Lowering A Loudspeaker's Mechanical Noise Floor, Pt 1

By Mark Sanfilipo

Discover how panel behavior affects speaker box responses.

When I design a loudspeaker system for home or studio use, with the goal of maximizing sonic accuracy, I focus on a handful of key areas in judging how successful the design is. Among the top three subjective criteria are soundstaging and transparency. One of the things that I as the designer can do to maximize transparency is to lower the noise floor of the system as much as possible, given a particular design.

In a past issue of *Speaker Builder* I presented a simple, easy way to lower the electromagnetic noise floor of loudspeaker systems that make use of passive crossover networks. In this article I look at cabinet panel vibration and its contribution to the mechanical noise floor of the system and present a method to mathematically model the Insertion Loss (IL) of a cabinet's panels and apply the results to actual driver response curves.

MODEL FEATURES

The test subject I model is a totally enclosed mid-bass system, operating with a bandpass response. The low-pass -3dB point is established by a crossover network at 430Hz, with the particular driver/cabinet alignment chosen providing for a high-pass -3dB point at 40Hz.

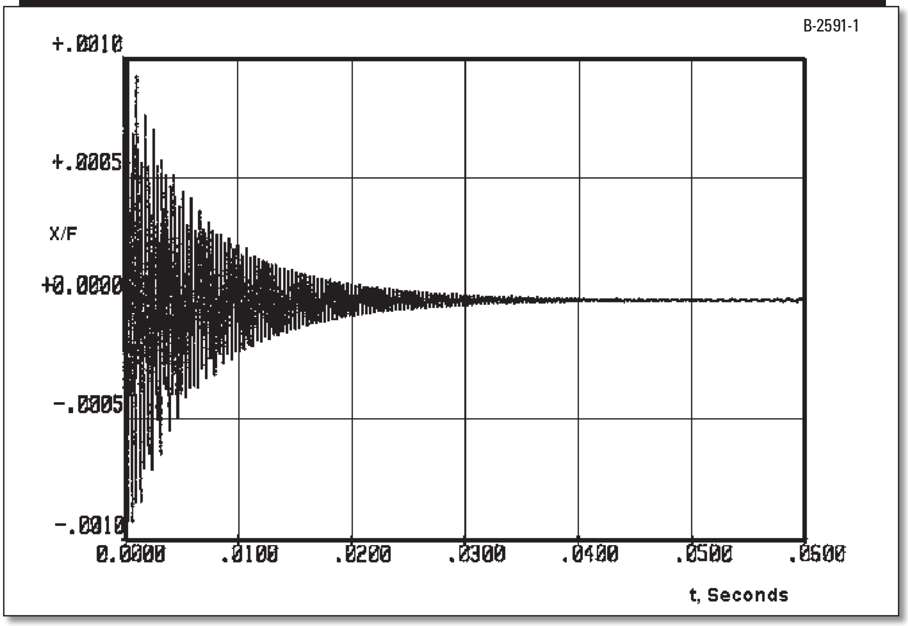
Whenever appropriate, I place each driver in its own cabinet. One of the benefits is that it usually becomes easier to identify and put in place appropriate design solutions to lower the system's mechanical noise floor. With the mid-bass cabinet's bandpass -3dB points set at 40Hz and 430Hz, my design goal becomes clearer and therefore easier to define: What panel output that can't be pushed outside the bandpass region of the system's response will be minimized to the degree that the design (and budget) will allow?

The cabinet's panels will, virtually, be built of a polymer composite. I chose the composite not only for its well-understood mechanical properties, but also because of its strength, excellent dimensional stability, and superior vibration damping properties (**Fig. 1**). Indeed, the material I used for this model has the ability to damp vibration four times faster than granite! Knuckle-rap a cabinet made of this composite and it feels—and sounds—as though you're knocking on a block of solid rock.

I could just as easily have used a wood product to construct the virtual cabinet simulations, but wood products—such as plywood—are orthotropic (its measured mechanical properties depend on the direction being considered) by nature, which would have added a needless degree of complexity to the following analyses. Compare the birch mechanical properties (**Table 1**) with that of the table outlining the mechanical properties of the composite material given in **Table 2**.

As well, if you'd like to run FEA simulations using wood products, you may find it a challenge to track down all the information (such as a full spectrum of mechanical properties) needed to run accurate, dependable simulations.

FIGURE 1: Composite sample impulse response.



Having said that, you may still find it's worth the effort, because running FEA simulations prior to actual construction can provide insight to your proposed project's mechanical and acoustical behavior, thus enabling you to make better-informed design decisions while the project is still in its formative stages.

TABLE 1: Sample Wood (Birch) Properties

Physical properties	Metric	English
Density	.55g/cc	0.0199 lb/in^3
Mechanical properties	Metric	English
Hardness, side indentation	3.3N	0.742 lb-f
Hardness, end indentation	3.6N	0.809 lb-f
Ultimate tensile strength (⊥ to grain)	6.3 MPa	914 psi
Modulus of rupture in bending	0.115 GPa	16.7 ksi
Flexural modulus	13.8 – 13.9 GPa	2000 – 2020 ksi
Flexural yield strength at elastic limit	31.7 MPa	4600 psi
Compressive yield strength, || to Grain	56 – 56.3	Mpa 8120 – 8170 psi
Compressive yield strength, ⊥ to Grain	6.7 MPa	972 psi
Poisson's ratio, µTL	0.024	0.024
Poisson's ratio, µRL	0.043	0.043
Poisson's ratio, µTR	0.426	0.426
Poisson's ratio, µLR	0.426	0.426
Poisson's ratio, µLT	0.451	0.451
Poisson's ratio, µRT	0.697	0.697
Shear strength	13 MPa	1890 psi
Toughness, radial	56.5 J	41.7 ft-lb
Toughness, tangential	70.1 J	51.7 t-lb

Panel acoustic output results from varying internal surface pressure arising from the motion of the driver's diaphragm as well as the mechanical vibration induced by the motion of the driver's frame, along with that of other panels. (The latter case is easy to envision when you consider the high degree of impedance matching that will exist between panels made of the same material, unless measures are taken to provide for an impedance mismatch between the panels; doing so will reflect vibrational energy back to the source panel.)

In either case, you can minimize panel vibration by a variety of means including isolation, absorption, a combination of both, or other such engineered solutions. In this first part of a two-article series, I'll focus directly on

TABLE 2: Mechanical Properties of Isotropic Polymer Composite

Properties	Metric Units	English Units
Density	2.32g/cm^2	0.084 lb./in^3
Compressive strength	110 – 117MPa	16,000 – 17,000psi
Tensile strength	15.2MPa	2,200psi
Flexural strength	17.2MPa	2,500psi
Young's modulus	35,982MPa	5.23E6psi
Poisson's ratio	0.23	0.23
Thermal expansion	16.9E-6 mm/mm/°C	9.4E-6 in/in/°F
Thermal conductivity	0.244W cm/°C	14.1 BTU/hr ft^2 °F/in
Specific heat	0.23 – 0.26 cal/g °C	0.23 – 0.26 BTU/lb. °F

the panels themselves and how you can minimize this sort of secondary acoustic radiation.

