A Detailled Shock Absorber Model for Full Vehicle Simulation

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Fichtel & Sachs, a Mannesmann company, is a supplier in the automotive industrie. Main products are shock absorbers, clutches, electronic clutch systems and torque converters.

ADAMS has been used at F&S for about six years to suppport the development of these products. In addition we are using ADAMS for dynamic analysis of suspension systems and axles. The accuracy of the shock absorber description has a perceptible influence on the results of vehicle- and axle - simulation. Therefore we focused on a detailled shock absorber description.

Double-tube pressurized shock absorbers consist of two chambers filled with oil. The working chamber containing the piston rod and the piston valve as well as the bottom valve is entirely filled with oil and devided into an upper and a lower part by the piston. The oil reservoir between the working cylinder and the outer tube is filled to about 2/3 with oil and about 1/3 with pressurized gas.The oil reservoir also serves as a compensating chamber for the oil displaced by the moving piston rod.

The damping valves - bottom and piston valve - consist of a system of small spring washers, coil springs and valve bodies machined with bores. Piston rod and seal are particularly important components, because the pressurized oil must be perfectly sealed while the piston rod stops or moves. There is no principal physical difference between double tube and single tube shock absorbers.

The damping force is due to the pressure differences between the upper and lower working chambers. These pressures result from fluxes caused by the piston movement and fluxes through valves. This is the starting point for our shock absorber model, which is based on the flux balance equationes of the damper.

Fluxes due to piston movement are

$$Q_{Piston} = A_{piston} v$$

$$Q_{Piston} = Q_{Elasticity} + Q_{Valve}$$

$$Q_{Elasticity} = Q_{Oil} + Q_{Tube}$$

A - piston cross-section, v - velocity.

From the flux balance equation

$$A_{Piston} v = Q_{Oil} + Q_{Tube} + Q_{Valve}$$

one can derive the pressure equation in the working chambers

$$\frac{dp}{dt} = \frac{A_{Piston} v - Q_{Valve}}{\frac{V_{Oil}}{k} + \frac{D_i^3 \pi L}{2E(D_a - D_i)}}$$

where

$$K = \frac{V_{Oil}}{k} + \frac{D_i^3 \pi L}{2E(D_a - D_i)}$$

is the contribution of elasticities with

k - coefficient of oil compressibility
E - coefficient of steel elasticity
V_{Oil} - oil volume
D_a, D_i - outher and inner tube diameters
L - active length.

Analytical expressions describing fluxes through valves may be very complex even if approximations are used.

However, at F&S there exist a huge digital database of measured pressure-flux characteristics of the different valves developed by our company. So, instead of deriving analytical expression for complex valves we think it is more efficient and precise to use the correct characteristics from our database for our model.

To simulate a specified shock absorber we need the pressure equations for the compression phase

$$\frac{dp_u}{dt} = \frac{-(A_{Piston} - A_{Piston\ rod})v + Q_{AP}}{K}$$

$$\frac{dp_l}{dt} = \frac{A_{Piston} v - Q_{Bottom} - Q_{AP}}{K}$$

and the rebound stroke

$$\frac{dp_u}{dt} = \frac{(A_{Piston} - A_{Piston\ rod})v - Q_{Piston}}{K}$$

$$\frac{dp_l}{dt} = \frac{-A_{Piston} v + Q_{Piston} + Q_{AB}}{K}$$

with Q_{AP}, Q_{AB} - auxiliar valves and p_u, p_l - pressure in the upper and lower working chambers.

