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Sound system equipment –
Electroacoustical transducers –
Dynamic measurement of suspension parts

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SOUND SYSTEM EQUIPMENT – Electroacoustical transducers – Dynamic measurement of suspension parts

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INTRODUCTION

A ready-to-use loudspeaker is the result of international division of labour. An important role is played by the suspension parts which must have reproducible behaviour for the manufacturing of equipment. This document gives a measurement method and parameters for the quality-assurance applications by suspension-part manufacturers and loudspeaker manufacturers.

The lowest resonance frequency of an electroacoustical transducer (for example, a loudspeaker) depends on the mechanical stiffness and the mass of the moving components. These elements include the mass of the diaphragm or cone, the mass of the voice coil, the mass of the air load, the stiffness of the spider and the stiffness of the surround and, if applicable, the stiffness of the air in the loudspeaker enclosure. Whereas the moving mass may be assumed as constant, the stiffness depends on the instantaneous voice coil position x, humidity, temperature and reversible and non-reversible changes versus time. The break-in effect of a new spider and natural ageing are examples of non-reversible processes. A reversible process is the reduction of the stiffness, $K(x = 0)$, at the rest position, $x = 0$, after performing a large excursion and the restoration of the original stiffness at $K(x = 0)$ after a few seconds. Closely related is the creep effect and the dependency of the stiffness, $K(f)$, on the frequency of a sinusoidal stimulus. The visco-elastic behaviour causes a discrepancy between the stiffness measured statically and dynamically. The results of a dynamic measurement technique are more relevant for the final application of suspension parts at audio frequencies.

SOUND SYSTEM EQUIPMENT –
Electroacoustical transducers –
Dynamic measurement of suspension parts

1 Scope

This PAS applies to the suspension parts of electroacoustical transducers (for example, loudspeakers). It defines the parameters and measurement method to determine the stiffness of suspension parts like spiders, surrounds, diaphragms or cones before being assembled in the transducer. The measurement results are needed for engineering design purposes and for quality control. Furthermore, this method is intended to improve the correlation of measurements between suspension-part manufacturers and loudspeaker manufacturers.

The measurement method provides the effective stiffness, K_{eff}, based on a linear model and the variation of the stiffness, $K(x)$, versus displacement x using a non-linear model. Both parameters are measured dynamically by exciting the suspension part to mechanical vibrations.

2 Terms and definitions

For the purposes of this document, the following terms and definitions apply.

2.1
inner clamp dimension, D_{I}
diameter at the neck of the suspension part which is clamped by inner clamping parts (for example, cone and cup)

2.2
outer clamp dimension, D_{o}
inner diameter of the outer rim of the suspension part which is clamped by the outer clamping parts (for example, the upper and lower clamping rings)

2.3
displacement, x
displacement measured in the perpendicular direction at the inner rim of the suspension part

2.4
driving force, F
force representing the total effect of the restoring force, friction and inertia of both the suspension part and the inner clamping parts at the neck of the suspension

2.5
transfer function, $H(f)$
function defined as the amplitude response

$$H(f) = \frac{|X(j\omega)|}{|F(j\omega)|} \qquad (1)$$

between the displacement spectrum $X(j\omega) = FT\{x(t)\}$ and the force spectrum $F(j\omega) = FT\{F(t)\}$

2.6
resonance frequency, f_{R}
frequency at which the restoring force, $F_K = K(x)x$, equals the inertia at the moving mass, m

2.7
moving mass m
mass defined as

$$m = \delta m_s + m_c \qquad (2)$$

where

m_s is the mass of the suspension part;

m_c is the additional mass of the inner clamping parts;

δ is the clamping factor (with $0 < \delta \leq 1$) describing the fraction of the suspension which contributes to the moving mass. If factor δ is not known, the moving mass is approximated by using the total weight of the suspension part ($\delta = 1$) and ensuring that the mass, m_c, of the inner clamping part dominates the moving mass, m ($m_c >> m_s$)

2.8
effective stiffness, K_{eff}
stiffness defined as

$$K_{eff}(X_{peak}) = (2\pi f_R)^2 m \qquad (3)$$

describing the conservative properties of the suspension part performing a vibration at the resonance frequency, f_R, using the moving mass, m