THE ANALYSIS

Loudspeaker cabinet panels are typically rectangular shaped and the cabinet itself can be considered—to a good first approximation—an enclosure comprising a collection of plates (defined as a construct whose lateral dimensions are much greater than its thickness), clamped at each edge. Therefore, for all analytical purposes of the following simulations I've run, I'll consider the individual panels to be clamped, rectangular plates. As previously mentioned, the vibrations that naturally set up within each panel arise from both airborne pressure fluctuations produced on the interior panel surface by the action of the driver's diaphragm as well as structural-borne mechanical vibration set up as a reaction to vibration of the driver's frame.

To understand the basics of panel resonance as well as panel acoustic output, it's useful to resort to a lumped-parameter, acoustic-equivalent circuit modeling driver/cabinet interaction (**Fig. 2**). The usual caveats for lumped-parameter simulation are applicable: driver operating in piston mode, all cabinet dimensions are smaller than the wavelengths of interest, and so on,

where:

Pas = acoustic pressure generator

Zas = acoustic equivalent of loudspeaker impedance

$Mw_{1,2,3}$ . . . = equivalent acoustic mass of individual cabinet panels and air load

$Cw_{1,2,3}$ . . . = equivalent acoustic compliance of individual cabinet panels

$Rw_{1,2,3}$ . . . = equivalent acoustic resistance of individual cabinet panels owing to damping

Cab = acoustic compliance of air in enclosure

Cab' = added compliance owing to cabinet filling and/or lining with fibrous material

Rar = acoustic radiation resistance of air load on the front of the cone and box wall(s)

Ralb = leakage around box compliance

Rab = acoustic resistance of box losses owing to absorption

Below the fundamental resonance (the frequency at which Cw resonates with Mw), the effect of Cw provides for the primary control of the circuit's impedance. At resonance, impedance is controlled by Rw, the panel's equivalent acoustic resistance owing to damping. At resonance the Mw-Lw-Cw circuit effectively shunts the Rar-Cab-Cab'-Ralb-Rab branch of the circuit, causing a dip in response. Just above resonance, where Mw resonates with the Cab branch of the circuit, the volume velocity through Rar is increased and you get a peak in the response plot.

In both cases (interestingly enough) a large enough Rw value would minimize the shunting effect, in turn minimizing the magnitude of the response peak or dip. The best practice, therefore, is to use a material with as high an Rw value as the project budget allows.

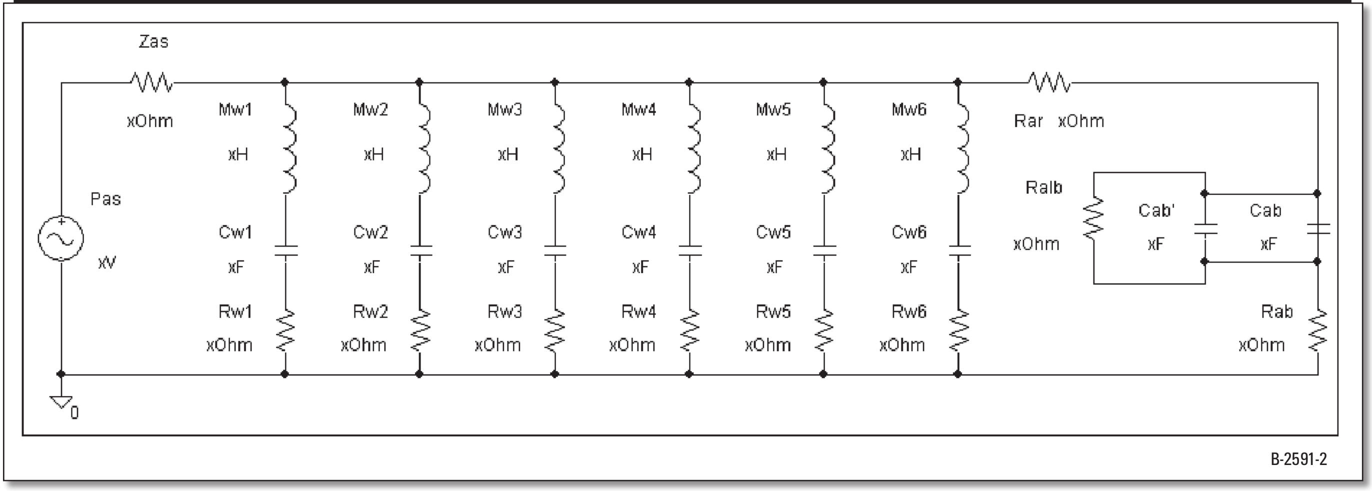
Any panel—whether simply supported or clamped at any (or every) edge—will exhibit a clearly defined, vibrational mode series known as eigenvalue, which are theoretically infinite in number and, in general, are not harmonically related. To demonstrate this vibrational series, I ran a FEA (finite element analysis) simulation, modeling a rectangular panel, clamped at all four edges, with the following physical dimensions:

Dimension	Named Variable	Value
Length	a	20″ (.508m)
Width	b	10″ (.254m)
Thickness	h	.75″ (.01905m)

In general, when working up a simulation series, I tend to model only the first 30 or so modes, because the resonance frequencies that occur above that range, as I already mentioned, generally contribute little to the system's total acoustic output. Phase-cancellation, owing to different segments of the panel moving in opposite directions and diminishing panel volume-velocity, all add up to a lowered panel contribution to total system acoustic output, when the panel is operating in the upper reaches of the mode series.

Figures 3-6 show the results, in graphical form, illustrating the magnitude of peak panel displacement at various mode numbers. The panel displacement,

FIGURE 2: Driver/cabinet interaction circuit.



as modeled for each simulation series, is exaggerated in appearance for the sake of visual clarity.

It's clear from this series of models that rectangular panels do exhibit resonance modes, each with a distinct pattern of panel deformation. Indeed, note that different sections of the panel can, at the same moment, be moving in opposite directions.

In general, the higher modes will move different areas of the panel in opposite directions, effectively diminishing the net acoustic radiation of the panel owing to the diminished net panel area capable of producing the acoustic radiation in the first place. In other words, phase cancellation, aris-

FIGURE 3: Panel displacement. Mode 1 = 67.778Hz.

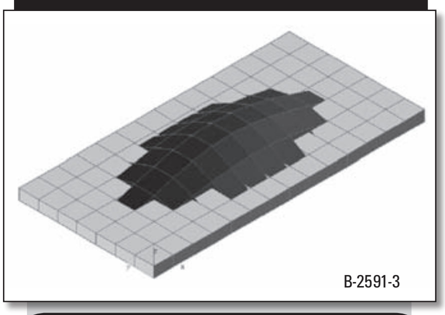


FIGURE 4: Mode 4 = 85Hz.

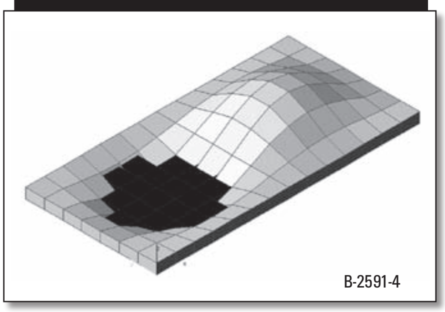


FIGURE 5: Mode 9 = 176Hz.

