Dynamical Measurement of Loudspeaker Suspension Parts

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ABSTRACT
The nonlinear stiffness $K(x)$ of 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 excites the suspension parts pneumatically under similar condition as operated in the loudspeaker. The nonlinear parameters are estimated from the measured displacement and sound pressure signal. This guarantees highest precision of the results as well as simple handling and short measurement time.

1 INTRODUCTION
Transducers such as loudspeakers, headphones, shakers have a suspension realized by using a surround, spider or the diaphragm itself to centre and adjust the coil in the gap and to allow a desired displacement of the moving armature. Due to the geometry of the suspension and the material properties the stiffness $K$ is usually not constant but depends on the instantaneous displacement $x$, time $t$ (frequency $f$) and the ambient conditions (temperature, humidity). The dependency of $K(x)$ on displacement $x$ is one of the dominant nonlinearities in loudspeakers generating substantial distortion for any excitation signal below resonance.

The EIA standard RS 438 [1] 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 have been 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. Furthermore, other practical concerns (reproducibility, practical handling, time) gave reason for the development of a new dynamical method which measures suspension parts in the small and large signal domain.

2 DYNAMIC MEASUREMENT TECHNIQUE
Some loudspeaker manufacturer use for years a pneumatic excitation for suspension parts and measure the resonance frequency of the vibrating suspension. Usually a powerful loudspeaker is used for generating a sound pressure signal. In the AES standard [2] the loudspeaker cone is excited in the near field of the loudspeaker which is operated in a small panel. For the measurement of spiders the loudspeaker is usually mounted in a test enclosure The outer rim (shoulder) of the suspension part is usually firmly clamped by using rings.

Contrary to the known methods this paper suggests that the inner rim (neck) of the suspension is also clamped on a moving slide as shown in Figure 1.
This increases the moving mass $m$ significantly. The stiffness $K(x)$ and the moving mass $m$ form a resonating system. At the resonance frequency the restoring force of the suspension equals the inertia of the mass. Due to the additional mass most of inertia acts directly to the neck of the suspension. Thus the suspension is operated in a similar way as in a real loudspeaker.

The new method presented here tests suspension in vertical position to avoid any offset in the displacement due to gravity. An additional guiding rod for the slide may be used to prevent eccentric deformation of the suspension part and to suppress other vibration modes.

The nonlinear vibration of the suspension is measured and the unknown stiffness parameters are estimated by system identification techniques.

The acoustical compliance $C_{\omega}$ depends on the volume $V$ of the enclosed air and the static air pressure $p_0$.

The loudspeaker used for pneumatic excitation is modeled by an acoustical impedance $Z_D$ and a pressure source $p_D$.

The clamped suspension is described by the displacement $x$ of the inner clamping part and the driving force $F = S_p p$ which is related to the sound pressure $p$ in the test box. The driving force

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

is the sum of the restoring force $K(x)x$ of the suspension, the force $Rdx/dt$ overcoming the friction of the guiding elements and the losses in the suspension material and the inertia accelerating the total mass $m$ of the suspension part and the inner clamping part.

For sinusoidal excitation the complex ratio $H_1(j\omega)$ of the fundamentals $X_1$ and $P_1$ in the displacement and sound pressure spectrum, respectively, may be expressed as a transfer function

$$H_1(j\omega) = \frac{X_1(j\omega)}{P_1(j\omega)} = \frac{S_p}{K_{ef} (X_{peak}) + j\omega R - j\omega^2 m}$$

using the effective stiffness $K_{ef}$ which depends on the stiffness characteristic $K(x)$ and the peak displacement $X_{peak}$.

At the resonance frequency

$$\omega_R (X_{peak}) = \sqrt{\frac{K_{ef} (X_{peak})}{m}}$$

the real part vanishes and the transfer function shows a distinct maximum if the resistance $R$ is low.

3 PERFORMING THE MEASUREMENT

The loudspeaker in the enclosure is excited by a sweep signal starting at least one-third octave below resonance ending approximately one-third octave above resonance $\omega_R$.

The identification of the parameter $K(x)$ requires measurement of some state variables such as force, displacement or pressure in the system.
A direct measurement of the total driving force $F$ is not possible because all areas of the suspension contributes to this force in a different way. However, the displacement of the inner clamping parts and the sound pressure in the box can easily measured by a laser triangulation sensor and microphone mounted inside the box.

Figure 3 Open test enclosure with work bench in horizontal position

3.1 Effective Stiffness $K_{eff}$

The maximum in the transfer function $H_1(j\omega)$ reveals the effective resonance frequency $\omega_R$. Using the moving mass $m$ the effective stiffness

$$K_{eff}(X_{peak}) = \omega_R^2 m$$

(5)

can easily be calculated. Since the resonance frequency $\omega_R$ depends on the amplitude, the effective stiffness should also be understood as a function of the displacement $X_{peak}$.

