Nonlinear Stiffness of Suspension Parts **AN 26**

Application Note to the KLIPPEL R&D SYSTEM

The nonlinear stiffness K(x) and the reciprocal compliance C(x) of any suspension parts (spider, surrounds, cones) and passive radiators (drones) are measured versus displacement x over the full range of operation. A dynamic, nondestructive technique is developed which measures the parts under similar condition as operated in the loudspeaker. This guarantees highest precision of the results as well as simple handling and short measurement time. Suspension parts are fixed in the measurement bench by using a universal set of clamping parts (rings, cones, cups) fitting to any size of circular geometries between 1.5 – 9 inch diameter. Special clamping parts for other geometries can be manufactured at low cost. The working bench excites pneumatically the suspension to vibration at the resonance frequency related to the stiffness and the mass of the suspension and inner clamping parts. The nonlinear stiffness is calculated from the measured displacement by using modules of the KLIPPEL R&D SYSTEM. The measured parameter is required for specifying the large signal properties of the suspension parts and to detect asymmetrical and symmetrical variation which are the cause for instable vibration behavior and nonlinear distortion.

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Theory

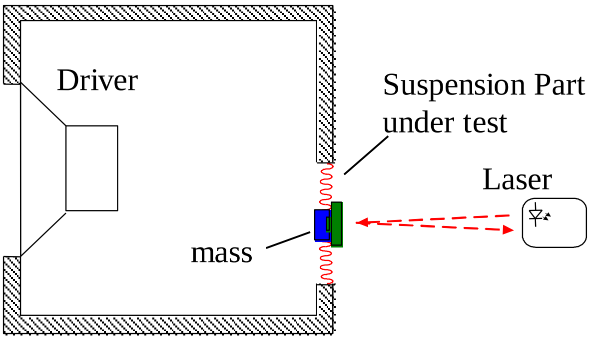
Static techniques

The EIA standard RS 438 describes a method for measuring the stiffness of a spider at a single displacement created by hanging a known mass from a cap at the inner diameter of the spider. While this method serves a purpose in providing a quickly-obtained estimation of spider stiffness using relatively inexpensive equipment, the measurement does not yield any information about the nonlinear behavior of the spider. Furthermore, this method may be prone to measurement error due to its highly manual nature. In the meantime additional computer controlled methods are developed that provides the stiffness K(x) versus displacement by using also a static technique. Since the stiffness K(x,t) of the suspension depends on displacement x and time t there are discrepancies between static measurement and dynamic application of suspension part:

1. The stiffness K(x) measured statically at peak displacement $x=+-X_{peak}$ is usually lower than the stiffness measured at this point with an audio-like signal. Generating a static displacement of $x=+-X_{peak}$ the required force decreases slowly with time (creep).
2. The stiffness K(x) measured statically at rest position $x \approx 0$ is usually higher than the stiffness found by dynamical techniques. The cause of this phenomenon is discussed below .

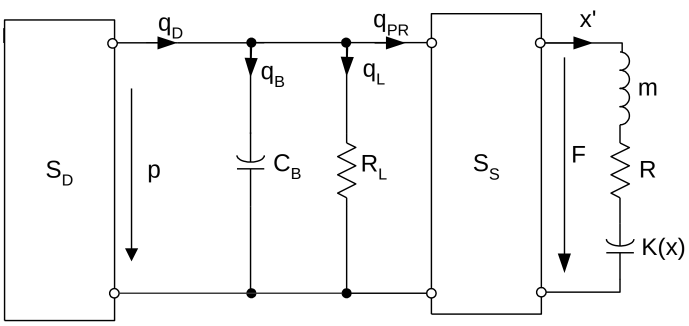
Furthermore, other practical concerns (reproducibility, practical handling, time) gave reason for the development of a dynamical method:

Theory of the dynamic technique



The figure above shows essential parts of the measurement setup. An 18" driver is mounted in a sealed enclosure having a second hole where the suspension part under test is placed. The outer rim of the suspension part is firmly clamped by rings.

The inner neck of the suspension part where usually the voice coil former sits there is clamped by an additional mass. The displacement of the suspension part is measured by using an optical sensor (laser).



The driver generates a volume velocity q_D that divides into three parts as shown in the equivalent circuit above. The volume velocity q_B flows into the volume of the box, q_L is leaving the box through leaks and the volume velocity q_{PR} produces the force F driving the suspension part under test

$$F = K(x) \cdot x + R\frac{dx}{dt} + m \cdot \frac{d^2x}{dt^2}.$$

The effective area S_s of the suspension part (spider) is usually not identical with the geometrical area because the material of a spider is porous. The moving mass *m* (formed by the inner clamping parts such as slide, cone, cup and nuts together with the moving part of the suspension and the air load) and the stiffness *K(x)* form a resonating system. The resonance frequency f_s is defined as the frequency where the displacement becomes maximal. The resonance frequency f_s corresponds with an effective stiffness

$$K_{mean} = m(2\pi f_s)^2$$

using the moving mass m.