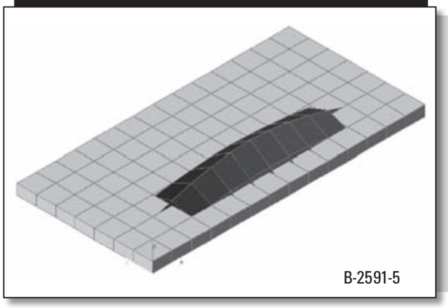
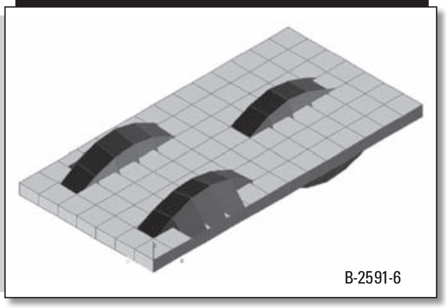


FIGURE 6: Mode 27 = 344Hz.



ing from different panel areas moving in opposite directions, limits the net amount of total acoustic radiation emitted. Add to this the fact that higher order modes are easier to damp, and you see these self-same modes are generally not in a position to affect system response to the degree the lower order modes can.

RESPONSE FACTORS

To deal with these problematic lower resonance modes, the best practice of the heavy-panel school of resonance control (as opposed to the light-panel school, which advocates using materials such as Aerolam) requires using panels of appropriate thickness, made of a stiff material exhibiting a high degree of internal damping. Further reductions result from shifting the fundamental resonance frequency of the panel high enough so that it no longer presents itself within the passband portion of the system's response spectrum, in this case 40Hz to 430Hz. In past projects, I've found cross-bracing cabinet panels—usually using a material stiffer than that from which the panels are made—a useful and effective means for shifting these natural panel resonance frequencies upward.

I've run some FEA simulations to look at the effects of panel cross-bracing. I re-ran the above simulation, this time modeling a series of bracing techniques. For the first simulation, I braced the panel vertically (across the mid-section of the panel parallel to the front-edge Y-axis, with the results in **Figs. 7-10**).

Bracing the panel vertically produced a series of modes clustered around 96Hz. This upward shift of the fundamental resonance is a step in the right direction, but not nearly high enough for my purposes.

I then modeled the panel braced horizontally, with the brace located at .5y, parallel to the x-axis and running the full length of the panel. The results are shown in **Figs. 11-15**.

In this series of simulations the upward shift of the natural panel resonance frequencies is quite apparent, producing superior results to that of the vertically positioned brace. For a cabinet to be used for the reproduction of the mid-bass portion of the total system response, this is again a step in the right direction, but still not high enough for my purposes.

In actual practice I use multiple horizontal braces for a mid-bass system of this size. Modeling two braces located approximately one-third and two-thirds up the y-axis (front) edge of the panel and running to the back edge of the panel produces the results

FIGURE 7: Vertical bracing. Mode 1 = 96Hz.

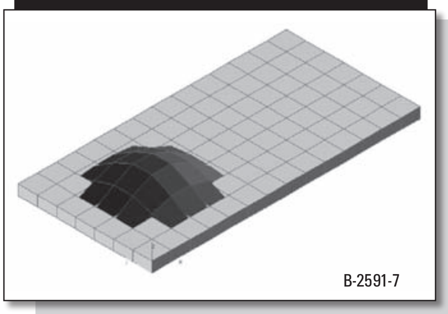


FIGURE 9: Mode 3 = 96Hz.

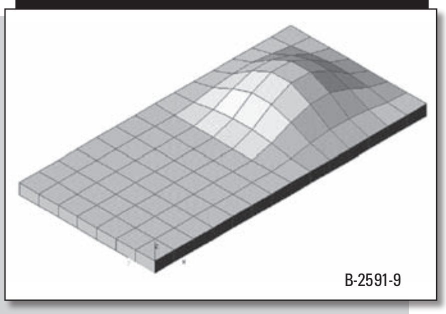


FIGURE 8: Mode 2 = 96Hz.

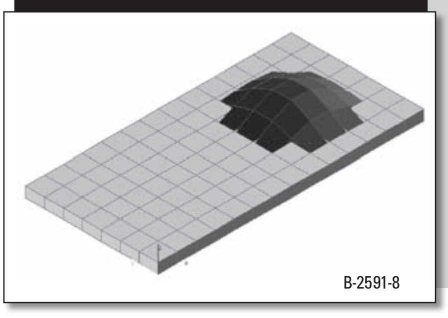


FIGURE 10: Mode 4 = 96Hz.

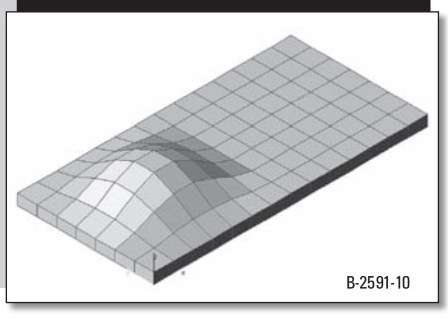


FIGURE 11: Horizontal bracing. Mode 1 = 195Hz.

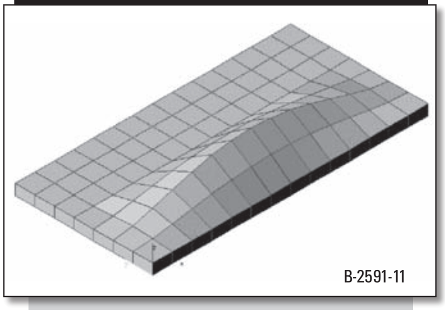


FIGURE 12: Mode 3 = 203Hz.

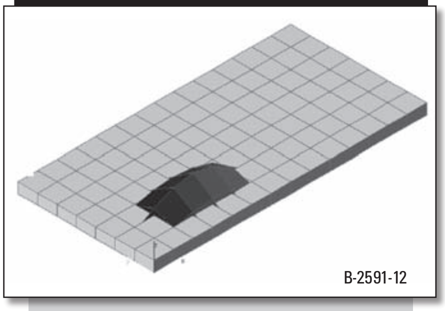


FIGURE 13: Mode 4 = 203Hz.

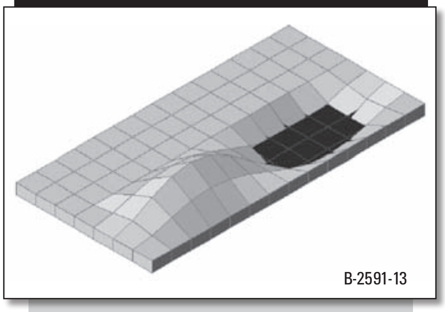


FIGURE 14: Mode 10 = 291Hz.

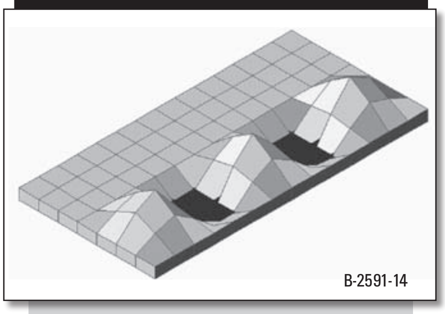


FIGURE 15: Mode 15 = 349Hz.

