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.

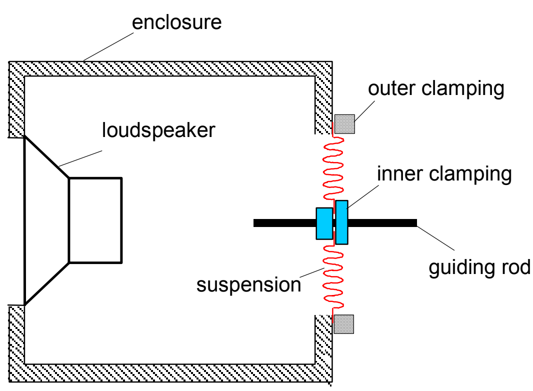

Figure 1 Setup

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.

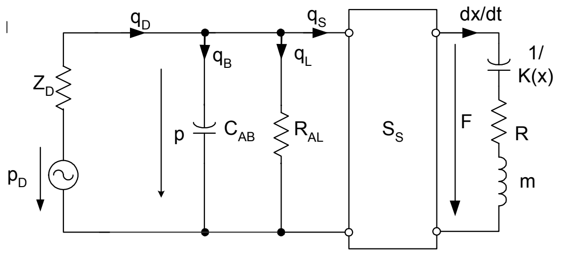

Figure 2 Equivalent Circuit

Considering the setup in Figure 1 at low frequencies where the wavelength is large in comparison to the geometry of the box the system may be modeled by the lumped parameter model shown in Figure 2. The loudspeaker generates a volume velocity q_D

$$q_D = q_B + q_L + q_S \qquad (1)$$

where 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_S produces the force F driving the suspension part under test. To get a maximal excitation of the suspension part it would be beneficial to keep the leakage of the enclosed air minimal and the resistance R_{AL} high. The majority of the leakage is caused by porosity of the suspension part under test. The pressure p in the box generates a force $F=S_S p$ on the suspension part using an effective area S_S. Due to the porosity of spider material the effective area S_S is much smaller than the geometrical area S_{geo} (S_s < 0.5 S_{geo}).

The acoustical compliance C_{AB} depends on the volume V of the enclosed air and the static air pressure p_o.

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 $F=S_s p$ which is related to the sound pressure p in the test box. The driving force

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

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

$$H_1(j\omega) = \frac{X_1(j\omega)}{P_1(j\omega)} = \frac{S_s}{K_{eff}(X_{peak}) + j\omega\, R - \omega^2 m} \qquad (3)$$

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

At the resonance frequency

$$\omega_R(X_{peak}) = \sqrt{\frac{K_{eff}(X_{peak})}{m}} \qquad (4)$$

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 ω_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 ω_R. *Using* the moving mass m the effective stiffness

$$K_{eff}(X_{peak}) = \omega_R{}^2 m \qquad (5)$$

can easily be calculated. Since the resonance frequency ω_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

$$e = K(x)\cdot x + R\frac{dx}{dt} + m\cdot\frac{d^2x}{dt^2} - S_S p \ . \qquad (6)$$

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

$$C_e = \frac{1}{T}\int_0^T e(t)^2 dt \quad \rightarrow \quad Minimum \qquad (7)$$

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

$$K(x) = \sum_{i=0}^{N} k_i x^i \ . \qquad (8)$$

Since there is a linear relationship between the unknown coefficients k_i $(i=0,1, \ldots N)$ 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

$$\frac{\partial C_e}{\partial k_i} = 0 \ . \qquad (9)$$

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}.

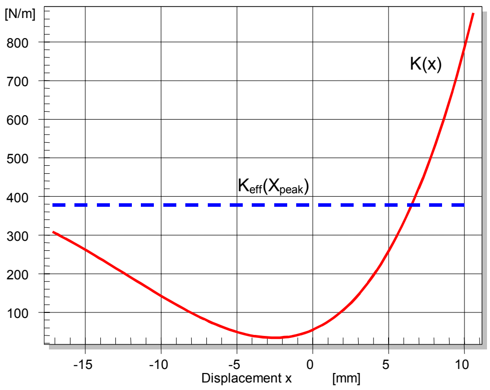

Figure 4 Effective stiffnes K_{eff} (dashed line) and nonlinear stiffness $K(x)$ of a spider

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

[1] "ANSI - Standard : Electronic Industries Association RS 438 – Loudspeaker Spiders, Test for Measuring Stiffness," Irvine, Global Engineering Documents (1976).

[2] "AES–ALMA Standard test method for audio engineering — Measurement of the lowest resonance frequency of loudspeaker cones," AES19-1992 (withdrawn 2003) (ALMA TM-100), *published by* Audio Eng. Society, (1992).

[3] W. Klippel, "Dynamical Measurement of Loudspeaker Suspension Parts, " presented at the 117th Convention of Audio Eng. Soc. in San Francisco, 2004, preprint 6179.

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