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Assessing High-Performance Diaphragm Materials

By Steve Mowry

Ideally, a transducer will be utilized over a bandwidth of frequencies from $f0 < f < f1$ with respect to the natural frequencies of the moving assembly. Energy stored in the diaphragm is released at frequencies above $f1$, typically causing a peak in the frequency response followed by non-uniform dips and peaks or modes as frequency is increased.

There are various materials used today to implement transducer diaphragms/cones that have properties that help to push breakup out of the audio bandwidth. This is considered a valid design approach but not a simple task.

Below is a list of some very high-performance loudspeaker diaphragm materials. These materials are listed by descending speed of sound in the material, *c*m, where *c*m = $\sqrt{\frac{MODULUS}{DENSITY}} = \sqrt{\frac{E}{\rho}}\left(ms^{-1}\right)$. I will show that the first breakup frequency, $f1$, is proportional to *c*m for a given, constant diaphragm geometry.

1.) CVD DIAMOND
ELASTIC MODULUS = 1000GPa
MASS DENSITY = 3.50g/cc
TENSILE STRENGTH = 1200MPa
POISSON'S RATIO = 0.31
SPEED OF SOUND = 17,000m/s

2.) BERYLLIUM, Be
ELASTIC MODULUS = 303GPa
MASS DENSITY = 1.85g/cc
TENSILE STRENGTH = 240MPa
POISSON'S RATIO = 0.08
SPEED OF SOUND = 13,000m/s

3.) CVD SILICON CARBIDE (CERAMIC), SiC
ELASTIC MODULUS = 310GPa
MASS DENSITY = 2.80g/cc
TENSILE STRENGTH = 220MPa
POISSON's RATIO = 0.18
SPEED OF SOUND = 10,500m/s

4.) ALUMINUM/BERYLLIUM METAL, AlBeMet
ELASTIC MODULUS = 192GPa
MASS DENSITY = 2.10g/cc
TENSILE STRENGTH = 200MPa
POISSON'S RATIO = 0.14
SPEED OF SOUND = 10,000m/s

5.) ALUMINUM SILICON CARBIDE, AlSiC
ELASTIC MODULUS = 188GPa
MASS DENSITY = 3.01g/cc
TENSILE STRENGTH = 250MPa
POISSON'S RATIO = 0.24
SPEED OF SOUND = 7,900m/s

6.) RESIN-BONDED WOVEN CARBON FIBERS
ELASTIC MODULUS = 100GPa
MASS DENSITY = 1.80g/cc
TENSILE STRENGTH = 650MPa
POISSON'S RATIO = 0.3
SPEED OF SOUND = 7,500m/s

7.) RESIN-BONDED WOVEN KEVLAR FIBERS
ELASTIC MODULUS = 76GPa
MASS DENSITY = 1.40g/cc
TENSILE STRENGTH = 1240MPa
POISSON'S RATIO = 0.3
SPEED OF SOUND = 7,400m/s

8.) ALUMINUM, Al
ELASTIC MODULUS = 71GPa
MASS DENSITY = 2.77g/cc
TENSILE STRENGTH = 166MPa
POISSON'S RATIO = 0.33
SPEED OF SOUND = 5,000m/s

9.) MAGNESIUM, Mg
ELASTIC MODULUS = 44GPa
MASS DENSITY = 1.80g/cc
TENSILE STRENGTH = 160MPa
POISSON'S RATIO = 0.35
SPEED OF SOUND = 4,900m/s

10.) TITANIUM, Ti
ELASTIC MODULUS = 100GPa
MASS DENSITY = 4.85g/cc
TENSILE STRENGTH = 800MPa
POISSON'S RATIO = 0.30
SPEED OF SOUND = 4,500m/s

What's interesting is that the lower performance materials seem to be the most popular. This is cost and manufacturing capability driven. All the materials listed are very stiff and have low inner loss characteristics.

The Chemical Vapor Deposition (CVD) process for manmade diamond leads the group in speed of sound with 17,000 (ms^{-1}). The properties of the manmade diamond are as good as or better than natural diamond. A diaphragm that properly utilizes CVD diamond will inherently be of the highest performance standards.

