

Analysis of Dynamic Behavior of Twin-Tube Vehicle Shock Absorbers

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ABSTRACT

The vibration characteristics of the shock absorber in the dynamic condition differ from the "quasi-static" characteristics. For analysis of the processes occurring in the shock absorber, measurements were carried out on a universal servo-hydraulic test rig. The principle arrangement is described. A twin-tube shock absorber was prepared with acceleration, oil pressure and valve lift sensors.

To understand this changing of the operating characteristics the analysis of the vibration system "shock absorber" was carried out. The root cause for damping force irregularities was found. With knowledge of the valve operating characteristics in conjunction with the internal pressure characteristics, effective optimization strategies can be pursued.

INTRODUCTION

In modern vehicle chassis development, with regard to damping characteristics within the scope of low-frequency excitation (primary ride), high frequency performance (secondary ride) is becoming increasingly important. Chassis components such as shock absorbers in particular should not develop vibrations detrimental to comfort.

Inharmonious vibrations induced by the road surface cause highly dynamic processes in the shock absorber. High-frequency irregularities in the wheel motion sequence, which become apparent in the damping force, result in vibrations on the piston rod. These vibrations are subsequently transferred to the vehicle body and perceived as unpleasant by the vehicle occupants. The vibration characteristic of the shock absorber during these highly dynamic processes clearly differs from the „quasi-static“ characteristic used as a data pool for structural design.

This paper describes the shock absorber dynamics on the basis of extensive analyses of the operating procedures in the shock absorber.

DYNAMIC BEHAVIOR OF SHOCK ABSORBERS

DAMPING PRINCIPLE

Vehicle vibration damping takes place mainly by means of shock absorbers arranged parallel to the suspension. The hydraulic action of the shock absorber is produced by pressure differences on the piston or base valve. This hydraulic resistance is converted into forces across the displacement areas of the piston rod cross-section and the piston ring area. Fig. 1 shows the damping principle of a twin-tube shock absorber. As can be seen, the total shock absorber volume is divided into three working chambers: upper working chamber (above the piston), lower working chamber (below the piston) and compensating chamber (between the working cylinder and outer tube). The characteristics of the individual valve groups can be influenced by means of varying hydraulic resistances. Common hydraulic resistances are diaphragms, laminar cross-sections and pressure relief valves. With combination of these resistances the required damping characteristics is created. It shows the force response of shock absorber for road excitation.

Typical shock absorber excitation during ride on smooth road is the almost sinusoidal superimposition of car body and natural axle vibrations, whereby wheel rebound and jounce correspond to shock absorber rebound and compression. Noteworthy excitation amplitudes are in the frequency range up to 20 – 40 Hz, although the main components are below 15 Hz.

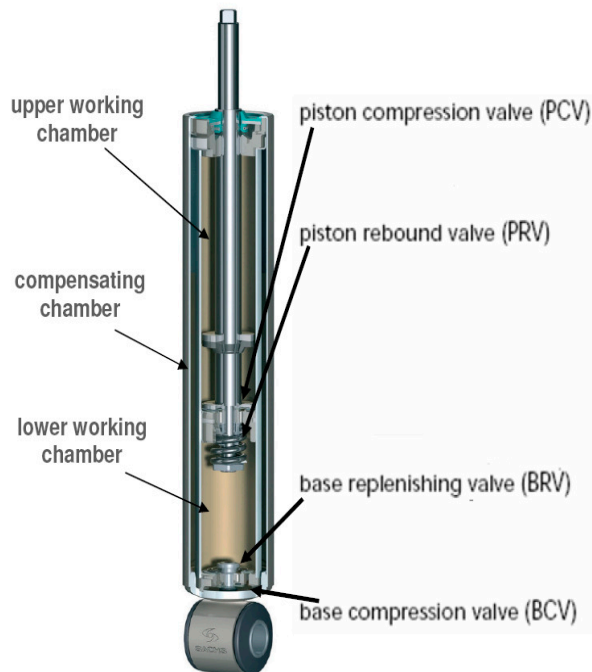


Fig. 1 Damping principle of a twin-tube shock absorber

METHODS FOR CHARACTERISATION OF SHOCK ABSORBER PROPERTIES

The performance of a shock absorber is primarily represented via damping force characteristics in form of a force-displacement (F/s) and/or force-velocity diagram (F/v). Two methods are used to measure these characteristic curves:

- continuous excitation frequency and variable amplitude (shock absorber strokes),
- continuous amplitude and variable excitation frequency.