2.9
non-linear stiffness, K(x)
stiffness describing the dependency of the stiffness on voice coil displacement, x, while performing a vibration at the resonance frequency, f_R

2.10
loss factor, Q
factor estimated as the ratio

$$Q = \frac{H(f_R)}{H(f_0)} \qquad (4)$$

between the magnitude of the transfer function, $H(f_R)$, at resonance frequency, f_R, and the magnitude of the transfer function, $H(f_0)$, at very low frequencies, f_0 (with $f_0 << f_r$). If the losses are sufficiently high ($Q > 2$), the transfer function, $H(f)$, has a distinct maximum (peak) at the resonance frequency, f_R

2.11
peak displacement, X_{peak}
peak value of the displacement occurring during the measurement at the resonance frequency, f_R

3 Test equipment

The essential elements of the test equipment needed are as follows:

– a sine wave generator and frequency counter;
– means for exciting the suspension part (for example, pneumatically);
– outer clamping parts (for example, a pair of matched clamping rings);
– inner clamping parts (for example, a cone and a cup);

– means for determining the displacement and force at the suspension part by performing a direct (mechanical) or indirect (acoustical) measurement.

4 Test method

Both the effective stiffness, $K_{eff,}$ and the displacement varying stiffness, $K(x)$, of the suspension part are measured dynamically by performing the following steps.

a) The neck of the suspension part is clamped at the inner dimension, D_I, by using inner clamping parts (for example, a cup and a cone).

b) The total mass of the suspension + inner clamping parts is measured by using a precision weigh.

c) The outer rim of the suspension part is clamped at the outer dimension, $D_{o,}$ by using the top and bottom clamp rings. The cup is mounted on the upper side while the cone is on the lower side. It is recommended that the upper side of the suspension part which points to positive displacement be marked.

d) The suspension part is excited (for example, pneumatically) by using a sinusoidal sweep starting at $f_s = 0{,}8^*f_R$ and ending at frequency $f_e = 1{,}2^*f_R$. During the sweep, the displacement, $x(t)$, and the total driving force, $F(t)$, at the suspension part are measured versus time. The measurement of the driving force, $F(t)$, may be omitted under certain conditions (see Clause A.4).

e) The transfer function, $H(f) = X(f)/F(f)$, is calculated from the FFT displacement spectrum, $X(f) = FT\{x(t)\}$, and force spectrum, $F(f) = FT\{F(t)\}$.

f) The loss factor, Q, is determined by using equation (4). If the loss factor $Q > 2$, the resonance frequency, f_R, equals the frequency at which the transfer function, $H(f)$, has a maximum.

g) The effective stiffness, K_{eff}, is calculated by using equation (3), which is valid for the peak displacement, $X_{peak} = X(f_R)$, at resonance frequency, f_R.

h) Optionally, the non-linear stiffness, $K(x)$, may be estimated from the measured displacement time signal, $x(t)$, and force, $F(t)$, by using a non-linear system identification technique [3][1].

5 Test result

The effective stiffness, $K_{eff}(X_{peak})$, shall be reported together with the peak displacement, $X_{peak,}$ such as

$$K_{eff} = 0{,}4\ \text{Nmm}^{-1} \ @\ X_{peak} = 17\ \text{mm}$$

The clamping factor shall also be stated; if not, the default value, $\delta = 1$, is used. It is strongly recommended that the inner clamp dimension, D_i, and the outer clamp dimension, D_o, and the geometry of the inner clamping parts used be reported.

The non-linear stiffness, $K(x)$, may be reported preferably as a curve in a diagram (as shown in Figure 1) showing stiffness, $K(x)$, versus displacement, x. Positive displacement, x, corresponds with a deflection of the suspension toward the side where the cup is clamped.

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1 Figures in square brackets refer to the Bibliography.