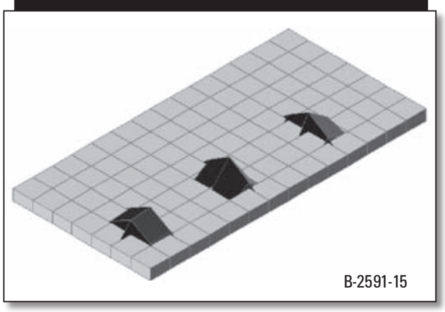


FIGURE 16: Two braces. Mode 1 = 344Hz.

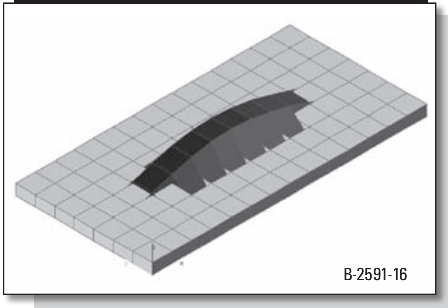


FIGURE 17: Mode 4 = 345Hz.

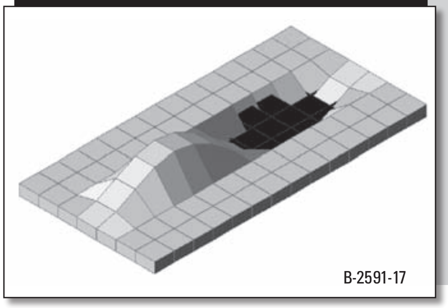


FIGURE 18: Mode 10 = 387Hz.

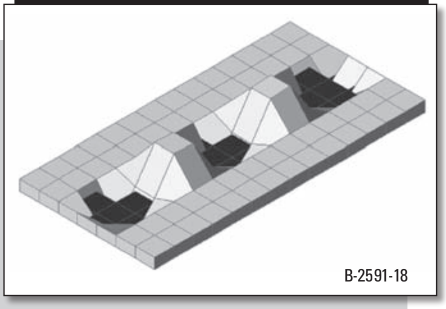
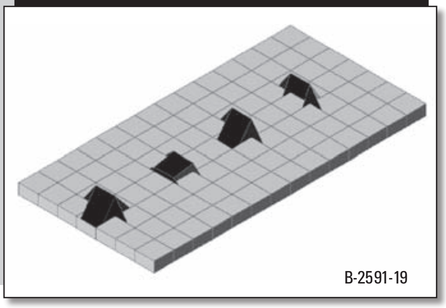


FIGURE 19: Mode 15 = 542Hz.

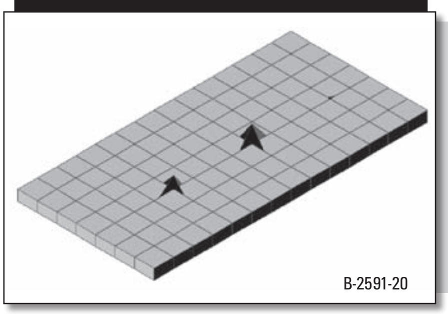


in **Figs. 16-19**.

The frequency at which the fundamental natural panel resonance occurs is now in the neighborhood of the 430Hz low-pass point. However, I'd like to see the mode series pushed higher. Modeling three braces, placed at .25y, .5y, and .75y and running the full length of the panel from front to back produces the results in **Fig. 20**.

The first mode of vibration is now above the low-pass cutoff frequency, but only by about a half octave. I could add another brace, which would certainly push the mode series higher. Because I already know, by virtue of the preceding simulations, that bracing works well, I chose at this point in the series instead to re-run the simulation, keeping the three braces in place, but now doubling the thickness of the panel to 38mm (1.5″). Doing so produced a first mode resonance frequency of 701Hz (**Fig. 21**).

FIGURE 20: Three braces. Mode 1 = 658Hz.



Not bad. Doubling the pane thickness again produced the result in **Fig. 22**. Here you see the fundamental mode frequency increase a bit to 720Hz. There are, of course, practical limits to how much bracing you can employ or how thick panel walls can be made in a design.

Note that both methods used—brac-

FIGURE 21: Three braces, panel thickness doubled. Mode 1 = 701Hz.

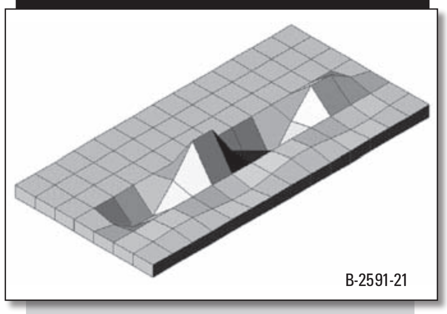
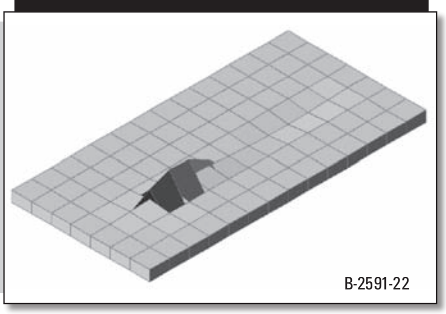


FIGURE 22: Three braces, panel thickness quadrupled. Mode 1 = 720Hz.



ing and increasing panel thickness—also increase Rw, thus increasing damping. There also exists another approach to panel resonance control worth mentioning: constructing the cabinet in such a way as to stagger the resonance mode frequencies, so that no two panels exhibit the same acoustic behavior.

Thus far you have seen that by using the appropriate panel material, with high enough damping, properly braced and of the correct thickness, a designer can effectively shift the mode series high enough and/or otherwise minimize the effect panel resonances can have on final system output and do so to the point they no longer pose response problems for the system. Sometimes, though, shifting panel resonances upward can be counter-productive. For example, if doing so shifts them into a portion of the response spectrum where they will have a deleterious effect on the system's output, doing more harm than good, then shifting them downward, or staggering the response aberrations is called for.

In a case where the techniques illustrated thus far don't sufficiently resolve panel resonance issues, there are other means you can try. For example, resorting to strategically located constrained layer damping CLD panels (typically a metal sandwich made up of a base layer and constraining layer, bonded by a visco-elastic material filling layer), which dissipates vibration energy via sheer deformation, can be very effective. Extensional damping (adding a single layer of visco-elastic material to the inside of the panels), which dissipates vibration energy by converting it into heat by way of friction arising due to material flexure, works well but depends upon the amount of flexure it is exposed to in order to do what it does—and the amount of flexure it's exposed to at any given moment may be very, very small. This works well against impact noise. Of course, simply choosing a different material better suited to your design requirements is another appropriate alternative.