At resonance the force accelerating the mass *m* equals the restoring force of the suspension. The driving force *F* compensates for the mechanical and acoustical losses represented by resistance *R.* Using a sufficiently large measurement box giving a high acoustical compliance C or sufficient leakage giving a low resistance R_L the stiffness *K(x)* can be calculated from the displacement measured at resonance and the known moving mass *m*. The displacement *x* is measured by using an inexpensive triangulation laser as provided by the KLIPPEL system. The mass *m* is approximately equal to the total mass of the suspension and inner clamping parts weighed before. The part of the suspension which is clamped and can not move is neglected. This simplification can be done, as the additional mass of the inner clamping parts is much larger than the mass of the suspension part.

Requirements

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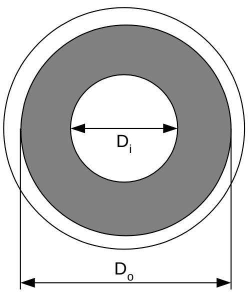
To measure the nonlinear stiffness of a spider the following equipment is required:

- Suspension Part Measurement Bench
- Suspension Part Software (SPS)
- Distortion Analyzer (DA1 or DA 2) + Cables
- Software modules Transfer Function (TRF)
- 2 clamping rings (or the ring set)
- 1 cone (or the cone set)
- 1 cup (or the cup set)
- slide, 2 nuts
- Power amplifier
- Scales for measuring the mass

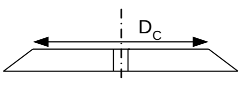
Clamping the Suspension Part

Dimensions of the Suspension



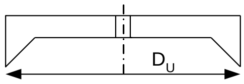
1. Measure the inner diameter D_i of the suspension part
2. Measure the outer diameter D_o of the suspension part (without rim)

Find the Cone



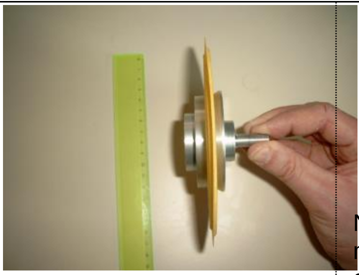
3. Look in the look-up table (User's Guide) for cones to find the optimal cone (for example **3**) having a inner diameter D_C which is just smaller than the measured inner diameter D_i .

Find the Cup



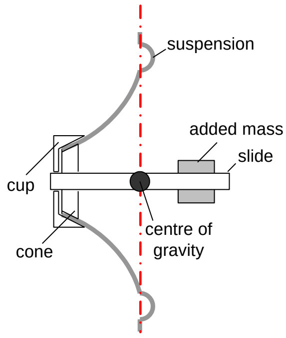
4. Look in the look-up table (User's Guide) for cups to find the optimal cup (for example **A3**) having an inner diameter D_U which is just larger than the measured inner diameter D_i .

Inner Clamping



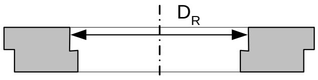
5. Clamp the inner rim by using the slide, cone, cup and two nuts.

Note: Use an additional mass to keep the centre of gravity between the clamping points of the outer rim.



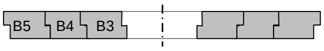
6. Weigh the surround with the inner clamping

Find the lower clamping ring



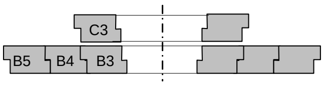
7. Look in the look-up table for rings to find the lower clamping ring (for example **B3**) having an inner diameter D_R which is just larger than the measured outer diameter D_o .

Find the lower ring set



8. Complete the lower ring set by selecting all rings which have the same character in the nomenclature (for example **B**) and are larger than the lower clamping ring (for example **B4**, **B5, B6**) to complete the lower ring set.

Find the upper clamping ring

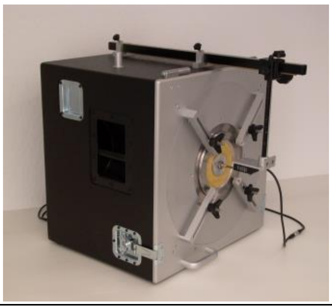


9. Look in the look-up table to find the one-step larger ring used as upper clamping ring (for example **C3**).

Outer Clamping

10. Bring the clamping platform into horizontal position for easy handling
11. Insert the set of lower rings into the clamping platform
12. Put the slide with the clamped suspension on the guiding rod. Make sure that the suspension part is in horizontal position when clamping. Support manually if needed to prevent any sagging due to gravity.
13. Fix the upper ring to clamp the outer rim
14. Insert and adjust the laser head (for details see Hardware Setup in the chapter Using the Laser Sensor of the Hardware Guide in the Manual).