There are a number of other interesting materials listed that are well suited for heavy-duty high performance (professional) applications as well as the high-end segment of the A/V and hi-fi markets. Kevlar does lead the group for tensile strength; however, you might expect this. Kevlar is intended to stop bullets. It must be strong. Beryllium is also a very high performance material, but nothing beats diamond.

Note that the resin-bonded woven fiber materials, carbon fiber and Kevlar, can be difficult to implement into large cone shapes for resultant isotropic material properties. Please remember the material is woven from fibers. In this case several layers of the material are utilized in a material matrix-like assembly. This is sometimes referred to as composite fiber structure.

I will use natural frequency finite element/boundary element analysis of a small direct radiating cone/dome diaphragm, a tweeter diaphragm assembly for this example of natural frequency analysis. Damping is assumed to be zero. This simplifies the analysis. Because all these materials have low inner loss tangents, zero is a reasonable approximation in this case. The results are displayed in ***Figs. 1–20***. ***Figure 0*** illustrates the moving system's fundamental resonant frequency, ƒ0.

FIGURE 0: $f_0 \approx$ 700Hz ±10% (all materials).

All models assume the material has an outside diameter of 25.0mm (1.0″) and a thickness of 50µm (0.002″) with lumped spring, mass and dashpot y-direction boundary conditions for a resultant, ƒ0˜700Hz (***Fig. 0***). The relationship between the speed of sound within the material and the frequency of ƒ1 is clearly illustrated. Each figure contains the static and displaced shape of the dome at the first axisymmetric bending vibration mode shape at ƒ1. Observe that for a small diaphragm, the material choice is a significant design consideration.

It does appear that Al, Mg, and Ti are materials that will have diaphragm resonance in band. The woven fiber materials, carbon fiber and Kevlar, may not be suitable in such small and thin diaphragm applications. The manufacturing processes may be prohibitive. However, the first five materials give excellent-to-good simulation results and could find themselves in high-performance high-frequency transducer applications.

Figure 21 illustrates a simplified circuit analog containing a nonlinear low-frequency model and a linear high-frequency model, ka > 2 and r ≥ 1, of the moving assembly of an electrodynamic audio transducer. The two modes of vibration can be identified respectively, $f_0(x) = \frac{1}{2\pi}\sqrt{\frac{K_{ms}(x)}{M_{ms}}}$ (Hz), the mechanical

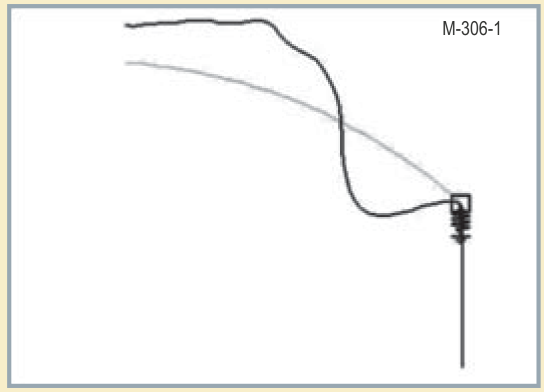

FIGURE 1: CVD diamond, f_1 = 41.5kHz.

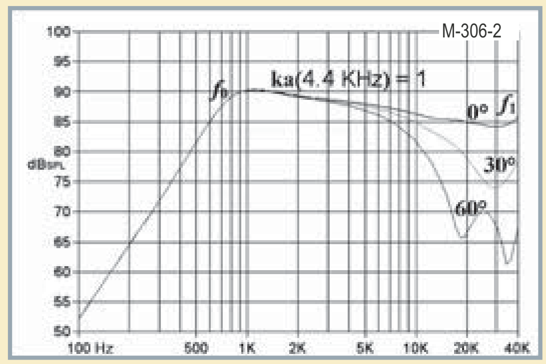

FIGURE 2: Simulation of 25mm CVD diamond dome.