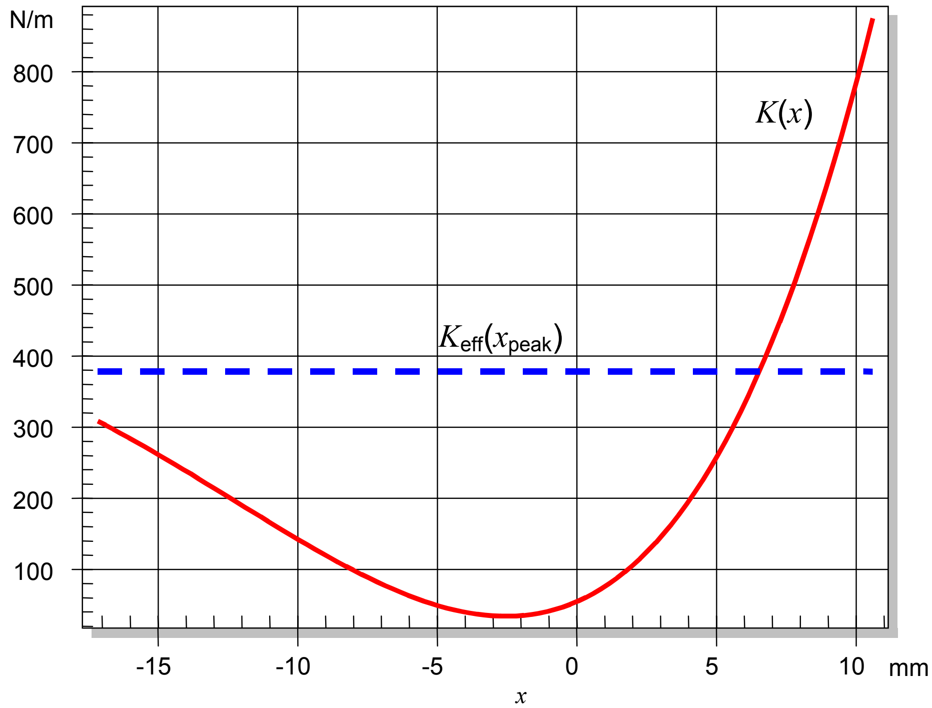

Figure 1 – Example for non-linear stiffness, $K(x)$, versus displacement, x

Alternatively, the non-linear stiffness, $K(x)$, may be reported by presenting the coefficient, k_i , of a power series expansion

$$K(x) = \sum_{i=0}^{N} k_i x^i \tag{5}$$

and the peak displacement, X_{peak}.

Annex A
(informative)

Code of practice

A.1 Clamping of the suspension part

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, non-destructive testing is preferred for comparing samples, storing reference units and for simplifying communication between manufacturer and customer. Since tooling of special clamping parts fitted to the particular geometry of the suspension is cost- and time-consuming, a more universal clamping system comprising a minimal number of basic elements (for example, rings, cups and cones) may be preferred.

The moving mass, m, depends on the mass of the moving parts of the suspension, the air load and the mass of the inner clamping parts. If the mass of the inner clamping part is much higher than the mass of the suspension, the total moving mass, m, can be approximated by the total weight of the suspension together with inner clamping parts, ($\delta = 1$). Here, the mass of the clamped areas at the outer rim of the suspension and the influence of the air load can be neglected.

The operation of the suspension part in the vertical position is not only mandatory, due to the additional mass of the inner clamping parts, but also important for larger cones where the weight of the cone material itself causes a significant offset in displacement giving a higher stiffness value if measured in the horizontal position. An additional guide for the inner clamping parts may be used to prevent eccentric deformation or tilting of the suspension and to suppress other kinds of vibration (rocking modes).

A.2 Excitation

Pneumatic excitation of the suspension part, which can be realized by using a large loudspeaker mounted in a test enclosure, as shown in Figure A.1, is recommended.

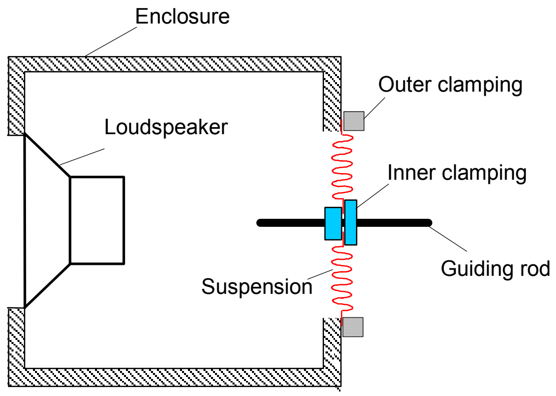

Figure A.1 – Pneumatic excitation of the suspension part

This technique allows a dynamic measurement of the suspension part vibrating at low frequencies (10 Hz < f < 30 Hz). Thus, visco-elastic effects of the suspension may be considered almost in the same way as in the final transducer.