THE MATHS

The classic equation for determining the resonance frequencies of a uniform, isotropic, rectangular panel, with boundary conditions set as all edges, simply supported (ss), is:

$$f_{ss} = \frac{1}{2*\pi} * \sqrt{(\frac{D}{\rho})} * \left[(\frac{m\pi}{a})^2 + (\frac{n\pi}{b})^2 \right]$$

and for the same panel with boundary conditions set as all edges clamped (c), the fundamental resonance frequency can be found by:

$$f_c = \frac{12}{2*\pi} * \sqrt{(\frac{D}{\rho})} * \sqrt{\frac{7}{2} * \left(\frac{1}{a^4} + \frac{4}{7} * \frac{1}{a^2 * b^2} + \frac{1}{b^4} \right)}$$

where:

f = Resonance frequency, Hz

$D = \frac{E * h^3}{12 * (1 - \eta^2)}$, flexural rigidity or bending stiffness of plate

m, n = mode numbers, integers

ρ = panel mass per unit area

E = Young's modulus, modulus of elasticity

h = panel thickness

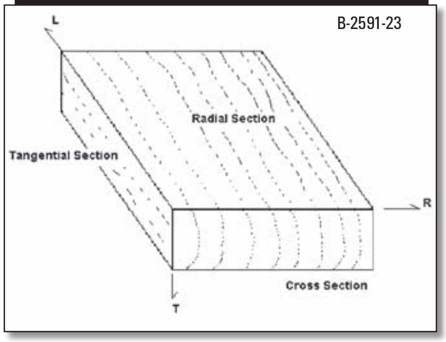
a = panel length

b = panel width

η = Poisson's ratio (ratio of orthogonal straining)

A few caveats about these equations are in order. First, note the phrase "isotropic," which means a material that exhibits mechanical properties (such as Young's modulus) that are independent of which direction the measurements are taken from which the mechanical properties are calculated. A good example of an isotropic material is MDF or, as mentioned previously, a polymer composite material (**Fig. 23**). A material such as plywood would not be considered isotropic, but rather orthotropic, which means (as you've likely already guessed) its measured mechan-

FIGURE 23: Direction affects measured box material properties.



ical properties depend on the direction (laterally, transversely, or radially) being considered.

A second caveat is: When you start cutting openings in the panel, the formulas are no longer applicable, and measured versus calculated resonance values will differ substantially. This doesn't mean the formulas aren't useful in speaker building, they are. But I would use them only as a starting point for working up a rough estimate of panel resonance frequency for those panels that don't have openings cut into them and only if I were using an isotropic material such as MDF or a polymer composite.

As I mentioned earlier, the contribution to total system acoustic output made by the panels does not arise solely from the mechanical vibrations set up in each panel by the motion of the driver's frame and other adjacent panels. You need to consider another factor: the outward transmission by the outer side of a panel due to fluctuating pressure incident to the internal side of a cabinet panel. For the following analysis, again assume that the panels behave as plates, clamped at each edge and under uniform load by the driver diaphragm-induced acoustic pressure fluctuations.

In any case, the internal pressure fluctuations found within a loudspeaker cabinet under normal operating conditions aren't very large to begin with and further decrease as frequency increases. The internal box pressure produced by an operational driver bolted into one panel of the cabinet can be found, to a good first approximation, by:

$$P_B = \frac{\sqrt{\frac{P_A * 2 * \pi * c}{\rho_0} * \frac{\rho_0 * c^2}{V_B}}}{(2 * \pi * f)^2}$$

where:
P_B = Rms box pressure, pascals
P_A = radiated acoustic power, watts
c = velocity of sound in air, ~345 m/s
ρo = density of air, 1.18 kg/m^3
V_B = net internal box volume, liters
f = frequency, Hz

Figure 24 shows the results of a series of simulations, holding P_A constant at .1W and varying V_B, using the above equation.

One common mathematical method used for expressing the degree to which a panel can reduce the outward expression of these internal pressure fluctuations is by transmission loss, which, in dB, is expressed mathematically by:

$$TL = 20 * \mathrm{Log}_{10}\left(\frac{1}{\tau}\right)$$

where τ, the transmission loss coefficient, is expressed by:

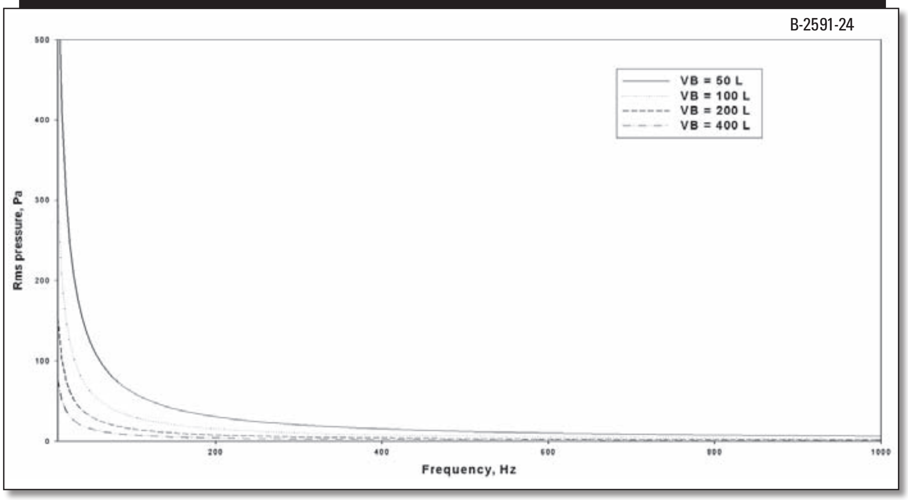
$$\tau = \frac{|P_i|^2}{|P_t|^2}$$

P_i = incident pressure
P_t = transmitted pressure

Factoring in panel mass and frequency, the TL equation then becomes:

$$TL = 10 * \mathrm{Log}_{10}\left(\left(\frac{m * \omega}{2 * z}\right)^2 + 1\right)$$

FIGURE 24: Steady-state RMS pressure vs. frequency.



where:
m = panel surface density, $(\rho * h)$
ρ = material density
h = panel thickness
Ω = radian frequency
z = panel material impedance

Taking into account m, and solving for B, the bulk modulus or effective stiffness of the panel is found by:

$$B = \frac{E_{Panel} * h^3}{12 * \left(1 - \eta_{Panel}^2\right)}$$

And from the results of the preceding equation you can now calculate the panel's critical frequency:

$$F_c = \frac{c^2}{2 * \pi} * \sqrt{\frac{m}{B}}$$

Finally, you can calculate the transmission loss slope below $F_c/2$:

$$TL = 20 * Log_{10}\left[\left(\frac{2 * \pi * \left(\frac{F_c}{2}\right) * F_c * m}{2 * \rho * c}\right)^2\right] - 5$$

And the slope above F_c:

$$TL = 10 * Log_{10}\left(\left(\frac{2 * \pi * F_c * m}{2 * \rho * c}\right)^2\right) + 10 * Log_{10}\left(\frac{2 * \eta}{\pi}\right)$$

Once you have calculated the TL values, assessing the effective reduction of transiting acoustical energy by the panel becomes a simple matter of subtracting TL from the dB SPL level of the source—in this case, those levels in the immediate vicinity of the cabinet-internal side of the panel.

These expressions are, of course, panel-focused, and the majority of transmission loss theory found in the literature—upon which these equations are based—is for wall-sized (or larger) "panels" dividing room-sized "enclosures" sporting ideal (or completely lacking) boundary conditions. The results are not necessarily applicable to panels of the material type dimensions and boundary conditions commonly found in loudspeaker enclosures.

