Large-Signal Analysis

Thermal Analysis

When a high level signal (input power) is applied to the loudspeaker for a prolonged period of time, the temperature of the voice coil will raise significantly. This is mainly due to the low efficiency of the loudspeaker itself, which in most cases falls between 1-5% for a typical HiFi driver. The remaining 95% of the energy provided by the amplifier is wasted as a heat in the voice coil. The raise in the temperature of the voice coil has a detrimental effect on the driver's performance. The final temperature of the voice coil depends on the input power of the applied signal over time and driver's mechanical design. Various mechanical solutions have been proposed for the motor system of the driver to minimize the voice coil temperature. All these solutions have common elements, which for the purpose of analysis, are translated into a thermal model characterized by heat flow and appropriate thermal resistances.

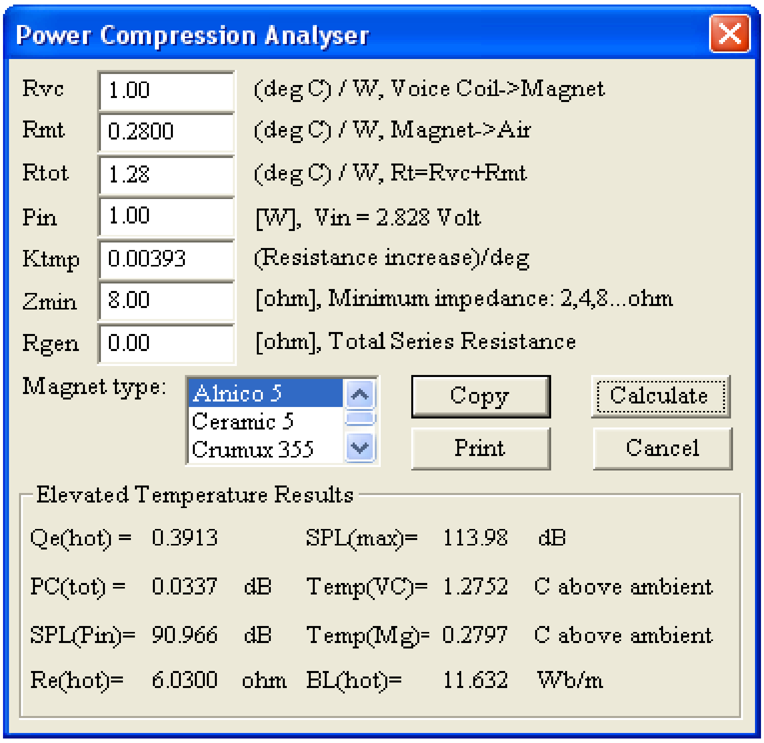


Fig 4.46 Thermal Compression analyser.

The mechanical structure closest to the voice coil is the magnet system and the thermal interaction between these two determines **Rvc**, the voice coil-to-magnet thermal resistance. Once the magnet heats up, it will radiate the heat directly into the surrounding air. The losses of this process are modeled by **Rmt**, the magnet-to-air thermal resistance. As one can see, the magnet, being larger structure and directly exposed to the air, has lower thermal resistance. The structure of the magnet will lose some of its flux due to the raise of its temperature. This process is reversible, but will cause reduction in SPL level. Another factor reducing the SPL is the raise of the voice coil resistance. The resistance increase per degree C is given by the **Ktmp** factor. For copper and aluminum wires **Ktmp**=0.00393 and for a 100.0 deg C temperature raise, there will be almost 40% increase in voice coil resistance (Re), (**Rtot = Rvc+Rmt**). Increased Re will cause less current to be drawn from the amplifier and also increase in electrical Q-factor (Qe).

All the data entered on the "Power Compression Analyzer" are real, positive numbers. Magnet type determines the rate of the flux loss. Generator’s internal resistance should be entered into the “**Rgen**” field. <u>Input power into the modeled loudspeaker is edited from the “**Pin**” data field</u>. “**Zmin**” is the nominal speaker impedance and should be entered as one of the standard impedances such as: 2 / 4 / 8 / 16 ohms. Once the frequency response curves are plotted for 1, 10, 100 and 1000 watts of the input power, it can be noticed, that the curves do not raise by 10 dB, but less and the shape of the plots is changing due to modified Qe values (see Figure above). All the data entered on the "Power Compression Analyzer" are real, positive numbers. Magnet type determines the rate of the flux loss and was given in Chapter 2.