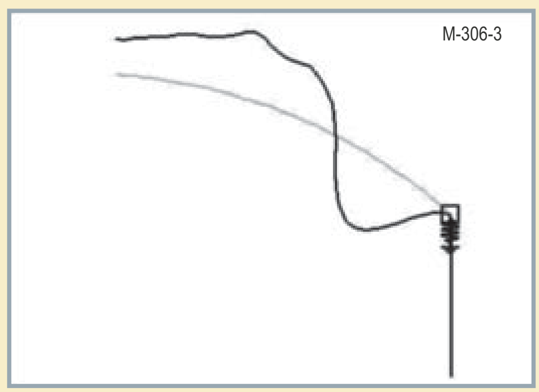

FIGURE 3: Be, f_1 = 31.7kHz.

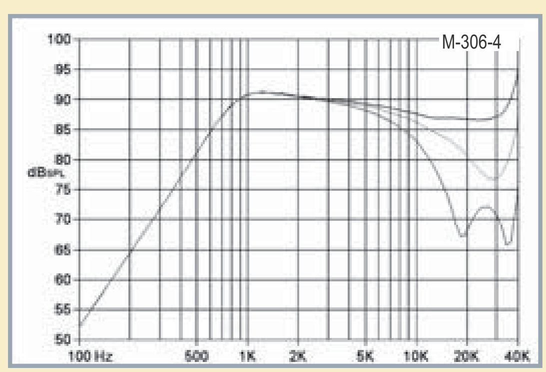

FIGURE 4: Simulation of 25mm beryllium dome.

suspension stiffness and effective moving mass including air load

related resonance frequency, and $f(ka) = \frac{kac}{2\pi a} = \frac{ka345}{\pi 0.025}$ (Hz),

the frequency at which the onset of bending occurs. This defines

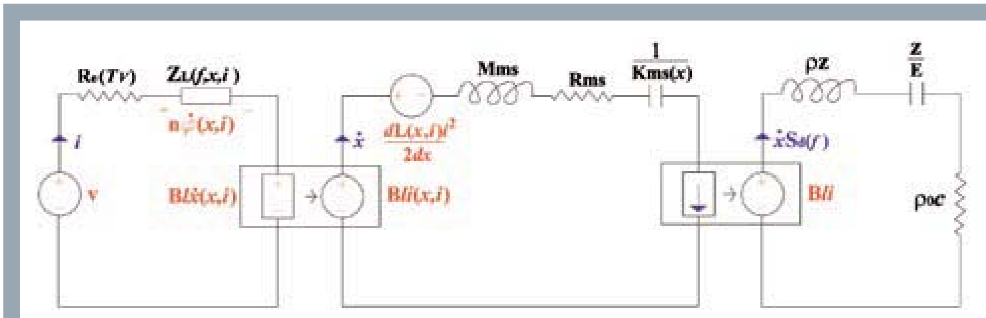

FIGURE 21: Equivalent circuit.

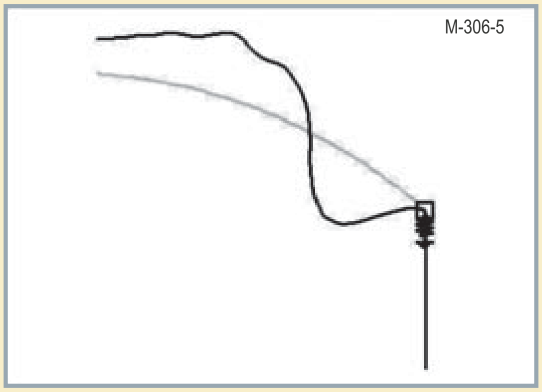

FIGURE 5: CVD SiC, f_1 = 26.0kHz.

