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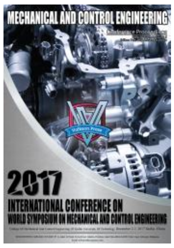

INFLUENCE OF SHOCK ABSORBER DEGRADATION ON VEHICLE NVH PERFORMANCE

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ABSTRACT

Damping characteristics degrade unavoidably in shock absorber's useful life. To study the effect of shock absorber degradation on NVH (Noise, Vibration and Harshness) performance of vehicles served in long term, double-action durability test was carried out on numerous shock absorbers. Afterwards, indicator diagrams and velocity characteristics of shock absorbers were obtained; damping coefficients were calculated at the pre-valve and post-valve opening. Based on the experimental result, a vehicle virtual prototype model was built by ADAMS/Car Ride. Then, vertical driver seat accelerations were measured, and weighted root-mean-square (RMS) accelerations were calculated at driver seat. In addition, influences of individual shock absorber degradation on vehicle high mileage performance were analyzed. The result shows that damping force degrades after durability test; the reduction rate of the weighted RMS acceleration is about 4.1%, while the increment rate is more than 7.5% at high mileage; the front-left shock absorber degradation causes a significantly worse vehicle NVH performance among the 4 individual one.

1. Introduction

Vehicles under the regular service condition inevitably experience fatigue damage and aging, resulting in decreasing performance couple with increasing malfunction. Vehicle NVH performance is one of the most concerned issues by the full vehicle and part manufacturing enterprises among the international automobile industry. The vehicle body joint degradation powertrain mount degradation, and even seat belt retractor degradation affects vehicle NVH performance [1-3]. Shock absorbers work as a primary functional component in the suspension to attenuate the vibration transmitted to the vehicle body, to enhance ride quality and comfort [4]. Therefore, shock absorber degradation has the most direct and palpable influences on vehicle NVH performance.

The shock absorbers have been designed and manufactured with energy absorption and liberation ability to guarantee good NVH performance of new or low-mileage-in-service vehicles. However, the damping force of the shock absorber declines with the duration of service [5]. Dong proposed a robust design method for shock absorbers to provide sufficient damping force under different temperature as a result of lower sprung mass acceleration, but the damping force degradation was not considered with the increasing useful life [6]. A group of scientists researched the effects of vehicle vibration parameters degradation on thermal load of shock absorbers, and the result showed that heat flux of the shock absorbers degraded after a long-term service [7]. Some of researcher investigated degradation of simultaneous spring force, shock absorber force couple with tire force on vibration aspect of vehicle comfort, but the influences of separate shock absorber damping force degradation were neglected [5]. A researcher presented a finite element model to study vehicle long term NVH performance degradation caused by high-mileage shock absorber bushings, but the weighted mean square root of accelerations at the seat were not measured [8]. Wei obtained defective indicator diagrams of shock absorbers based on a full vehicle durability test, but variation of the damping coefficients was not mentioned [9]. A group of researchers also investigated the effect of shock absorber strut insulators aging on vehicle noise and found the NVH performance degraded with the rubber permanent deformation, but the influence of individual strut insulator was not discussed [10].

In the present paper, durability test was carried out on numerous shock absorbers. Afterwards, degradation data of damping force were obtained, based on which ride quality analysis of a full vehicle was achieved to investigate the effects of the shock absorbers degradation on vehicle NVH performance.

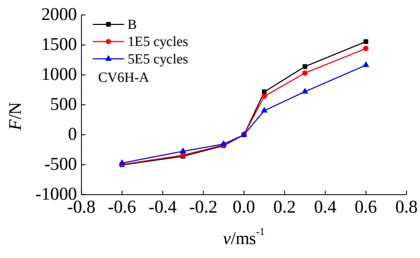
2. DURABILITY TEST FOR SHOCK ABSORBERS

NVH performance degradation can be estimated by fatigue analysis on the basis of laboratory test [11]. In this section, durability tests are conducted with several types of shock absorber samples; subsequently, specific experimental process and results were represented for the type CV6H-A.

2.1 General test

Although there are a variety of external factors and conditions when the shock absorber serves as a component of vehicle suspension, most of the conditions have little effects on the degradation. Therefore, the most dominant and obvious conditions were considered during the laboratory durability test. In addition, the durability test was conducted at the velocity of 0.1, 0.3 and 0.6 (1.2) m/s, respectively, to obtain the velocity characteristics of the shock absorbers. The test was conducted with several types of hydraulic shock absorbers served in Chinese passenger cars. The samples were randomly drawn from qualified products produce in batches.

According to the experimental data, the velocity characteristics were obtained for individual type of hydraulic shock absorber, as shown in Figure 1. It displays that the damping property degrades to some degree after numerous durability test cycles.