As a rule, the measurements are carried out on a dynamometric test rig. The shock absorber is clamped into place with the receptacle on the actuator and the piston rod on the cross beam. The test rig actuator generates sinusoidal displacement excitation. With the second method the excitation amplitude is +/-50 mm and the frequency vary from 0,01 to 6 Hz. Data such as force and displacement, required to calculate the characteristic curve of the shock absorber, is acquired by sensors (load cell and linear displacement measurement) integrated in the test rig.

Depending on the chassis design and car manufacturer's tuning philosophy shock absorbers with linear or digressive characteristics are used in modern passenger cars. By way of example, Fig. 2 shows F/s and F/v diagrams for both types of characteristic curve [1].

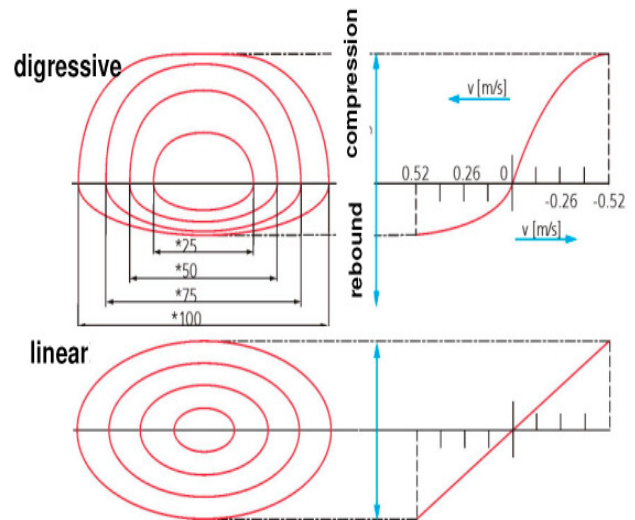


Fig. 2 F/s and F/v diagrams

The advantage of such presentations of the shock absorber characteristic is the ability to read directly the connection between the damping force and the compression / rebound velocity. The disadvantage of general F/s and F/v characteristics is that shock absorber properties are described only in the low frequency excitation range. Vibration development in the chassis and input of vibration energy in the vehicle structure are significantly affected by the dynamic structural properties of a shock absorber, above all within the high frequency range. These properties are precisely described via dynamic stiffness. This parameter shows the frequency-dependent relationship between complex force on the rigid fixed output of a shock absorber and the complex displacement on the input side during ordinary harmonic vibrations. The general formula for determination of dynamic stiffness is:

$$C_{\text{dyn}} = \frac{F}{u} \quad (1)$$

with C_{ody} – dynamic stiffness
 F – complex damping force
 u – complex displacement

The overall force of a shock absorber during harmonic excitation, with disregard to friction, can be divided up into three proportions:

$$F = \underbrace{(m_D \times a)}_{\text{inertial proportion}} + \underbrace{(k \times v)}_{\text{damping proportion}} + \underbrace{(c \times s)}_{\text{stiffness proportion}} \quad (2)$$

with m_D – shock absorber mass
 k – damping constant
 c – stiffness (spring) constant

A harmonic vibration can be transformed to the complex plain by means of a complex pointer rotating at angular frequency around the zero point. Dynamic stiffness can be divided up into its individual proportions with the aid of this representation:

$$C_{\text{dyn}} = \underbrace{C - \omega^2 \cdot m_D}_{\text{real proportion}} + \underbrace{j \cdot \omega \cdot K}_{\text{imaginary proportion}} \quad (3)$$

with C - stiffness proportion
 $\omega^2 \times m_D$ - mass inertia proportion
 $\omega \times K$ - damping proportion

Analysis of measuring displacement and force using the Fast Fourier Transformation permits calculation of the dynamic stiffness in the frequency domain. Usually following Spectral Characteristics are calculated:

- *Amplitude* transfer function (Transfer magnitude) - dynamic stiffness;
- *Phase* transfer function (Transfer phase) - the phase between output force and input displacement. The "ideal" damper has exactly 90° phase. The "ideal" spring has exactly 0° phase;
- *Real part* of transfer function (Transfer real) - stiffness part. The part of the output force that has a 0° phase shift related to the input displacement;
- *Imaginary part* of transfer function (Transfer imaginary) - damping part. The part of the output force that has a 90° phase shift related to the input displacement.