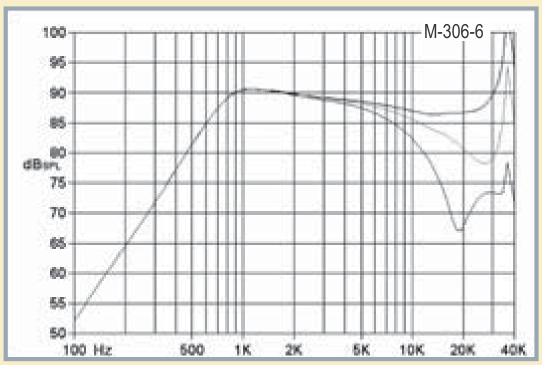

FIGURE 6: Simulation of 25mm CVD silicon carbide dome.

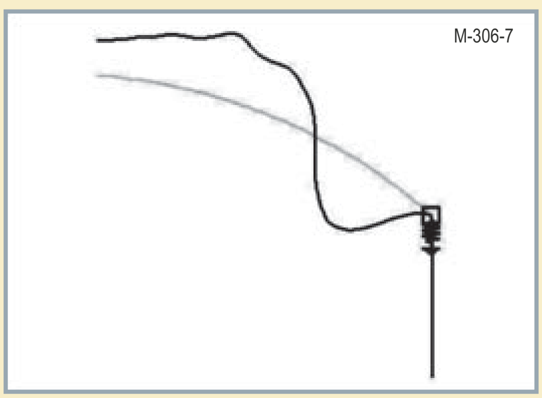

FIGURE 7: AlBeMet, f_1 = 23.6kHz.

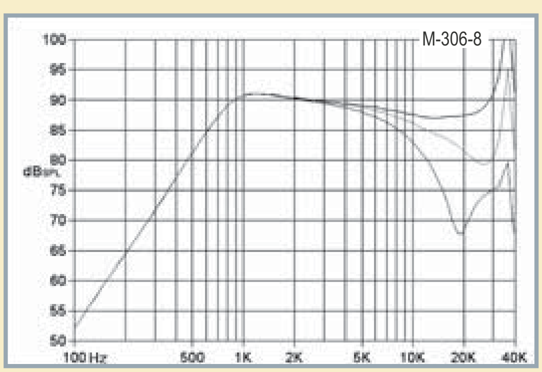

FIGURE 8: Simulation of 25mm AlBeMet dome.

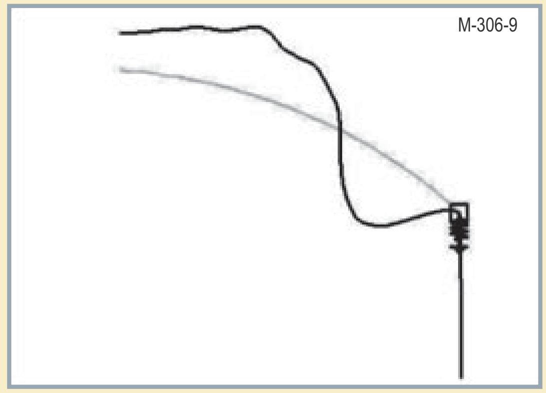

FIGURE 9: AlSiC, f_1 = 19.5kHz.

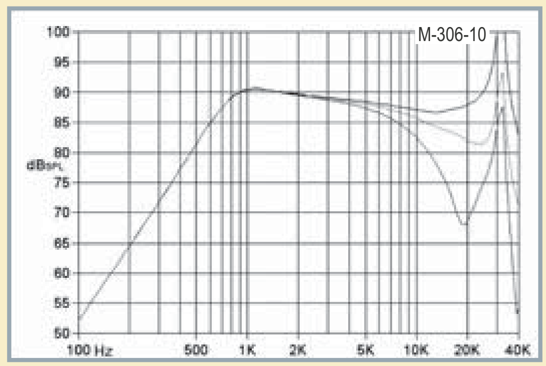

FIGURE 10: Simulation of 25mm AlSiC dome.