INSERTION LOSS

Searching for a better enclosure-focused approach to characterize the outward re-radiation of acoustic energy by an enclosure's panels leads to the concept of insertion loss. Mathematically, the general expression of insertion loss (IL) is defined as:

$$IL = Log_{10}\left(\left(\frac{SPL_{External}}{SPL_{Internal}}\right)^2\right)$$

where:
$SPL_{External}$ = acoustic sound pressure level, external to cabinet panel
$SPL_{Internal}$ = acoustic sound pressure level, internal to cabinet panel

The equation is instructive but nevertheless descriptive in nature.

Fortunately, there exists a predictive mathematical expression for determining the insertion loss of a loudspeaker cabinet panel that takes into account various factors such as enclosure volume impedance, panel mechanical impedance, and so forth. For a close-fitting cabinet (one in which the distance from internal sound source to any panel is 1m or less), with clamped boundary conditions, IL can be mathematically stated by:

$$IL = 10 * Log_{10}\left[\left(Cos(k * d) + \left(\frac{\pi^2}{4 * K * \omega * \rho_0 * c}\right) * Sin(k * d)\right)^2\right]$$

where:
k = acoustic wave number, $\frac{2 * \pi}{\lambda}$
λ = wavelength
d = distance from source to panel
ρ_0 = density of air
c = speed of sound in air
and

$$K = \frac{1.35}{\left[3.86 * D_i\left(\frac{129.6}{a^4} + \frac{78.4}{a^2 * b^2} + \frac{129.6}{b^4}\right) - \omega^2 * \rho * h\right]}$$

where:
a = panel length
b = panel width
h = panel thickness
ρ = density of panel material
Ω = frequency of interest
and

$$D_i = \left(\frac{E * h^3}{12 * (1 - \upsilon^2)}\right) * (1 + i\eta)$$

which is the complex bulk modulus of the panel material
η = internal damping coefficient of the panel material
E = Young's modulus or modulus of

elasticity
ν = Poisson's ratio (ratio of orthogonal straining)

For a polymer composite panel with the material properties as listed in **Table 2** and with the dimensions: a = 20″ (.508m), b = 10″ (.254m), and h = .75″ (.01905m), you can calculate D_i, K, and finally IL.

Figure 25 shows the IL graph. Here

FIGURE 25: Insertion loss vs. frequency.

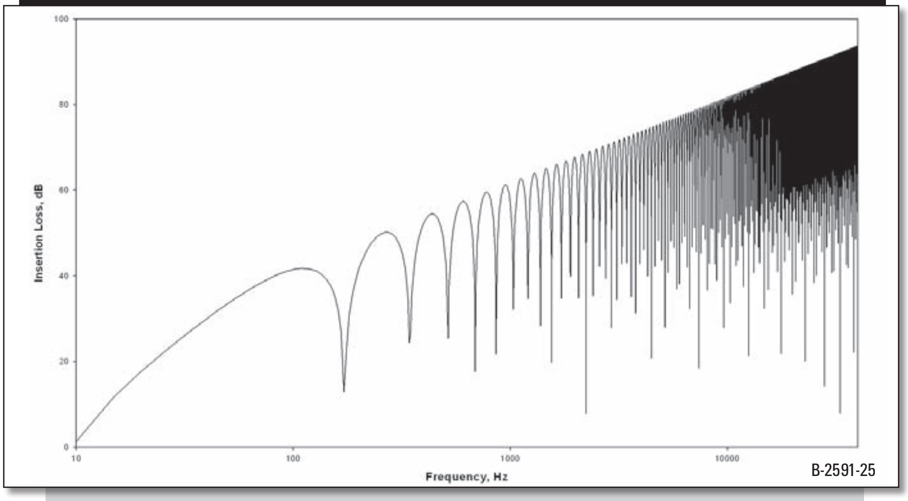


FIGURE 26: Varying panel length.

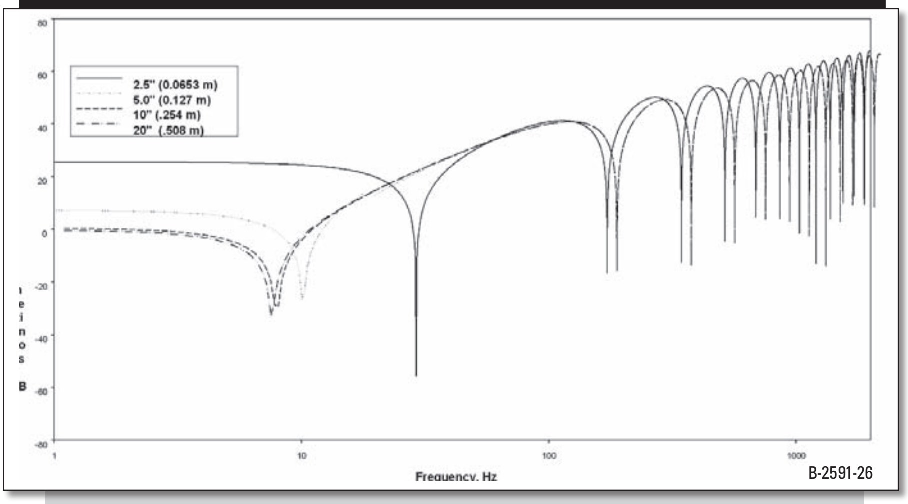
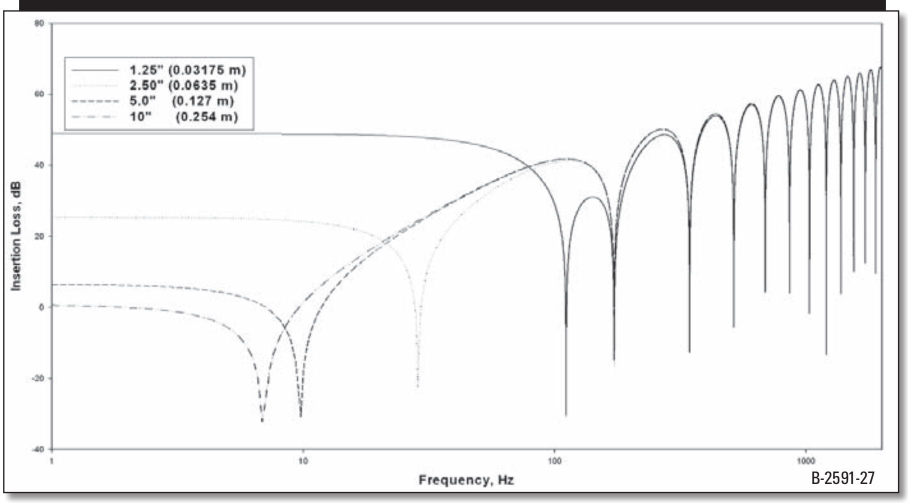


FIGURE 27: Varying panel width.



you see a series of nulls alternating between peaks in IL values. It's clear the responses of a cabinet's panels depend on both the frequency of the various excited modes and the frequency generated by the driver enclosed within the cabinet. As well, the IL response at higher frequencies is controlled by various material properties along with the distance between the panel and driver. When panel vibration is in phase with the incident pressure wave, the panel becomes acoustically "invisible" to the incident pressure wave. Standing waves within the enclosure have the characteristic values of:

$$\omega_x = \frac{c * \pi}{lx} * n_x, \omega_y = \frac{c * \pi}{l_y} * n_y, \omega_z = \frac{c * \pi}{l_z} * n_z$$

and frequencies can be approximated by:

$$f = \frac{c}{2} * \sqrt{\left(\frac{n_x}{l_x}\right)^2 + \left(\frac{n_y}{l_y}\right)^2 + \left(\frac{n_z}{l_z}\right)^2}$$

where:
c = speed of sound in air,
nx, ny, nz = 0, 1, 2, 3 . . .
lx, ly, lz = internal length, width depth

These standing waves can produce acoustic wave pressures greater than that generated by the driver itself. This can in turn, under certain conditions—such as when panel vibration is in phase with the incident pressure wave—result in panel vibration of a magnitude sufficient to be considered a sort of negative insertion loss.