The measurement of the effective stiffness can be accomplished with straightforward measurement equipment.

3.2 Nonlinear Stiffness $K(x)$

More detailed information about the properties of the suspension give the displacement varying stiffness $K(x)$. The curve can be calculated from the nonlinear distortion found in the sound pressure and displacement signal.

The balance of the forces in the resonator expressed in equation (2) is the basis for the identification of the nonlinear stiffness by using the model error equation

$$e = K(x) \cdot x + R \frac{dx}{dt} + m \frac{d^2x}{dt^2} - S_s p \cdot$$

(6)

The shape of the nonlinear $K(x)$ characteristic is estimated by straightforward optimization where the squared error in the cost function

$$C_e = \frac{1}{T} \int_0^T e(t)^2 dt \rightarrow \text{Minimum}$$

(7)

is minimized over a certain time interval $T$. To search between a wide variety of candidates for the curve shape, $K(x)$ is expressed by a truncated power series expansion

$$K(x) = \sum_{i=0}^N k_i x^i$$

(8)

Since there is a linear relationship between the unknown coefficients $k_i (i=0,1, ... N)$ and the error signal $e(t)$ the coefficients can be estimated by searching for the minimum in the cost function in a $(N+1)$-dimensional space by solving a linear set of equations

$$\frac{\partial C_e}{\partial k_i} = 0 \cdot$$

(9)

The error equation (6) still requires precise values for the additional parameters $m$, resistance $R$ and effective area $S_s$. While the moving mass $m$ can easily be measured by weighing the suspension part with inner clamping, the resistance $R$ and effective area $S_s$ can not be measured directly.

This problem can be solved by using the Two Signal Method explained in reference [3] in greater detail. For spiders and smaller sized cones the sound pressure measurement can be omitted and the simple One Signal Method [3] can be used.

4 INTERPRETATION

The stiffness $K(x)$ displayed versus displacement $x$ is the most important parameter for suspension parts. The effective stiffness $K_{eff}(X_{peak})$ are integral measures of the corresponding nonlinear parameter $K(x)$ in the used working range defined by the peak value $X_{peak}$. The effective parameters are directly related with the resonance frequency and is represented as dashed line in Figure 4. However, the effective parameters can only be compared if the
measurement are made at the same peak displacement $X_{peak}$.

![Figure 4](image)

**Figure 4** Effective stiffness $K_{eff}$ (dashed line) and nonlinear stiffness $K(x)$ of a spider

The stiffness $K(x)$ versus displacement is shown as solid line in Figure 4. For a positive displacement $x=+11$ mm the stiffness is approximately 30 times higher than at the rest position $x=0$. The symmetrical variation of the stiffness leads to dominant third- and other odd-order distortion and limits the maximal displacement. The curve in Figure 4 has also a distinct asymmetry. 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 of $-3$ mm is generated. Thus an asymmetrical shape of the $K(x)$ should be avoided because it produces high 2nd-order distortion and a dc displacement which shift the coil dynamically away from the optimal rest position and causes instabilities.

The reproducibility and repeatability of the new measurement technique has been proven in extensive testing considering the influence of the stimulus, way of clamping, additional mass, order of the power expansion, and other factors (further detail see [3]).

### 5 CONCLUSION

A new technique for measuring the most important mechanical properties of suspension parts (cones with surround, spiders) is presented which also reveals the nonlinear characteristic in the full working range.

The suspension part should be clamped during the dynamic testing in a similar way as mounted in the final loudspeaker. In some cases it may be convenient to use adhesive and original loudspeaker parts (voice coil former, frame) for clamping. However, nondestructive testing is preferred for comparing samples, storing reference units and for simplifying the communication between manufacturer and customer. An universal clamping system has been developed which fits to a large variety of centric suspensions by using a minimal number of basic elements.

The dynamic measurement technique is also convenient for the investigation of the break in and other ageing effects of the suspension.

Exploiting modern signal processing and identification techniques in combination with pneumatic excitation leads to a new measurement which provides not only repeatable and reproducible results but is also very fast, robust and simple to use.

### 6 REFERENCES


**Wolfgang Klippel** (wk@klippel.de) after graduating in speech recognition at TU Dresden in 1982 he joined a research group of a loudspeaker company, in 1987 he received a doctor/engineer degree in technical acoustics. After spending a postdoctoral year in Waterloo, Canada and working at Harman Int., CA he moved back to Dresden where he founded the Klippel GmbH www.klippel.de which develops novel kinds of control and measurements systems dedicated to loudspeakers and other transducers.

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