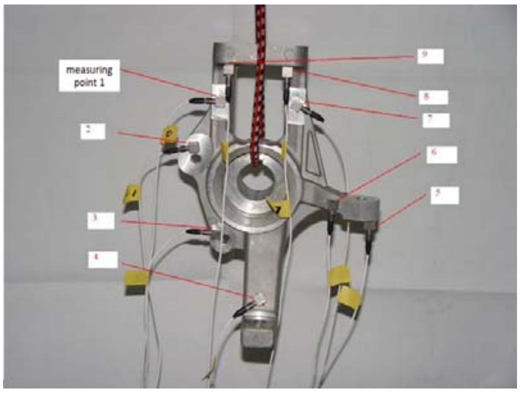


Fig. 19. Measuring points at knuckle

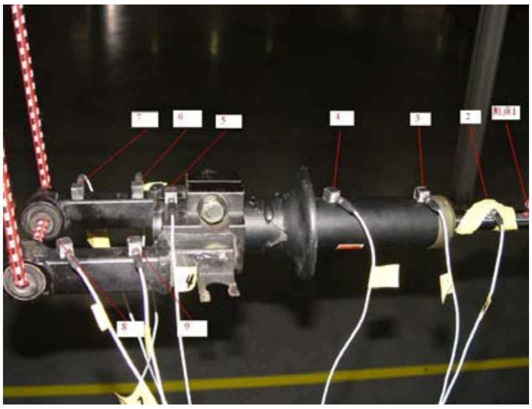


Fig. 20. Measuring points at shock absorber

4) Data processing and mode analysis

By applying a hammer force in z direction at the excitation points, measure accelerations of all points in z direction and calculate frequency response functions from every measuring point to excitation point. Ensemble averaging of FRF is utilized to determine the orders of mode, in which the modal parameters have been normalize

REFERENCES

[1] Yu Zengliang, Zhang Lijun and Yu Zhuoping, “Semi-Empirical Parameterized Dynamic Model of Rubber Bushing Mechanical Properties”, *Journal of Mechanical Engineering*, vol. 46(14), pp. 115-123, 2010.

[2] Yu Zengliang, Zhang Lijun and Luo Ying, “A Novel Semi-Empirical Dynamic Model of Rubber Bushing”, *Automobile Technology*, pp. 6-11, Aug 2010.

[3] Ichiro Kido, Akeru Nakamura, Takeshi Hayashi and Makoto Asai, “Suspension Vibration Analysis for Road Noise Using Finite Element Model”, *SAE Technical Paper*, 1999-01-1788, 1999.

[4] Hu Hongxing, “Experimental Validation of a Half-Vehicle Suspension Model”, *SAE Technical Paper*, 931966, 1993.

[5] A. Berry, H. Douville and P. Masson, “On-resonance transmissibility methodology for quantifying the structure-borne road noise of an automotive suspension assembly”, *Applied Acoustics*, vol. 67, pp. 358-382, 2006.

[6] Hisanori Tachibana, Kazuhiko Gotoh and Hitomi Sakab, “Vibration Analysis of Driveline and Suspension Using Finite Element Models”, *SAE Technical Paper*, 931306, 1993.

[7] Cai Zhanglin, Song Chuanxue, An Xiaojuan, Gao Hongling and Li Jianhua, “Virtual simulation research on vehicle ride comfort”, *SAE Technical Paper*, 2006-01-3499, 2006.

[8] Gi-Ho Lee, Jong-Hoon Lim and Gi-Tae Kim, “Improving ride quality on the cab suspension of a heavy duty truck”, *SAE Technical Paper*, 962151, 1996.

[9] Odilon T. Perseguim, A. Costa Neto, Marcos A. Argentino, C. F. Nogueira, P. Federico Neto, M. A. Fogaça, A. Dybal and M. D. Torre Garcia, “Comfort and Vibration Study of a Tractor and Trailer Combination Using Simulation and Experimental Spproaches: the Jumping Ride Behavior”, *SAE Technical Paper*, 2000-01-3517, 2000.