Curious to see how varying the values of various parameters would affect the IL dB levels, I ran a detailed series of simulations. First I varied panel length (**Fig. 26**). Then I varied panel width (**Fig. 27**). Varying length and width, individually, did indeed alter the IL response spectrum. I then varied length and width simultaneously (**Fig. 28**). In all three cases, as the panel became smaller, you see an increase in the frequency of the first null as well as an increase in IL.

I then varied panel density (**Fig. 29**). Across the IL response spectrum you see that increasing density increases the amount of insertion loss. All else being equal, you can thus expect that the denser a given panel's material, the quieter it will be.

Then, I varied panel thickness (**Fig.

FIGURE 28: Varying panel length and width.

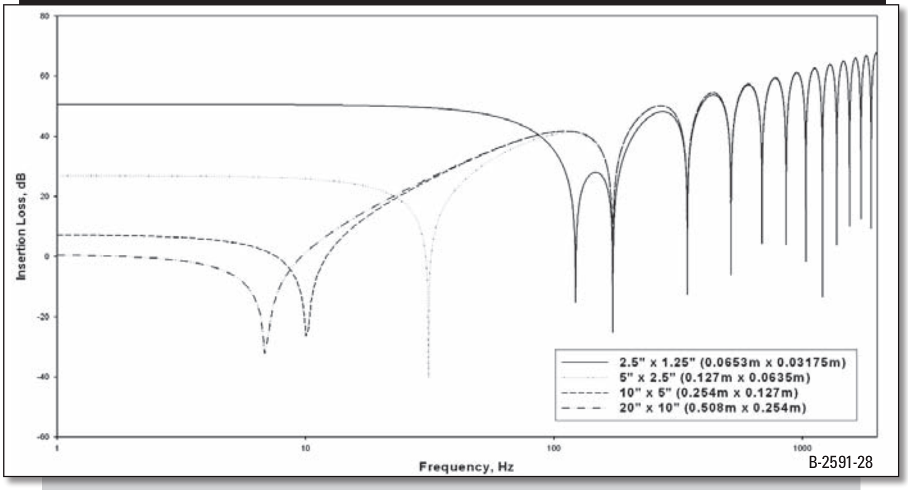


FIGURE 29: Varying panel density.

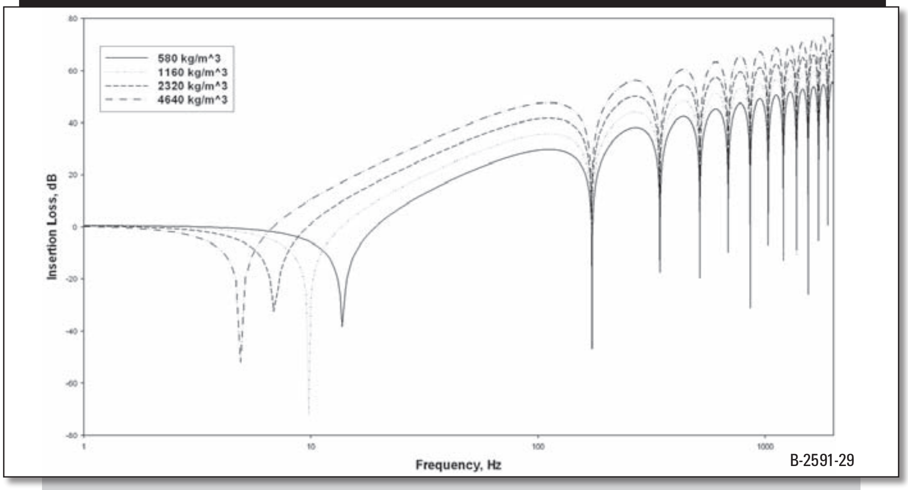
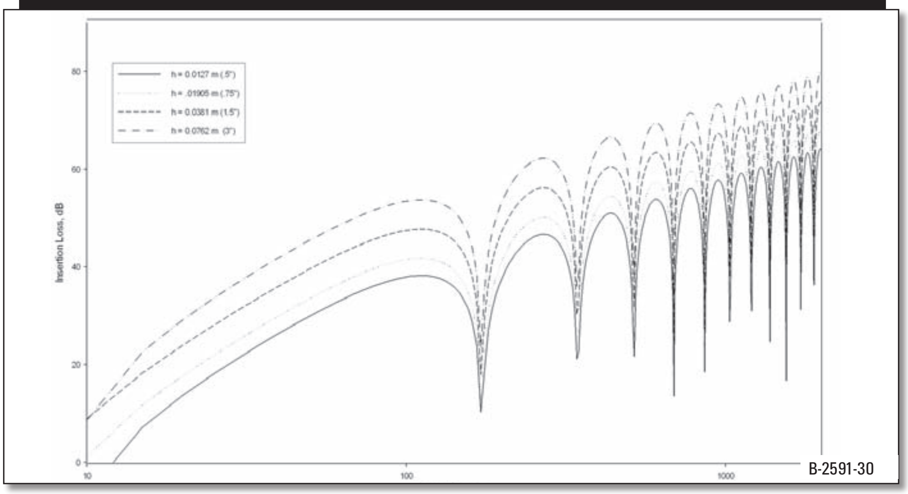


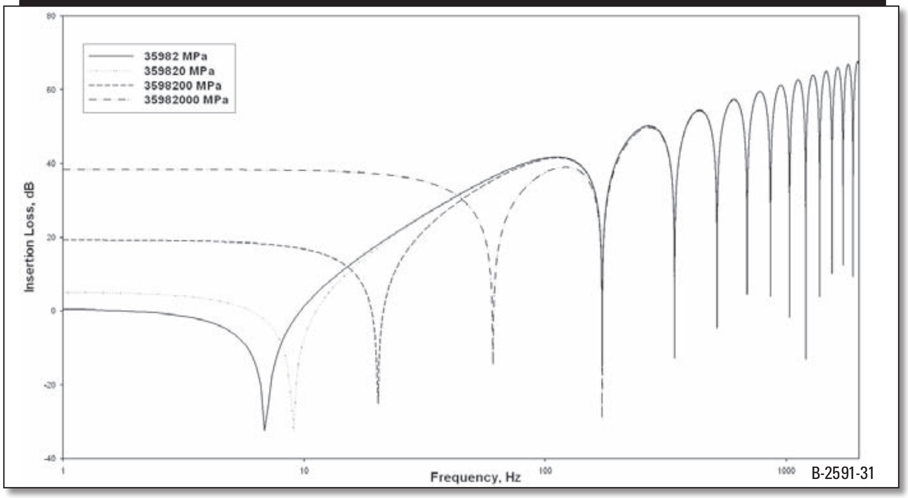
FIGURE 30: Varying panel thickness.



