

# Double Tube Shock Absorber Model for Noise and Vibration Analysis

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## ABSTRACT

The “structure-borne” noise of the shock absorber is often responsible for undesirable noise in the car interior cabin. These vibrations are attributed to friction, opening/closing of the valves, fluid cavitation or other complex phenomena.

Early numerical prediction of the level of these vibrations in the car development process saves time and money. Most of the shock absorber models existing in the literature are limited to analysis of vehicle ride and handling. For noise and vibration analysis, the published works do not explicitly describe any model with its associated assumptions and a clear correlation with the experiments for high frequencies. Moreover there is no interpretation of the physical meaning of the high-frequency content of the response.

The objective of the present work is to build a double tube shock absorber model correlated up to 700 Hz. Experimental testing results are presented in this paper in order to put in evidence the non-linear phenomena localized on the piston shock absorber. The model includes the fluid compressible behavior, non-linear flow-rate pressure relationship and spring valve dynamics. Unknown parameters like bulk modulus are identified with experimental data.

The numerical results are compared to the experiments, showing that the shock absorber model is able to reproduce very well the local non-linear phenomena. This confirms our hypothesis and improves our knowledge of the potential noise sources in the shock absorber.

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## INTRODUCTION

Structure-borne noise of the shock absorber occurs when the shock absorber is excited with a low frequency (10-15 Hz) and with low amplitude (few millimeters). This type of excitation leads to a shock absorber response with transient high frequency content typically between 200 Hz and 700 Hz. These high-frequency forces are transmitted to the car body causing what we call the “rattle noise”. The structure-borne noise of the shock absorber is often attributed to valve opening/closing, friction and valve impacts on its seat.

Numerical prediction of the structure-borne noise early in the car development process necessitates a correlated model up to 700 Hz. Most of the shock absorber models found in the literature are limited to analysis of vehicle ride and handling, that is for frequencies [0 30 Hz] [1]. Lauwery's [1] uses

numerical simulation to study high-frequencies, but the model is not detailed and the correlation between experiments and simulation is not clearly presented. Sacramento in [2] shows that his model is correlated up to 250 Hz. But the choices of the model and equations are not clearly presented. To our knowledge, in the literature, we found no shock absorber model with correlation up to 700 Hz.

The objective of the present work is to present our methodology to build a shock absorber model for temporal simulation based on experimental results and which is correlated up to 700 Hz with the experiments. The first part of this paper aims at describing the experimental tests conducted in order to identify the physical sources of the noise and to identify the model parameters. The second part is a description of the shock absorber model and its

assumptions. Finally the third part gives the methodology of parameters identification and the correlation results.

## SHOCK ABSORBER TESTING

This paper is focused on an automotive double-tube rear shock absorber. [Figure 1](#) is a scheme of this type of shock absorber. It has three chambers. The chamber above the piston is the rebound chamber, the chamber below the piston is the compression chamber and the third chamber is the reserve chamber. The fluid flows between the rebound chamber and compression chamber via the piston valves and orifices, and between the compression chamber and the reserve chamber via the base valves. The reserve chamber is partially filled with gas to compensate the volume of the rod when entering or exiting the rebound chamber. A lot of technologies exist to obtain a non-linear force-velocity characteristic. The [Figure 2](#) shows the technology studied in this paper, the piston is made of spring valves and the base valve is made of discs stack for compression and a check valve for rebound.

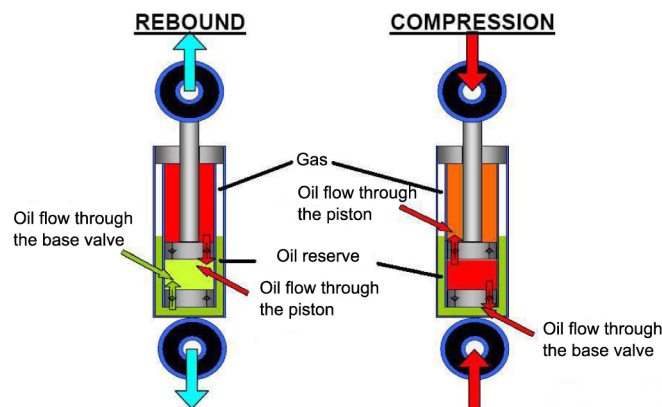


Figure 1. Double-tube shock absorber functioning.