[10] Michel Constant, Jan Leyssens and Filip Penne, “Tire and Car Contribution and Interaction to Low Frequency Interior Noise”, *SAE Technical Paper*, 2001-01-1528, 2001.

[11] Derek K. Ward, Oliver Hanke, Torsten Bertram, Manfred Hiller and Roberto Bardini, “A generic suspension model for middle class passenger vehicles”, *SAE Technical Paper*, 2001-01-1280, 2001.

[12] A. Eskandari, O. Mirzadeh and Sh. Azadi, “Opimization of a McPherson Suspension System Using the Design of Experiments Methods”, *SAE Technical Paper* , 2006-01-1953, 2006.

[13] Yoichiro Kato, Yoshiro Tateishi and Naoto Ogawa, “Development of a Multi-link Beam Rear Suspension for Front-Wheel-Drive Cars”, *SAE Technical Paper* , 950585, 1995.

[14] Yang Xiaobo, Sudhakar Medepalli, “Sensitivities of Suspension Bushings on Vehicle Impact Harshness Performances”, *SAE Technical Paper* , 2005-01-0827,2005.

[15] Yang Xiaobo, Zhang Dajun, Sudhakar Medepalli and Mohammed Malik, “Suspension Tuning Parameters Affecting Impact Harshness Performance Evaluation”, *SAE Technical Paper* , SAE2006-01-0991.

Li Xueying was born in Beijing, China, in 1987. She received the B.E. degree in mechanical engineering from Tongji University, Shanghai, China, in 2009. She is currently working toward the M.E. degree in the Department of Automotive engineering , Tongji University, shanghai, China.

Her area of research is dynamic characteristics of vehicle.

Yu Zhuoping was born in Nan Chang, China, in 1960. He received the Ph.D. degree in automotive engineering from Tsing Hua University, Beijing, China,in 1996.

He once worked in Braunschweig automotive research institute, research department of Volkswagen automotive company, and automotive institute of Technical University Darmstadt. Since 1985 he has been working as a professor in Tongji University, he is the assistant of the president of Tongji University and dean of school of automotive. His professional field focuses on research and teaching of automotive engineering.

He is the leader of “Chang Jiang Scholars and Innovation Team of Energy-saving and Environment-friendly”. He is a member of science and technology commission of ministry of education, member of overall deployment experts of the Energy Efficient & New Energy Vehicles organized by MOST, member of experts of admittance committee of new energy vehicle, vice president of executive council of SOE-China, administrative vice dean and secretary general of intelligent transportation group of SOE-China, vice president of executive council of China auto talents society, vice president of executive council of SOE-Shanghai, dean member of automotive industry senior academic title assessment committee, etc. In 2005, fuel cell passenger car got the top 10 technical progresses of high-education institute in china. In 2008, he got the first grade of Science and Technology Progress of China. In 2009, he got the first co-production and research Innovation Award in China. He took participate in edition or translation of total 6 books, and published some 130 technical paper in periodicals or meeting seminars.

Xiong Lu was born in China, in 1978. He received the Ph.D. degree in automotive engineering from Tongji University, Shanghai, China, in 2005.

Since 2005 he has been working as an associate professor in Tongji University. From 2006 to 2008 was he associate director of ThyssenKrupp advanced vehicle technology laboratory, responsible for the development of a new generation of electric vehicles, vehicle platforms; also responsible for suspension matching, road test assessment and development of micro fuel cell car platform in the "863" project "research and development on dynamic system of fuel cell vehicle”. From 2008 to 2009, sponsored by China Scholarship Council, he went to the University of Stuttgart, Germany, to do postdoctoral research, cooperated with Daimler AG. His research area is the characteristics and control of dynamic system of vehicle and integration technology of new energy vehicle.

He is invited as peer-reviewed expert of the National Natural Science Foundation, accreditation experts of academic journals,” the Vehicle System Dynamics ".