The gas pressure is determined by the adiabatic state equation. The gas volume changes as the piston rod moves and causes a continious change of the system pressure. This pressure change results in a hysteresis in the force-velocity characteristic at all excitation frequencies of the damper.

$$p_{Gas} = -p_{Gas0}\left(\frac{V_{Gas0}}{V_{GAS}}\right)^{\kappa}$$

with

$$V_{Gas} = V_{Gas0} - Q_{Bottom} t$$

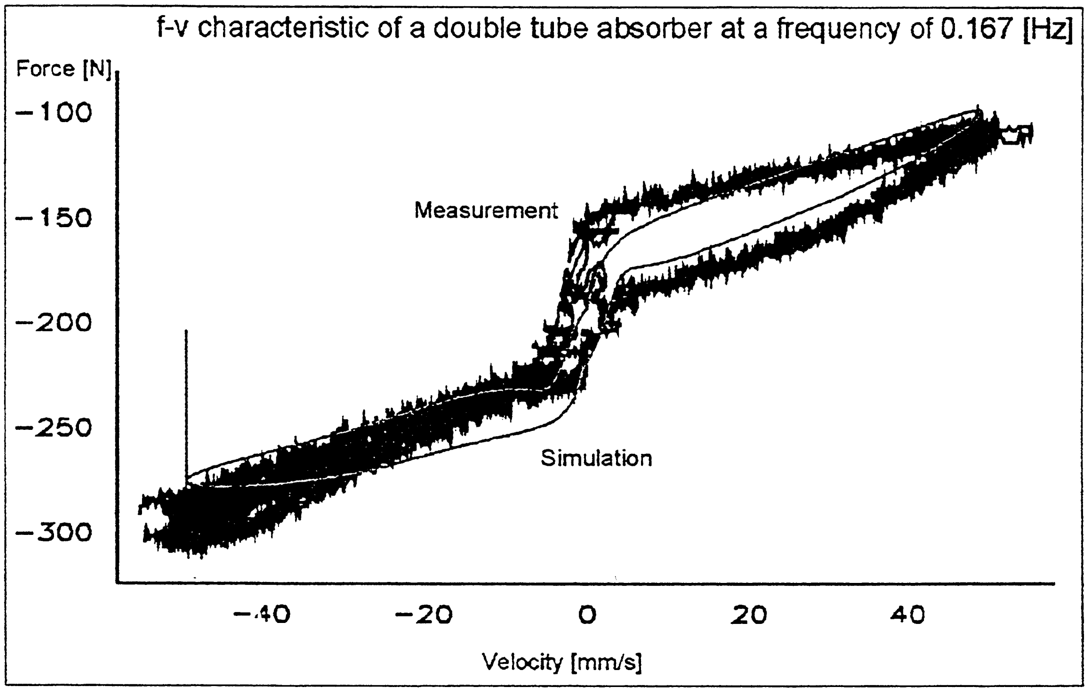
and t - time.

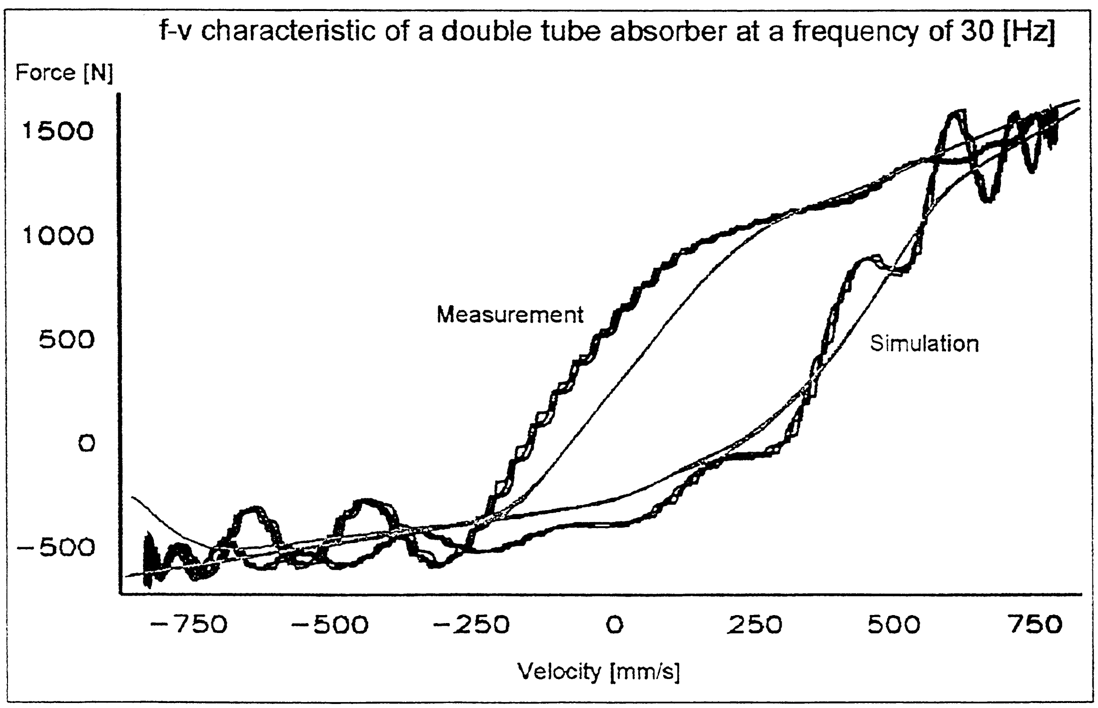
As the piston sealing and the piston rod guide sealing have to be very tight friction forces up to several hundred Newton may appear in the damper especially in McPherson struts. The friction should be proportional to the forces in the sealings when a lateral momentum is applied to the damper. From the momentum balance equation one can derive expressions for these forces.