Measurement Position



15. Bring the clamping platform into vertical position and close the bench
16. Mount the laser rod and adjust the laser sensor
17. Check slider for easy movement on the sliding rod.

Measurement of the Nonlinear Stiffness

Procedure

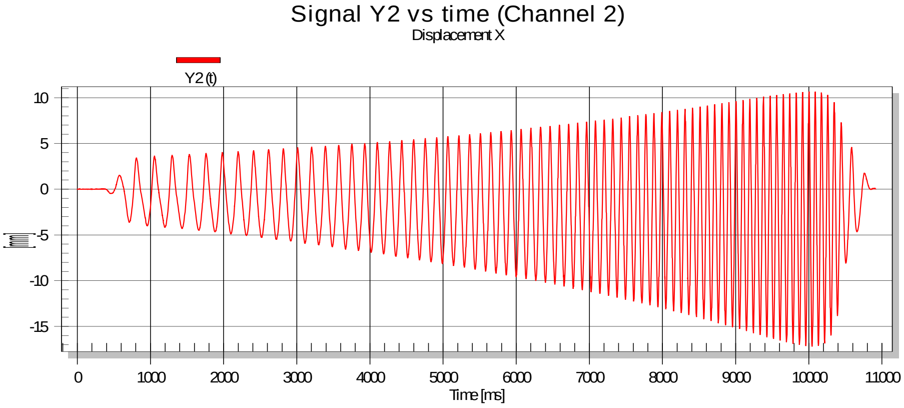
Create a new operation and select **SPM Pro Suspension Part Measurement** from the module list.

Enter the **Moving Mass,** which is the mass of the suspension part including the inner clamping. Set the desired **Target Displacement** and click on **Run.**

The measurement will automatically adjust the stimulus and voltage settings to reach the desired target displacement and measure the nonlinear stiffness.

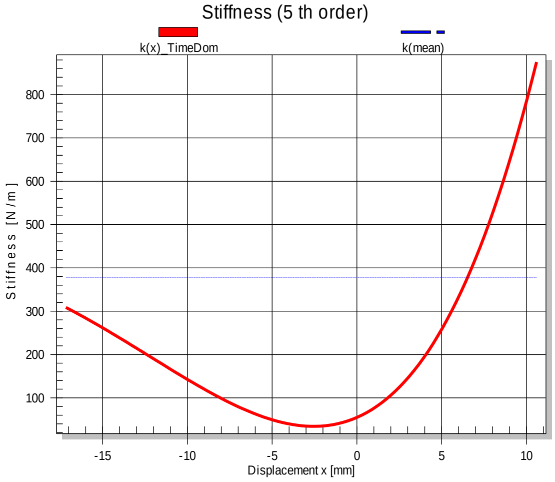
Example

X(t) Displacement



The window **X(t) Displacement** shows the displacement time signal swept from 4 to 12 Hz. The displacement rises slowly up to [-16…+10] mm peak but decreases rapidly after the resonance. This is caused by the suspension nonlinearity which causes a bifurcation and a jumping effect to lower amplitudes. The time signal also reveals a dc part in the displacement generated by the stiffness asymmetries of the suspension.

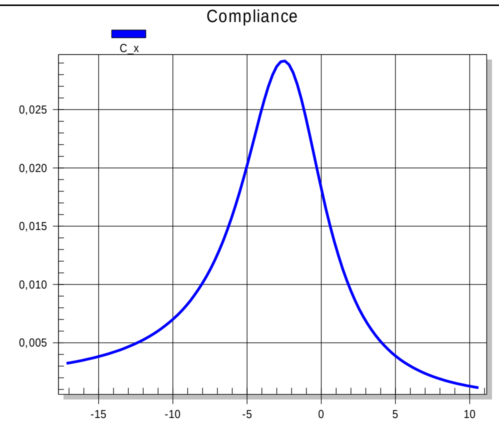
Stiffness k(x) of a spider



The stiffness curve K(x) versus displacement is shown in the figure. For a positive displacement x=+11 mm the stiffness is approximately 30 times higher than at the rest position x=0. Please note the distinct asymmetry of the curve. The stiffness at negative displacement x=-11 mm is only 16 % of the stiffness at positive displacement x=+11 mm. Under dynamic operation an ac-signal is partially rectified and a negative dc-component is generated.

The dashed blue curve shows the mean stiffness K_{mean} of the suspension in the working range [-17 mm to 11] mm. This value depends on the amplitude and corresponds with the effective resonance frequency found in the large signal measurement. This value is simple and convenient for QC applications. It can also calculated directly from the resonance frequency f_s and the moving mass m.