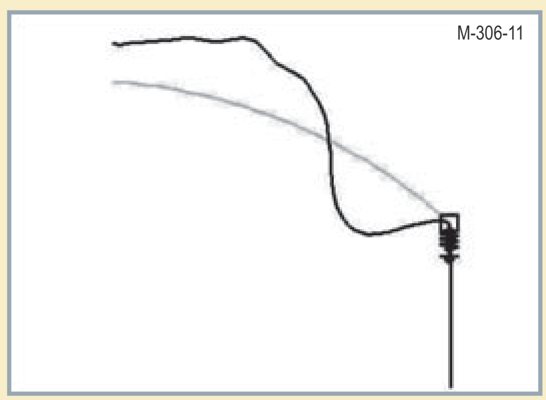

FIGURE 11: Resin-bonded woven carbon fibers, f_1 = 18.3kHz.

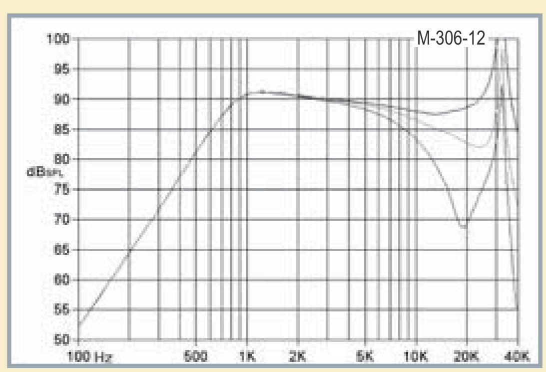

FIGURE 12: Simulation of 25mm bonded carbon fiber dome.

the first two natural frequencies, $f0$ and $f1$, where r and z are assumed to be constants with the dimension (m). r is the distance from the diaphragm, z is a one-dimensional geometry related constant, and Rms is the total mechanical loss, (kgs^{-1}). A bandwidth can now be defined as $f0 \le f \le f1$.

The current in the low-frequency model segment is the velocity, x(ms^{-1}), while the voltage drops are the forces (N) acting on the moving assembly. In the diaphragm model segment all the voltages are pressure (Nm^{-2}), and the current is the volume velocity divided by the area of a hemisphere based on the distance from the point source.

Please note that the speed of sound in air, *c*, is approximately 345 (ms^{-1}), and $\rho 0$ is the density of air, approximately 1.2 (kgm^{-3}), where $\rho 0c$ is the specific acoustic impedance, approximately 415 ($kgm^{-2}s^{-1}$). $Sd\rho 0c$ is the radiation impedance (kgs^{-1}), where Sd is the effective piston area of the diaphragm (m^2), $Sd = \pi a^2$. Finally, $2\pi r^2$ is the surface area of a

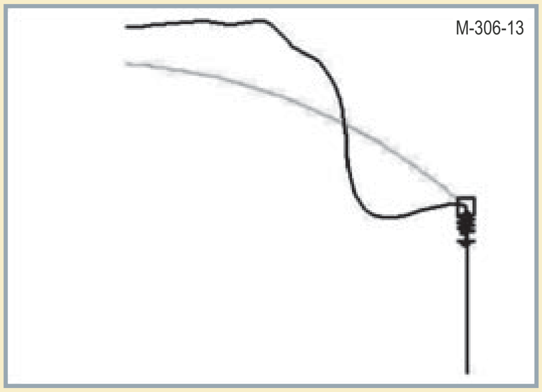

FIGURE 13: Resin-bonded woven Kevlar fibers, f_1 = 18.0kHz.

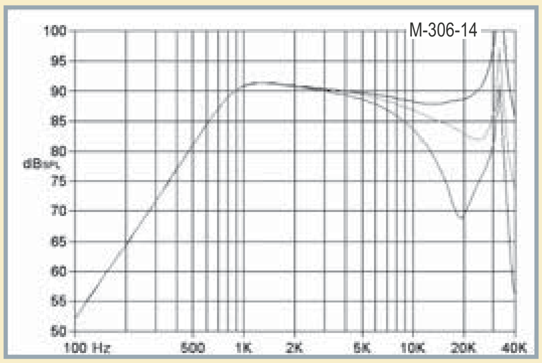

FIGURE 14: Simulation of 25mm bonded Kevlar fiber dome.