If your driver uses totally different magnet material, select the flux loss, which is closest to your drive's magnet material (using the "Editor" screen) or contact Bodzio Software for ***free*** customized version of the program. The maximum SPL level is calculated for the voice coil reaching 200 deg C. This is the maximum temperature level for the cone assembly before the glue starts to deteriorate. The "Power Compression Analyzer" is shown on Fig 4.31. A hard copy of the "Power Compression Analyzer" calculations can be obtained by pressing the **"Print"** button from the Analyzer dialog box.

How to determine Thermal Resistances ?.

The thermal resistance model used by SoundEasy takes into account three major elements: (1) loss of electro-motor force due to heating up of the magnet assembly, (2) raise of the Qe factor and (3) raise of the Re (DC resistance of the voice coil) due to rising temperature of the voice coil. Combination of these factors is known as "Thermal Compression". The first factor will be reflected as reduction in SPL. The second factor will cause the Qt to raise and then changes in the amplitude response of the driver, particularly around the box tuning frequency will be observed.

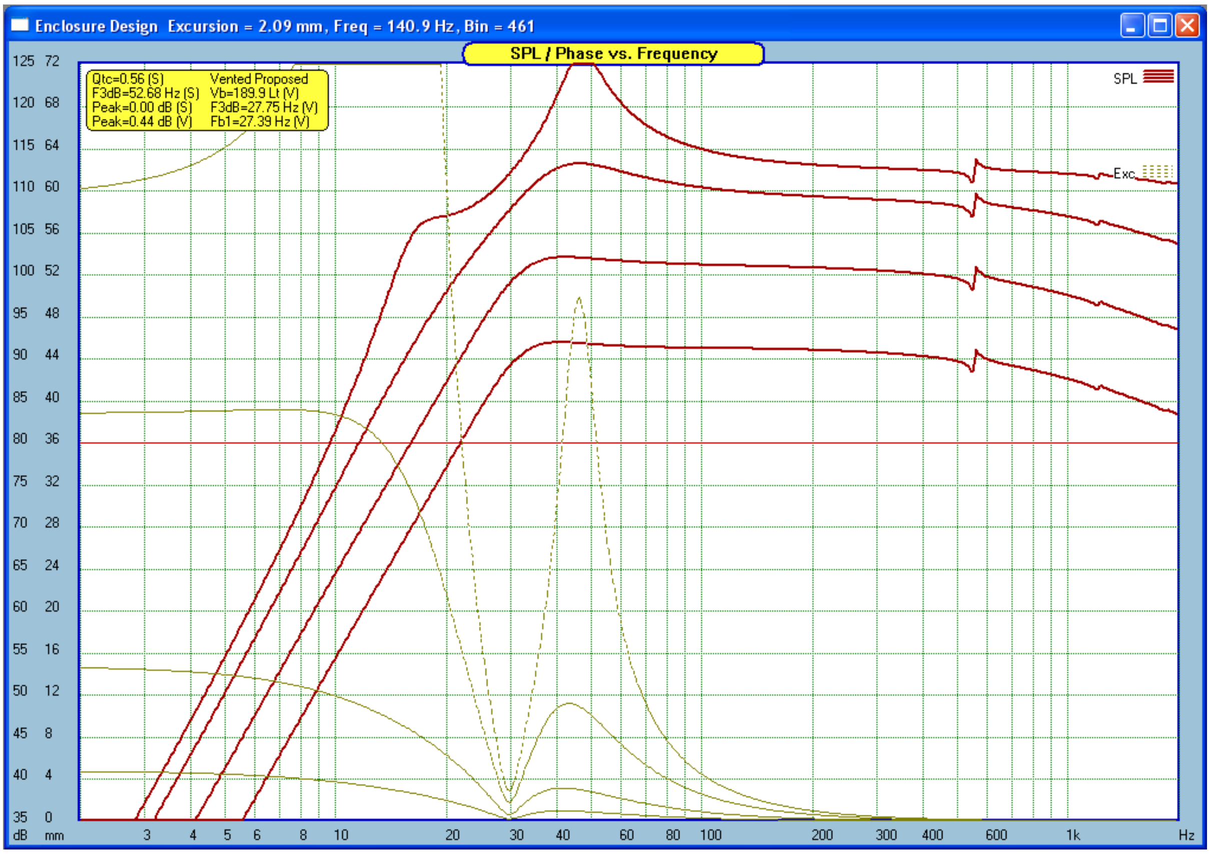
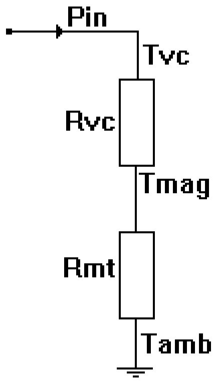


Fig 4.47 SPL and Cone Excursion at various Pin: 1, 10,100 and 1000W – BL(x) and Cas(x) NOT included

The third factor will cause less current being drawn from the amplifier (assuming constant voltage source) and as a consequence, further reduction in SPL. Several very useful thermal models of the loudspeaker have been proposed and Bodzio Software acknowledges here the works of C.A. Henricksen and D.J. Button. The thermal resistance model consists of two elements:

1. Rvc - Voice coil to magnet thermal resistance. This element lumps together several associated thermal resistances.
2. Rmt - Magnet to air thermal resistance.