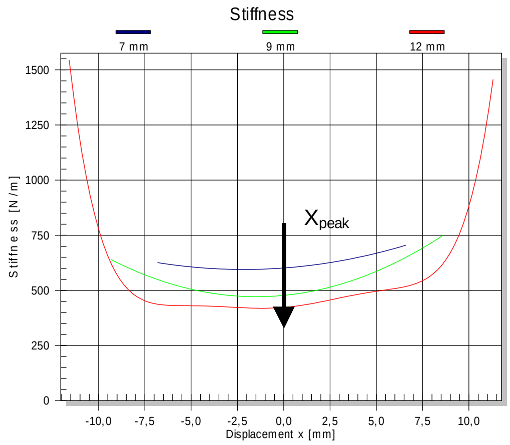
Compliance C(x)



Activating the window **C(x) Compliance** shows the nonlinear compliance C(x) versus displacement x which is identical with the inverted stiffness. Note that the stiffness curve reveals the limiting effect more clear than the compliance curve and is more recommended for graphical representation.

Phenomena

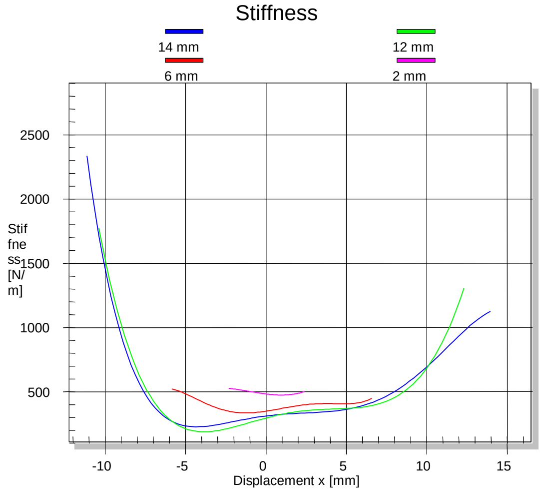
K(x=0) varies with amplitude



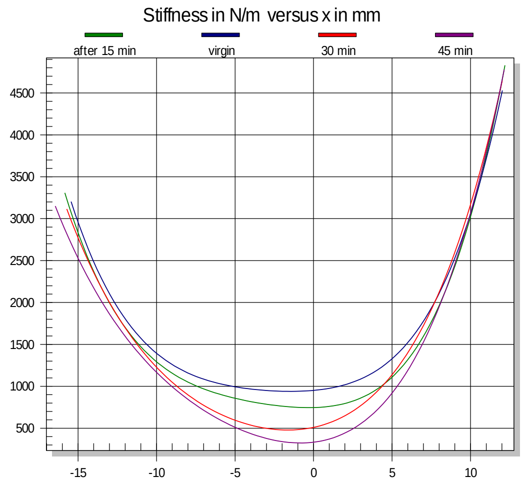
Exciting a suspension part (especially spiders) by an ac signal with a peak amplitude X_{peak} the stiffness K about the rest position x=0 will depend on the peak displacement occurred in the last instance. At high amplitudes the stretching of the corrugation roles causes a temporary deformation of the fiber structure and makes the suspension softer at medium amplitudes. However, this kind of deformation is reversible. It stays only for multiple periods of the ac signal and recovers completely after a few seconds. Thus the effect can not be measured by using a static technique.

This effect depends on the geometry and impregnation of the suspension material. It increases the nonlinearity of the suspension which becomes not only stiffer for larger displacement but also softer between the excursion maxima.

DC Displacement



The dynamic measurement technique also reveals the dc displacement caused by asymmetries of the suspension. The ac displacement is partly rectified and produces a dc component which operates the suspension at a region of lower stiffness. The example shows a positive dc-displacement of 1.5 mm generated by a cone where the surround has a significant asymmetry at negative displacement.

Ageing of the suspension		The dynamic measurement technique is also convenient for investigation of the break in and other ageing effects of the suspension. The example shows the change of the stiffness versus time while permanently exciting the spider with the audio-like test signal and performing measurements after 15 min intervals. It is interesting to see that the stiffness at higher displacements stays constant but the stiffness at the rest position.

More Information

Papers	W. Klippel, *"Dynamical Measurement of Loudspeaker Suspension Parts"*, presented at the 117th Convention of the Audio Engineering Society, San Francisco, October 28–31, 2004.
KLIPPEL USER'S GUIDE	Suspenion Part Measurement (SPM)
Application Notes	Separating Spider and Surround, Application Note 2 (10/2001) Adjusting the Mechanical Suspension, Application Note 3 (10/2001)

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