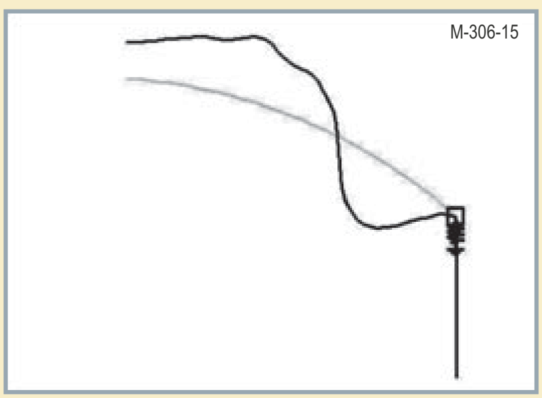

FIGURE 15: Al, f_1 = 12.4kHz.

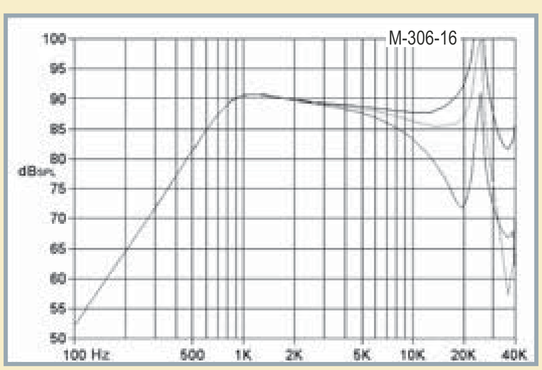

FIGURE 16: Simulation of 25mm aluminum dome.

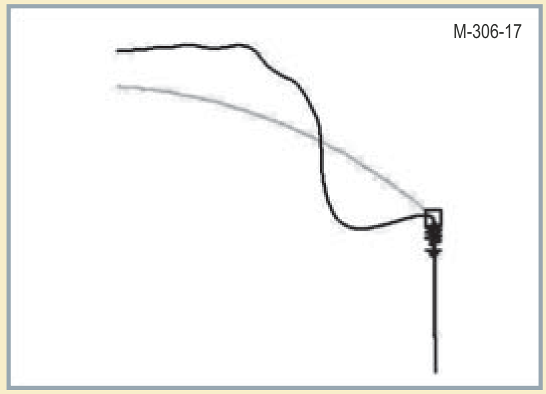

FIGURE 17: Mg, f_1 = 12.1kHz.

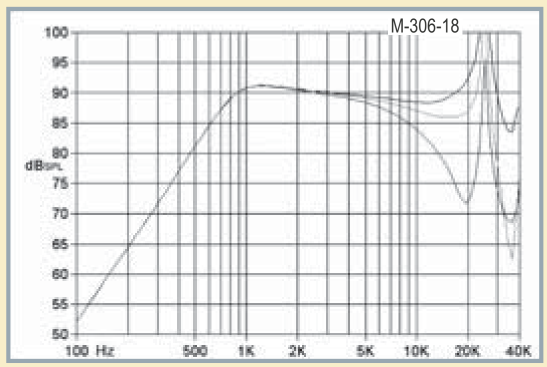

FIGURE 18: Simulation of 25mm magnesium dome.

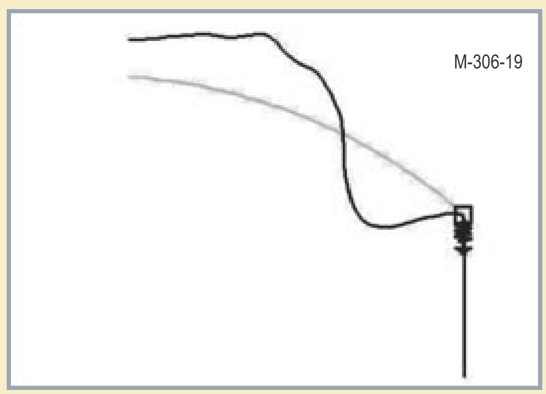

FIGURE 19: Ti, f_1 = 11.1kHz.