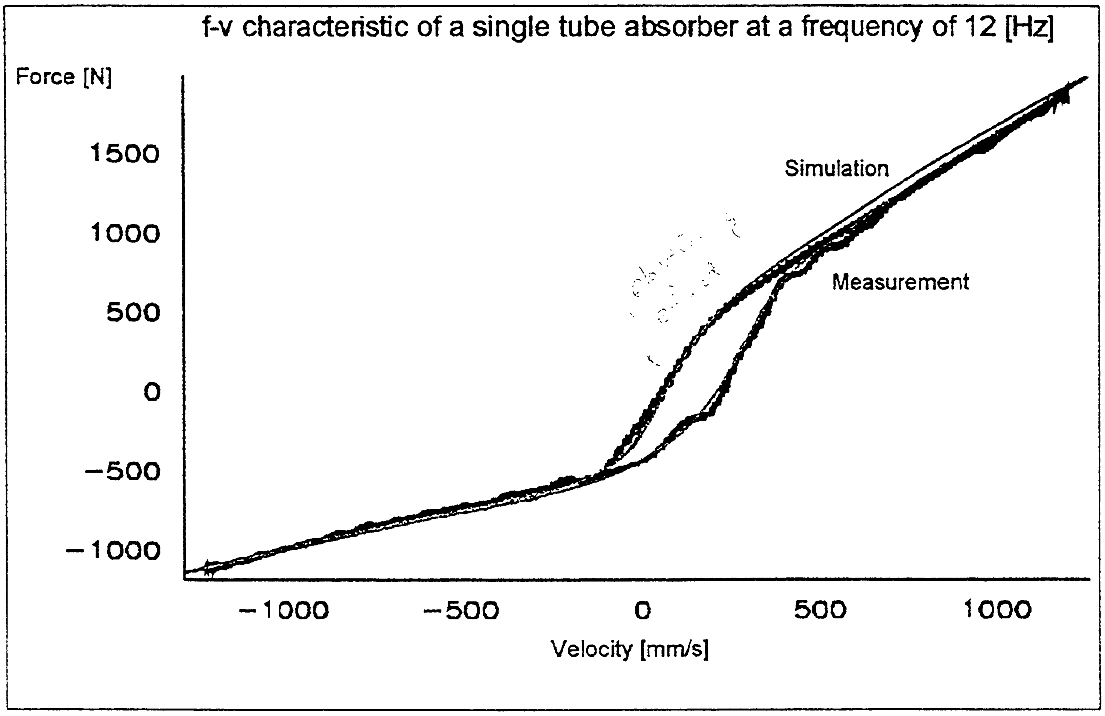
To include slip-stick effects, which influences to some extend the nvh-properties of the car body, we decided to use an analytical formulation of these effects. It gives the possibility of an adaption of the friction properties by a simple change of several parameters.