Dynamic stiffness measurements are possible on the basis of excitation profiles with *continuous amplitude* (amplitude in each excitation block is kept constant while the frequency is varied) and *continuous velocity* (shock absorber velocity is kept constant in each excitation block while the frequency is varied). Since the damping proportion and the valve opening time are directly dependent on the excitation velocity, an excitation profile with continuous velocity across the frequency is more appropriate for evaluation and analysis of the dynamic stiffness of the shock absorber. Experience in tuning work on shock absorber shows that suspensions with high dynamic stiffness have poor harshness comfort.

SHOCK ABSORBER CHARACTERISTICS DURING HIGHLY DYNAMIC EXCITATION

For analyze of dynamic shock absorber characteristics, the measurements are carried out in the high frequency range (up to approx. 75 Hz) on a servo-hydraulic test rig. Piston rod acceleration is recorded in addition to the force and displacement signals.

By way of example, Fig. 3 shows the measurement results of a twin-tube shock absorber. The black curve (1) shows the F/v - characteristic during an excitation frequency of 2 Hz and shock absorber velocity of 0.5 m/s, the red curve (2) shows the F/v characteristic at the same shock absorber velocity, yet with an excitation frequency of 25 Hz. An astonishing result is ascertained here: The vibration characteristics of the shock absorber during highly dynamic processes clearly differ from the „quasi - static“ characteristics used as a data pool for structural design.

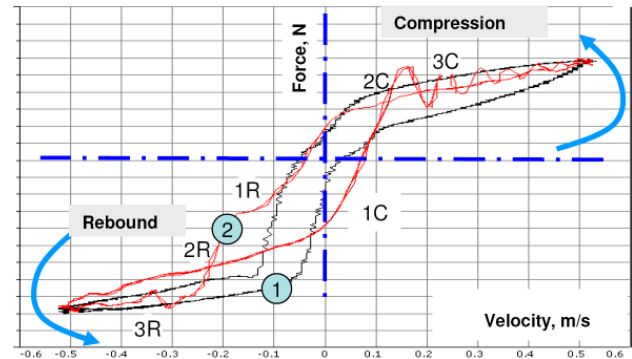


Fig 3 F/v - characteristics of damper during an excitation with low and high frequency

The following differences in force characteristics are noticeable during high-frequency excitation:

- hysteresis spread (delayed build-up of force) in the bleed area (points 1R and 1C), which means higher spring proportions in the damping force,
- intersection of ascending and descending characteristic curves (points 2R and 2C),
- distinctive discontinuities in force (3R and 3C).

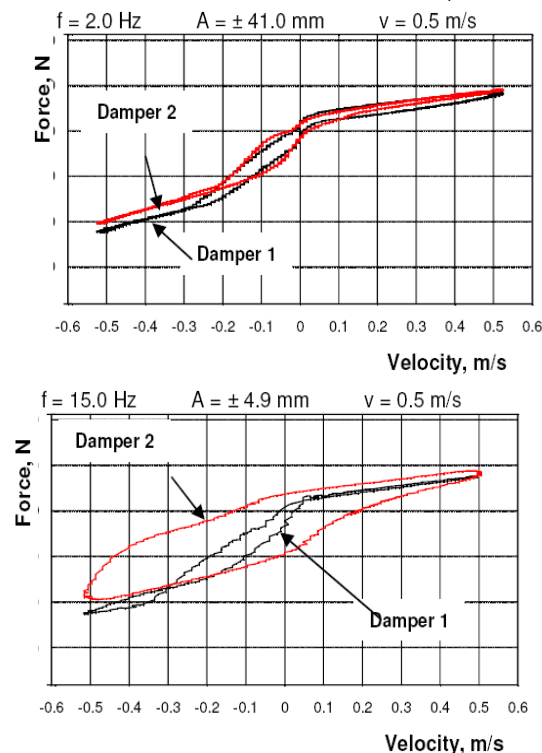


Fig. 4 F/v-diagrams measured for two shock absorbers

A further example is given by means of the F/v-diagrams measured for two shock absorbers that primarily differ in dimensions and valve design (Fig. 4). Upper diagram shows the measurement during an excitation frequency of 2 Hz and shock absorber velocity of 0.5 m/s and lower diagram shows the measurement at the same shock absorber velocity, yet with an excitation frequency of 15 Hz. It is apparent that during low-frequency excitation the