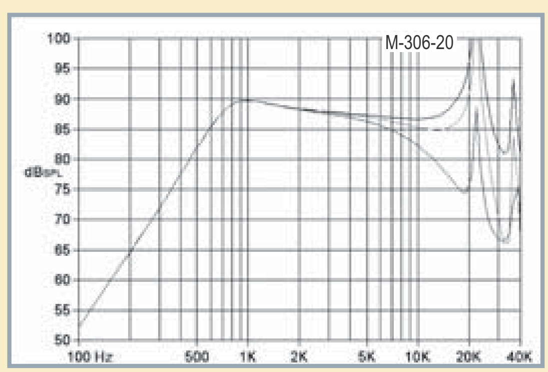

FIGURE 20: Simulation of 25mm titanium dome.

hemisphere (point source/half space).

The equivalent circuit identifies the critical design parameters. The simulations of $|P(f)|$ versus frequency are illustrated in ***Figs. 2, 4, 6, 8, 10, 12, 14, 16, 18***, and ***20***, with 2.83V input at 0°, 30°, and 60° off-axis for the respective material.

Thus, the results are essentially the same for the lumped parameter and finite element models. For each illustration in ***Figs. 1–10***, the equation $f \approx \frac{1}{2\pi\sqrt{LC}}$ holds in each case. One limitation is that without FEA/BEM, z and f cannot be identified nor can they be theoretically optimized. The lumped parameter model does provide meaningful information and clearly illustrates the dominant material properties.

Using the FEA/BEM solutions, you can approximate z in this case such that $z \approx \frac{1}{4.9\pi}$ (m) or ~65 (mm) for this problem's geometry only. With that information, you can obtain a good approximation of $f1$ for new material candidates without subsequent FEM models. The main objective of this model is to relate the lumped parameters to real-world design variables and thus decisions. There is also the limitation of ka, where k is the wave number, k = $\frac{2\pi f}{c}$ (m^{-1}), and a is the effective piston radius of the diaphragm/cone (m). Then ka is unitless. This imposes a diaphragm dimension with regards to a frequency; in this case, a 25 (mm) diameter dome, where

$$f(ka) = \frac{kac}{2\pi a} = \frac{ka345}{\pi 0.025}.$$

With regard to the investigation of materials, ka = 3.4 at 15 (kHz)! Here ka is a figure of merit for the directivity of the transducer, where ka ≤ 1.2 at 5.3kHz is considered ideal. The $|P(f,°)|$ simulations support this.

Material-related damping was assumed zero. In the model the diaphragm is damped only by the air load. With very high modulus materials as discussed within this article, this is a reasonable assumption.

MLS (maximum length sequence) measurements of the impulse response should indicate $f0$ and $f1$ and typically more including but not limited to 30°, 60°, and 90° off-axis acquisitions. You can observe detailed graphic outputs of the diaphragm behavior utilizing a laser interferometer technology measurement system in ***Fig. 22***.

Measurement of the large signal parameters is considerably more difficult. The Klippel Analyzer System (DA) is designed to acquire the large signal parameters, B*l*(*x*,*i*), L(*x*,*i*) and Kms(*x*) and is the only turnkey system available that performs this critical verification and evaluation of nonlinearity for a parameter-based design methodology, http://www.klippel.de. ***VC***

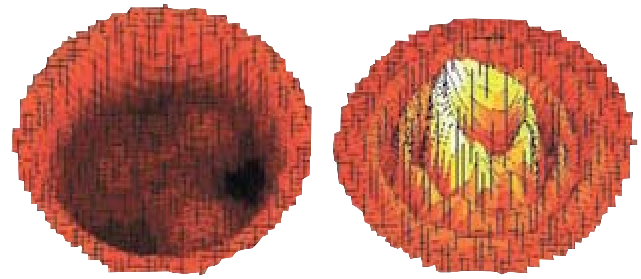

FIGURE 22: 3D graphic output from Optonar Vibromap 1000 of a diaphragm at two frequencies.

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