A.3 Measurement

An optical method (for example, the laser triangulation technique) is recommended for measuring the displacement, x, of the suspension part.

If the loudspeaker is excited pneumatically, the driving force, $F(t)$, may be calculated from the sound pressure, $p(t)$, measured inside the enclosure.

A.4 Detection of the resonance frequency

The transfer function, $H(f) = X(f)/F(f)$, is calculated from the Fourier-transformed displacement and force.

If the friction of the guidance of the inner clamping parts is small, the mechanical loss factor is relatively high. ($Q > 2$), producing a distinct peak in the transfer function, $H(f)$, as shown in Figure A.2.

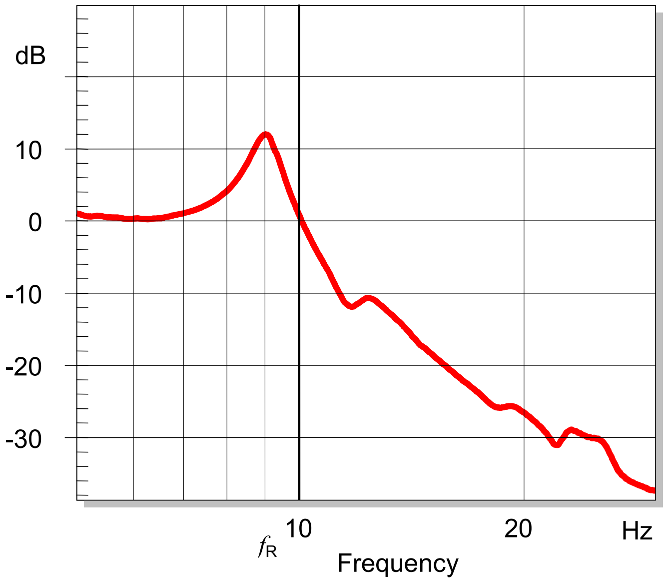

Figure A.2 – Magnitude response of the normalized transfer function, $H(f)/H(0)$, versus frequency, f

If the test enclosure used for the pneumatic excitation has a large volume and the acoustical compliance, C_{ab}, of the enclosed air is much larger than the equivalent acoustical compliance of the suspension part under test, the driving force, $F(j\omega)$, becomes almost constant and the transfer function, $H(f) \approx |X(j\omega)|$, can be approximated by the amplitude response of the measured displacement. Thus, the sound-pressure measurement may be omitted for spiders and cones with sufficiently small diameter operated in a large enclosure (D_o < 200 mm for 100 liter air volume).

A.5 Interpretation of K_{eff}

The effective stiffness, $K_{eff}(X_{peak})$, or compliance, $C_{eff}(X_{peak})$, are integral measures of the corresponding non-linear parameters, $K(x)$ and $C(x)$, in the working range used, defined by the peak value, X_{peak}. The effective parameters are directly related to the resonance frequency and may be measured with minimal equipment. However, the effective parameter can only be compared if the measurements are made at the same peak displacement, X_{peak}.

A.6 Interpretation of $K(x)$

The non-linear stiffness, $K(x)$, or compliance, $C(x)$, reveal the causes of the non-linear signal distortion generated by the suspension. This parameter, together with parameters of the motor such as force factor, $Bl(x)$, inductance, $L(x)$, are the basis for numerical prediction of the loudspeaker behaviour at high amplitudes. In this way, the maximal output and the generation of harmonic and intermodulation distortion can be simulated. For example, a symmetrical increase of the stiffness, $K(x)$, versus positive and negative excursions generates third-order and other odd-order distortion. A symmetrical increase of stiffness is desirable to some extent because it limits the maximal displacement and provides natural protection of the voice coil from hitting the back-plate. Asymmetries should always be avoided. They generate not only second- and higher order distortion but also generate a d.c. displacement which shifts the coil dynamically away from the optimal rest position and causes unstable behaviour.

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