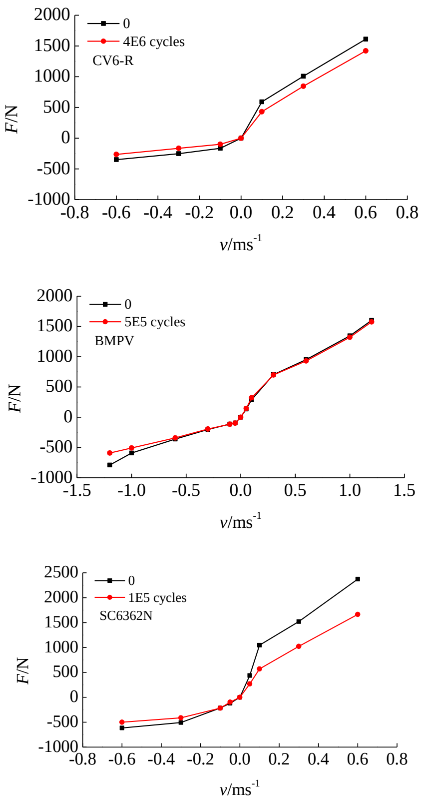


Figure 1: Velocity characteristic

2.2 Test and results for CV6H-A

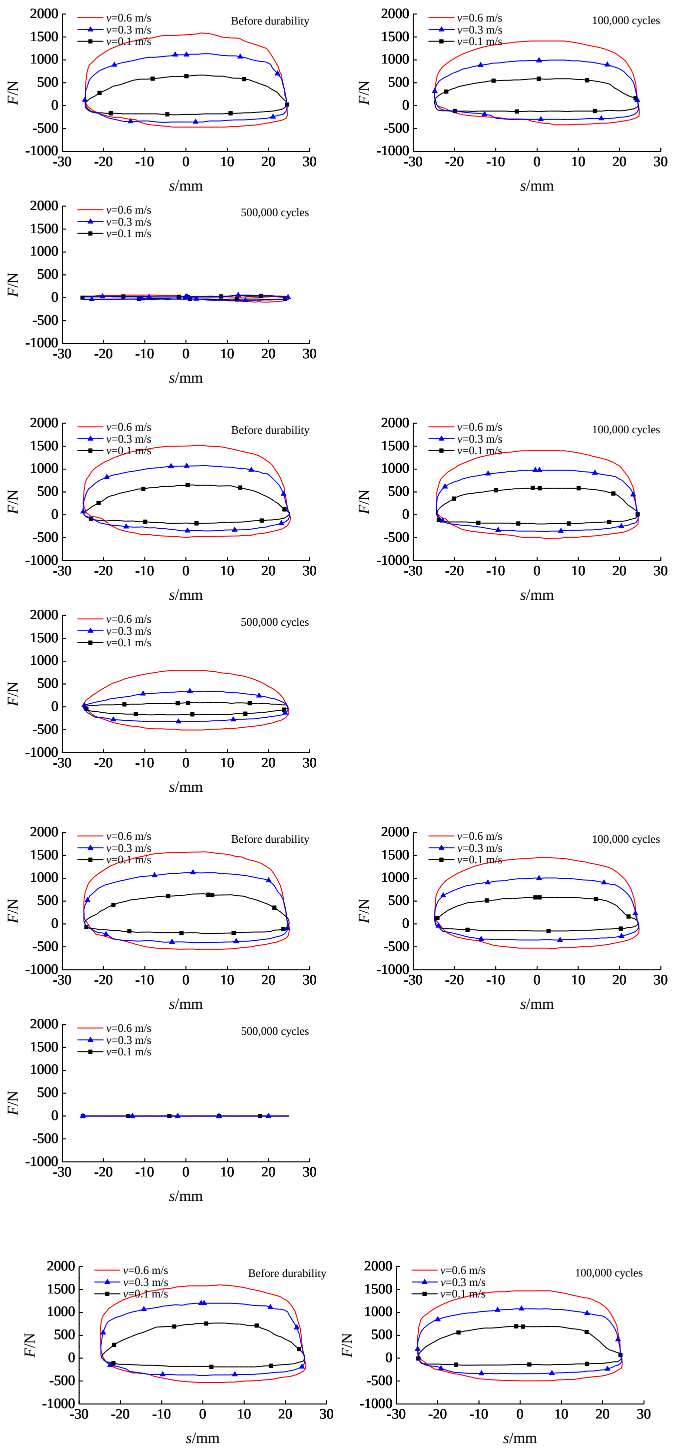
Double-action durability tests were carried out on the CV6H-A shock absorbers. The experimental apparatus contained shock absorber indicator instrument and a double- action test-rig. Here, a vertically mechanical double-action test-rig was employed to simulate work condition of the shock absorber during a vehicle driving on a real road. The double-action test-rig consists of a rack, an upper motion mechanism which could be utilizing to simulate the vibration of vehicle body, a lower motion mechanism which was able to simulate the vibration of the wheels, a lateral force loading mechanism, an automatic cooling system, an electronic control system, etc. The maximum amplitude of the upper and lower motion mechanism was 70 and 28 mm, respectively. The vibration frequency ranged from 0 to 5 Hz for the upper motion mechanism, while 1-15 Hz for the lower motion mechanism. The 4 shock absorber samples are shown in Figure 2, and the double-action durability test is shown in Figure 2.