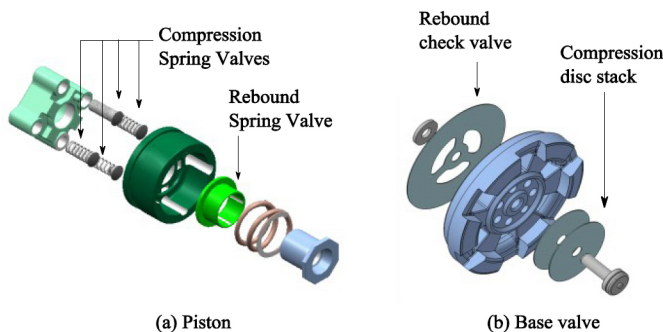


Figure 2. Technologies studied in the present paper.

## Experimental Configuration

The piston and the base valve of the shock absorber are placed in a special device in order to be able to measure pressure in the chambers. The experiments are done without the shock absorber mounts for two reasons:

- Reminding that we are interested in the source of the structure-borne noise of the shock absorber, the first reason is that the mounts are involved in the transmission of the vibrations to the car body and play no role as a source.

- The second reason is that the high frequency phenomena are better observed when mounts are removed to prevent their “masking” of high frequency content.

The excitation is imposed on the shock absorber body. The rod is maintained fixed. The imposed signal is a 12 Hz - 2 mm sinusoidal displacement for the structure-borne noise study and 1 Hz - 25 mm for parameters identification. The sampling frequency is 23.4 kHz, enough to catch very well oscillations until 1 kHz. Our sign convention on the body displacement is positive axis oriented downward. The recorded data are the rod force, the pressure in the chambers, displacement and velocity of the shock absorber body.

Two configurations are tested:

- Configuration 1: Piston and base valve are both placed in the special device ([Figure \(3a\)](#)).
- Configuration 2: The base valve is removed and the piston is kept alone. This configuration allows tracking the phenomena on the piston only ([Figure \(3b\)](#)).

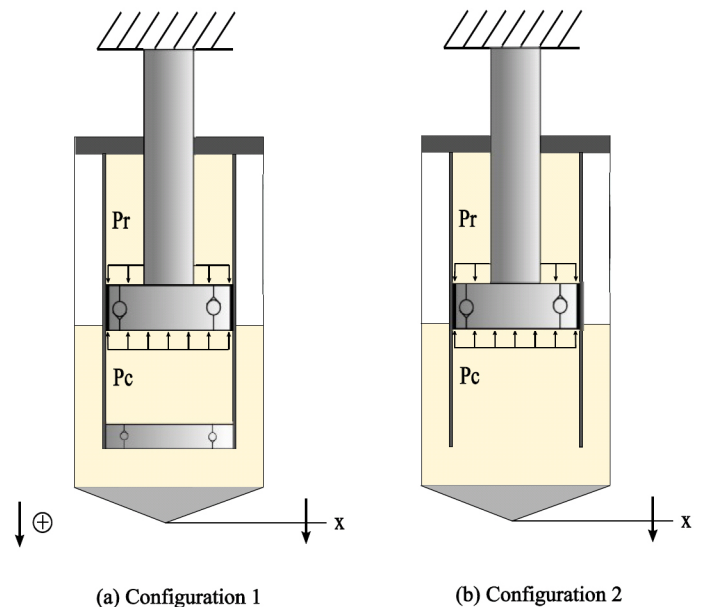


Figure 3. Illustration of the two tested configurations.

## Experimental Results

The high-frequency content of the shock absorber is observed by applying a band-pass filter [200 - 700] Hz to the temporal measured rod force at the reference excitation. A “good” shock absorber from a noise point of view is a shock absorber with a low level of the rod force above 200 Hz.

The [Figure 4](#) plots the rod force recorded for configuration 2 during one period. The upper figure is the measured rod force and the bottom figure is the rod force

with applied band-pass filter [200 - 700] Hz. The bottom figure shows great impulse between 0.96 s and 0.97 s. The phenomenon occurs in both configurations, with a higher level in Configuration 1. The moment where the phenomenon occurs is at the beginning of the compression phase, when the first compression spring valve opens. The shock absorber model will be able to confirm this hypothesis.