30). Increasing panel thickness increases effective stiffness, which results in increased insertion loss, particularly noticeable at the lower end of the IL response spectrum, where insertion loss is controlled directly by the panel's effective stiffness.

Last, I varied the panel material's Young's modulus, a mechanical parameter that effectively characterizes a panel's stiffness (**Fig. 31**). As expected, you see that IL response variations happen primarily in the lowest decades of the IL response spectrum; IL increases as E increases. Note too that with increased stiffness you see the first,

FIGURE 31: Varying panel Young's modulus.



lowest frequency IL minimum shift upward in frequency.

To complete the simulation run, I applied various simulated IL response spectra to actual db-SPL driver response curves, which resulted in predicted panel acoustic response, when driven by the acoustic pressure waves generated by the driver enclosed *within* the cabinet.

Specifically, I measured a raw driver, then simulated a totally enclosed cabinet, made of the polymer composite with the driver placed within the cabinet (not bolted into the face of one of the cabinet's panels). The simulations produced predicted results of dB-SPL measurements, taken at 1m, outside the cabinet (**Fig. 32**).

BASELINE

The top curve is the actual dB-SPL response of the raw driver, measured at 1m. The bottom curve is that of the predicted response as measured outside the simulated cabinet that totally encloses the driver, at a distance of 1m.

Keep in mind these results are for an enclosure that contains no sort of sound-absorbing filling or liner; thus the thermodynamic condition within the enclosure is adiabatic. Using an absorptive filling and/or liner would alter the predicted results, likely lowering the actual dB SPL measured outside the cabinet walls. As well, the simulation does not account for panel acoustic output resulting as a consequence of mechanical vibrations originating at the driver's frame.

At any rate, once having established a performance baseline, I then ran a further set of simulations, each with the purpose of exploring (and illustrating) the effects of varying mechanical properties of the cabinet panel's material.

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FIGURE 32: Raw driver simulation.

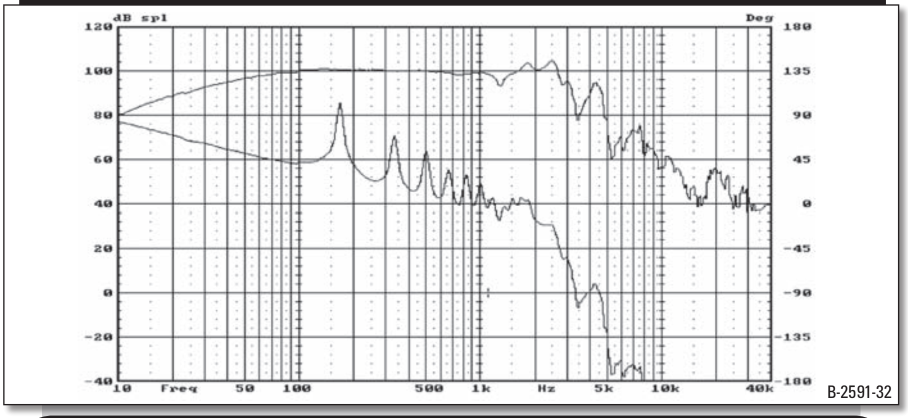
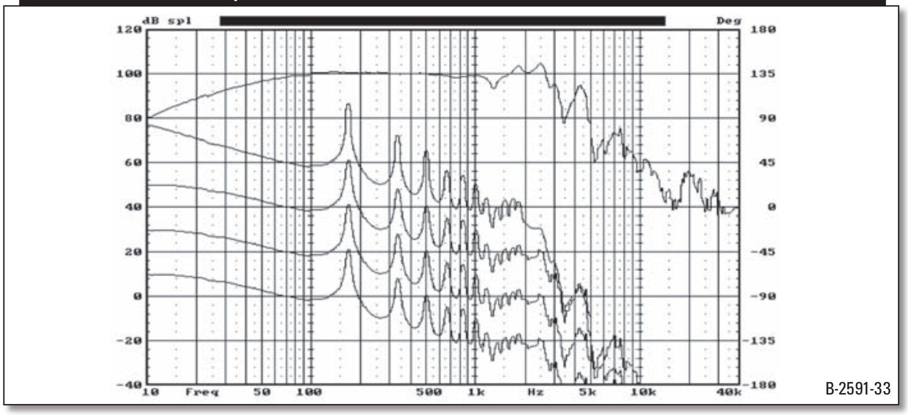


FIGURE 33: Density variations.



My first simulation series (**Fig. 33**) held all properties constant, except for density. Varying density produced substantial differences in the amount of insertion loss and in turn substantial effects on panel acoustic output. The resulting curves show that the more dense the material, the higher the insertion loss, resulting in decreased panel dB SPL levels.

Next, I re-ran the simulation, this time holding all parameters constant, save stiffness (**Fig. 34**). Increasing stiffness did affect IL, but it took quite a substantial increase in panel stiffness before significant changes in IL and thus panel acoustic output became evident. You can see from **Fig. 33** that IL did increase as stiffness increased, and at the lower end of the response spectrum, just as predicted by theory.

Last, I ran a series of simulations holding all parameters constant, save thickness. As you can see in **Fig. 35**, increasing panel thickness increases IL.

CONCLUSION

In this article you have seen that panel resonances—either structural or airborne in origin—can have effects, usually deleterious in nature, on system response. You have also seen that the designer has at his or her disposal a number of means to damp, disperse, or shift these resonances out of the frequency range of interest. I think it also fair to say that the use of such modeling techniques as FEA can be an invaluable, efficient tool for exploring various design solutions when seeking to lower the mechanical noise floor of the system.

In Part 2, I'll look at additional panel-quieting measures such as constrained layer damping. *aX*

FIGURE 34: Stiffness variations.

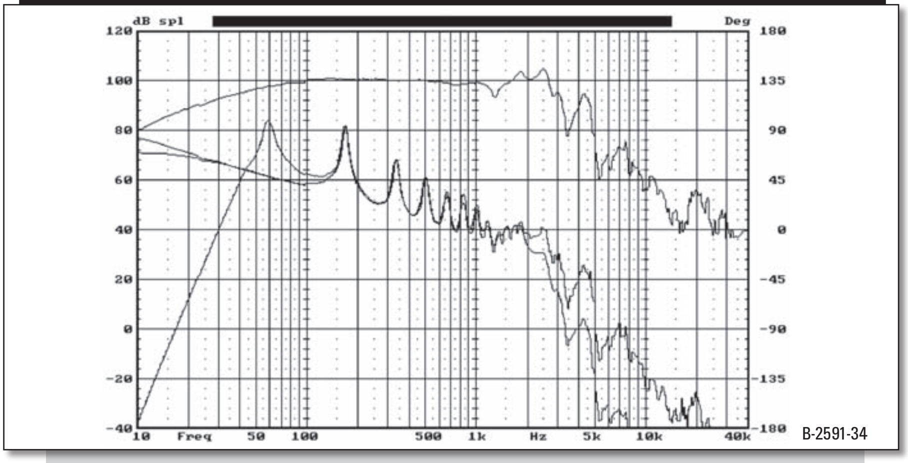


FIGURE 35: Thickness variations.