Figure 1: Four shock absorber samples

Figure 2: Double-action durability test

The specific test process is shown as follow:

1. Test of the indicator diagram was carried out on individual shock absorber with the indicator instrument, recording the experimental data and defective indicator diagram for each sample.
2. Every sample was weighed and marked with an electronic scale.
3. During the durability test, in accordance with the actual vibration acting on the shock absorber, a low frequency stimulus with the frequency of 1 Hz and the amplitude of 80 mm was exerted on the up-lifting lug, while a high frequency stimulus with the frequency of 12 Hz and the amplitude of 20 mm on the down-lifting lug. The experimental temperature was controlled within the range from 60 to 80°C.
4. After the durability test of 100,000 cycles (counted by the low-frequency motion), the indicator diagram testing and weighing were conducted again for individual sample.
5. If there was no oil leak, the durability test would be carried on to 500,000 cycles (counted by the low-frequency motion). The indicator diagram is shown in Figure 4. The experimental data before and after the durability test were shown in Table 1 and 2.



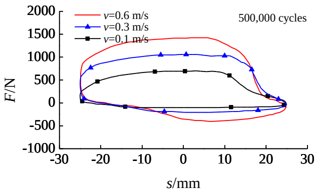


Figure 4: Indicator diagrams

Table 1: Damping force after100,000 cycles

Travel	Velocity /ms-1	Before durability/N				After 100,000 cycles/N			
		1	2	3	4	1	2	3	4
	0.1	661	659	668	774	581	589	581	698
Rebound	0.3	1131	1074	1128	1205	999	981	1006	1079
	0.6	1588	1517	1582	1595	1426	1411	1450	1469
	0.1	-191	-179	-190	-187	-141	-199	-164	-142
Compression	0.3	-352	-347	-386	-372	-322	-350	-366	-331
	0.6	-479	-472	-552	-534	-443	-501	-547	-493

It could be noticed from Figure 3 that all the indicator diagrams are defective. The oil seal of sample-3 cracked after 100,000-cycle durability test, with a little oil leakage. There was too much oil loss, about 100 g; furthermore, the damping force dropped near to zero after 500,000 cycles. Therefore, sample-3 lost efficacy. Similarly, sample-1 lost efficacy after 500,000 cycles, without any damping force. Generally, the damping force increases with the increase of the testing velocity. The maximum damping forces in the compression and rebound stoke at each velocity of the samples are shown in Table 1 and 2.

There is information of only sample-2 and -4 in Table 2, as the other two samples were in failure after 500,000 cycles. The velocity characteristics of sample-2 and -4 could be obtained from Table 1and 2, as shown in Figure 1. It shows that the damping forces in both rebound and compression stoke degraded a little after 100,000 cycles, while remarkably after 500,000 cycles. Moreover, the damping force of sample-2 degraded obviously in the rebound stoke while less visibly in the compression stoke; however, the damping force of sample-4 degraded in an opposite result.

Table 2: Damping force after500,000 cycles

Travel	Velocity / ms-1	Before durability/N		After 500,000 cycles/N		Rate of difference/%	
		2	4	2	4	2	4
	0.1	659	774	113	694	-82.8	-10.3
Rebound	0.3	1074	1205	382	1059	-64.4	-12.1
	0.6	1517	1595	894	1430	-41.0	-10.3
	0.1	-179	-187	-183	-125	2.2	-33.1
Compression	0.3	-347	-372	-326	-227	-6.0	-38.9
	0.6	-472	-534	-532	-414	-12.7	-22.5

The damping coefficient before the pre-valve opening in the rebound stoke is given as (Zhou and Meng, 2008)

$$c_{\mathrm{d}1} = \frac{F_{\mathrm{d}k1}}{V_{k1}} \tag{1}$$

where F_{dk1} and V_{k1} are the damping force and velocity at the pre-valve opening in the rebound stoke, respectively; k_1 is the slope of the velocity characteristic curve before the pre-valve opening in the rebound stoke.
The damping coefficient after the pre-valve opening varies with the velocity, which yields

$$c_{\mathrm{d}2} = \frac{F_{\mathrm{d}V}}{V} = \frac{F_{\mathrm{d}k1} + k_2\left(V - V_{k1}\right)}{V} \tag{2}$$

where V is the arbitrary velocity after the pre-valve opening in the rebound stoke; F_{dV} is the damping force at the velocity V; k_2 is the slope of the velocity characteristic curve after the pre-valve opening in the rebound stoke.