Because of the similarity of the physics of slip-stick effects in shock absorbers and the physics of the tyre-road contact patch, we applied the first version of the so called magic formula used in tyre models.

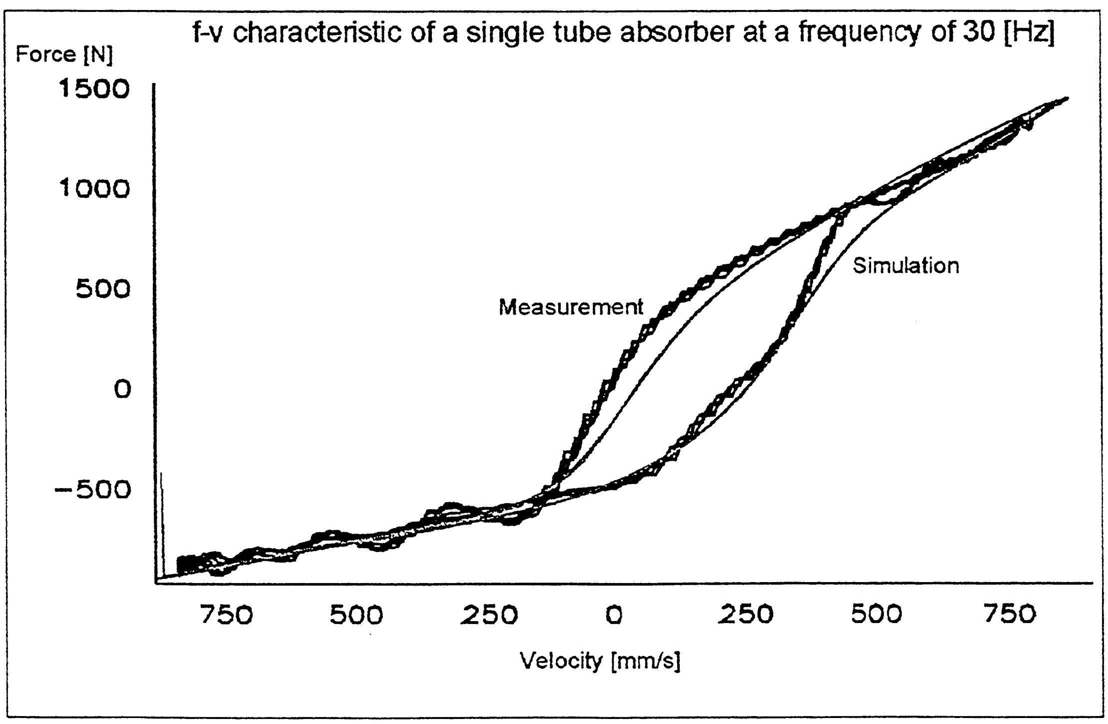
The next graphs show some results of our model compared to measurements. As seen from these pictures our model fits well to experimental data.


f-v characteristic of a double tube absorber at a frequency of 0.167 [Hz]
Force [N]
−100
−150
−200
−250
−300
Measurement
Simulation
−40
−20
0
20
40
Velocity [mm/s]


f-v characteristic of a double tube absorber at a frequency of 30 [Hz]
Force [N]
1500
1000
500
0
−500
Measurement
Simulation
−750
−500
−250
0
250
500
750
Velocity [mm/s]



f-v characteristic of a single tube absorber at a frequency of 12 [Hz]
Force [N]
1500
1000
500
0
−500
−1000
Simulation
Measurement
−1000
−500
0
500
1000
Velocity [mm/s]



f-v characteristic of a single tube absorber at a frequency of 30 [Hz]
Force [N]
1500
1000
500
0
−500
Simulation
Measurement
750
500
250
0
250
500
750
Velocity [mm/s]

From the flux balance equation

$$A_{Piston} v = Q_{Oil} + Q_{Tube} + Q_{Valve}$$

one can derive the pressure equation in the working chambers

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