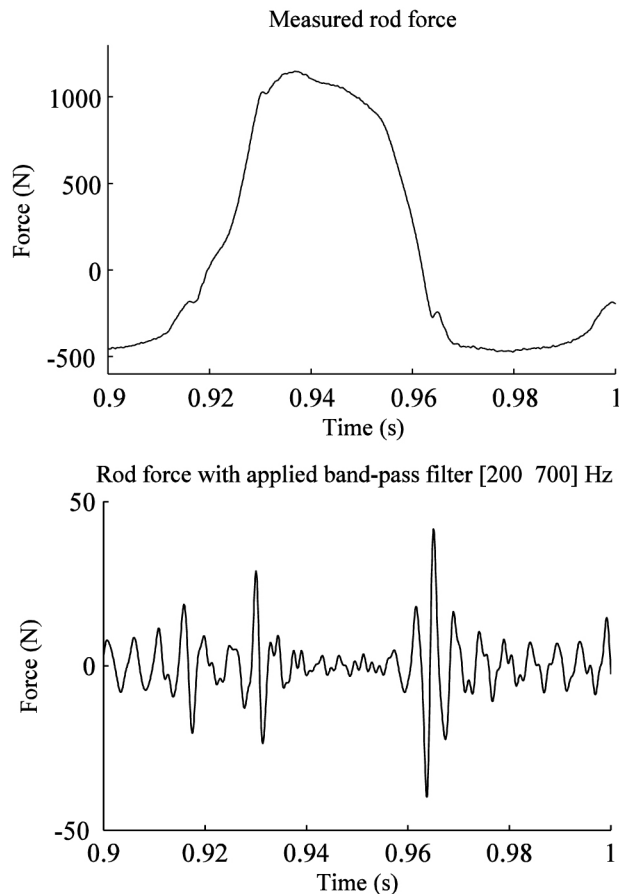


**Figure 4. Measured rod force and its dynamic content between 200 Hz and 700 Hz for configuration 2.**

The Figure 5 shows the results of the measured pressures in the three chambers and the input displacement. The pressure measurements help dissociating phenomena on the piston and on the base valve.

The first remark is that the pressure in the reserve chamber is almost constant. This is due to the small input displacement and high compressibility of the gas.

The second remark is about the observed discontinuities:

- At the beginning of the rebound phase around 0.92 s, we observe a “bump” in the compression chamber. This moment is the beginning of the rebound phase. This is probably due to a phenomenon preventing the rebound check valve to open such as stiction. This bump is not visible in the force measurement and we believe it does not play an important role in the high frequency emissivity.
- At the beginning of the compression phase around 0.96 s there is another “bump” in both compression and rebound

chamber. This moment is the beginning of the compression phase. This one is visible in the rod force measurement and we believe it is due to the first spring valve opening.

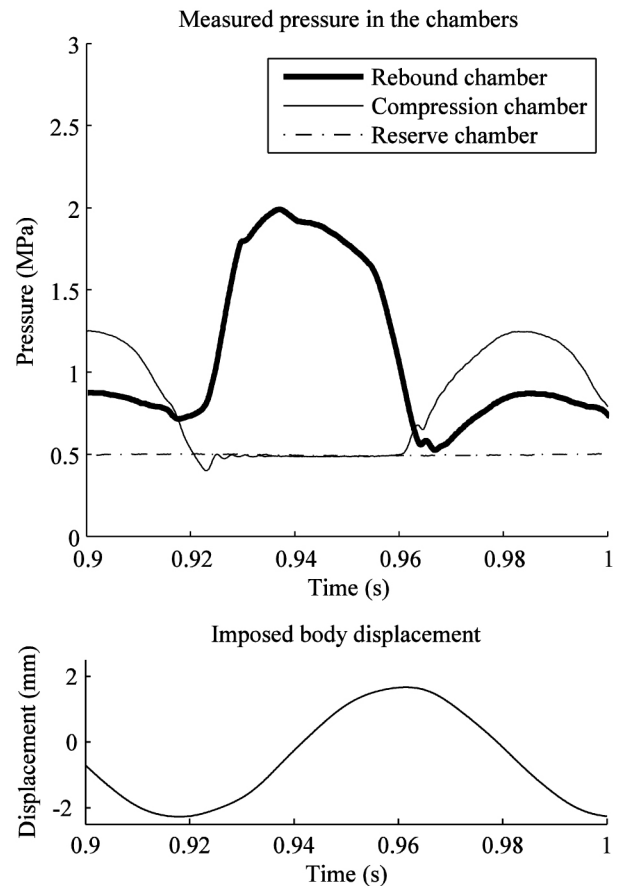


**Figure 5. Measured pressure in the chambers for Configuration 2.**

In order to simplify our study, we will study each component (piston and base valve) separately. As the piston carries most of the shocks and nonlinearities, we will focus our study on this component. All the dynamic phenomena are not visible with a force-velocity characteristic diagram of the shock absorber. It is necessary to build a detailed model of the shock absorber components and fluid properties.

## SHOCK ABSORBER MODEL

The experiments showed that the shock absorber generated force contains local phenomena responsible for the high-frequency emissivity of the shock absorber. The shock absorber model proposed by Reybrouck in [3] gives good results in the frequency range of the ride and comfort analysis for passenger cars. This model was improved by Duym in [4] with a more detailed model of the oil behavior in contact with gas. The fluid model takes into account the effects of the variation of fluid compressibility due to gas bubbles present in the oil. Models including a mechanical model of valves made of diaphragms are found in [5] and [6] and for spring valves in [7]. More detailed spring valves model and analysis