According to Eq. (2), The damping coefficient at the post-valve opening is given as

$$c_{\mathrm{d}k2} = \frac{F_{\mathrm{d}k2}}{V_{k2}} = \frac{F_{\mathrm{d}k1} + k_2\left(V_{k2} - V_{k1}\right)}{V_{k2}} \tag{3}$$

Where V_{k2} is the velocity at the post-valve opening in the rebound stoke; F_{dk2y} is the damping force at V_{k2}.

Similarly, the damping coefficient before and after the pre-valve opening, couple with the damping coefficient at the post-valve opening in the compression stoke is individually given as

$$c_{\mathrm{d}1y} = \frac{F_{\mathrm{d}k1y}}{V_{k1y}} \tag{4}$$

$$c_{\mathrm{d}2y} = \frac{F_{\mathrm{d}Vy}}{V_y} = \frac{F_{\mathrm{d}k1y} + k_{2y}\left(V_y - V_{k1y}\right)}{V_y} \tag{5}$$

$$c_{\mathrm{d}k2y} = \frac{F_{\mathrm{d}k2y}}{V_{k2y}} = \frac{F_{\mathrm{d}k1y} + k_{2y}\left(V_{k2y} - V_{k1y}\right)}{V_{k2y}} \tag{6}$$

According to Eq. (1)-(6), the damping coefficients could be calculated, as shown in Table 3. After the durability test, the damping coefficients of sample-4 were decreased; and those of sample-2 were decreased in the rebound stoke as well, while increased in the compression stoke.

Table 3: Damping coefficients (Nsm-1)

Test	Sample-2				Sample-4			
	Rebound		Compression		Rebound		Compression	
	c_{d1}	c_{dk2}	c_{d1y}	c_{dk2y}	c_{d1}	c_{dk2}	c_{d1y}	c_{dk2y}
Before durability	6590	2528	1790	787	7740	2658	1870	890
100,000 cycles	5890	2352	1990	835	6980	2448	1420	822
500,000 cycles	1130	1490	1830	887	6940	2383	1250	690

3. SIMULATION

3.1 Ride quality evaluation

The weighted root-mean-square (RMS) acceleration is a basic method to evaluate the effect of a vibration on comfort and health of human body. The weighted acceleration time history $a_w(t)$ can be obtained by the filter network of the corresponding frequency weighting function for the recorded acceleration time history $a(t)$. The weighted RMS acceleration is defined as

$$a_{\mathrm{w}} = \left[\frac{1}{T}\int_0^T a_{\mathrm{w}}^2\left(t\right)\mathrm{d}t\right]^{\frac{1}{2}} \tag{7}$$

where T is the time of vibration analysis.

The weighted RMS acceleration is determined for each axis (x, y and z) of translational vibration on the seat surface. The assessment of the vibration is made with respect to the highest frequency-weighted acceleration determined in any axis on the seat pan. When tri-axis translational vibrations at the seat surface along x-, y- and z-axis are considered simultaneously, the total weighted RMS acceleration along the three axes is given as below [12].

$$a_{\mathrm{v}} = \left[\left(1.4a_{xw}\right)^2 + \left(1.4a_{yw}\right)^2 + a_{zw}^2\right]^{\frac{1}{2}} \tag{8}$$

3.2 Vehicle model

To investigate the effects of the shock absorbers degradation on vehicle NVH performance, a vehicle model originated from a Chinese passenger car which the CV6H-A shock absorber served on was built by ADAMS/Car, as shown in Figure 5. This model consisted of the front and rear suspension, steering system, body system, braking system, tires and powertrain. The main configuration parameters are shown in Table 4. The same type of shock absorbers was adopted in the vehicle suspension.

Figure 5: Vehicle virtual prototype model

the shock absorbers results in an obvious peak in the vicinity of 3.6 Hz and causes a vehicle NVH performance which is worse in the frequency range from 2.2 to 7 Hz while better in that below 2.2 Hz and above 7 Hz. The front-left shock absorber degradation causes a significantly worse NVH performance, comparing to that at 0 mileage among the 4 individual shock absorbers ($p<0.05$, Wilcoxon matched-pairs signed rank).

This study can help in vehicle design for long term custom satisfaction. Further research will be carried out on the effect of shock absorber degradation on wheel bouncing and handling stability.

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