The simplification is quite deliberate and because of this, both components can be measured quite simply. The model thus looks as follows:



Where:
Tvc - Temperature of the voice coil.
Tmag - Temperature of the magnet assembly.
Tamb - Temperature of the surrounding air, usually 20 Deg C.
Pin - Input power to the loudspeaker applied for min 1 h.

$$R_{VC} = \frac{T_{vc} - T_{mag}}{P_{in}} \qquad R_{mt} = \frac{T_{mag} - T_{amb}}{P_{in}}$$

Example: Tvc = 100 deg C.
Tmag = 35 deg C.
Tamb = 20 deg C.
Pin = 60 Watt.

$$R_{vc} = \frac{100-35}{60}[\deg/\,watt] = 1.08[\deg/\,watt]$$
$$R_{mt} = \frac{35-20}{60}[\deg/\,watt] = 0.25[\deg/\,watt]$$
$$R_{tot} = R_{vc} + R_{mt} = 1.33[\deg/\,watt]$$

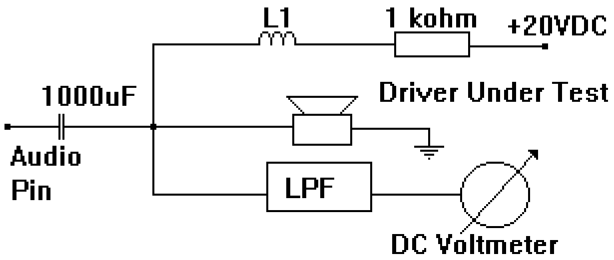
It is clear, that in order to calculate the Rvc and Rmt, the temperature of the voice coil and magnet assembly must be known. There should be no problem with measuring the temperature of the magnet. A good, accurate thermal probe (such as the DVM thermal probe) attached to the magnet, with the help of some thermal compound should suffice. Measuring the temperature of the voice coil is a little trickier. The method proposed here is similar to the work of C.A. Henricksen (JAES. Vol 35, No 10, 1987 October). Essentially, two signals are fed to a loudspeaker: (1) an AC power, to heat up the voice coil and (2) a small DC current, which us used to determine the resistance of the hot voice coil. The DC current will cause a small voltage drop across the loudspeaker's DC resistance (Re). The voltage is sufficiently small, that it will not cause any extra heating or mechanical offsetting the voice coil. When the voice coil is cold, the DC resistance of the loudspeaker is Re (known), and causes a DC voltage Vcold, to appear across the loudspeaker terminals. The AC power is generated by an ordinary audio power amplifier and should be kept at about half of the maximum loudspeaker's rated power. When the AC power is applied to the loudspeaker, the voice coil starts to heat up. After 1-2 hours, of continues operation, the temperature in the system starts to stabilize. The DC resistance of the voice coil has increased and now the DC voltage across the loudspeaker reads Vhot. Temperature of the magnet assembly, as measured by the thermometer, is now Tmag. The AC power is now turned off and the loudspeaker disconnected from the system and replaced by a variable resistor (you may use several smaller resistors of different values). The variable resistor is that set to such value (Rhot), that produces the same voltage (Vhot) on the DC voltmeter. The Rhot is the final DC resistance of the hot voice coil. From the formula:

$$R_{hot} = R_e * (1 + 0.00393 * (T_{vc} - T_{amb}))$$

valid for copper and aluminum voice coils, one may calculate:

$$T_{vc} = \frac{R_{hot} - R_e}{0.00393 * R_e} + T_{amb}$$

All the values: Tvc, Tmag, Tamb=20deg, and Pin are known and Rvc and Rmt can be calculated from the formulas shown before. Block diagram presented below shows typical test arrangement. Audio power is fed to the loudspeaker through a large capacitor (1000 uF), to block the DC current flowing back to the amplifier. The large coil (L1) prevents the audio power from going back to the DC voltage source. The 1kohm resistor converts the 20V DC voltage source into a 20mA current source. The LPF prevents the audio power to reach the DC voltmeter and degrade the accuracy of the readings. The LPF should have at least 100dB attenuation at the audio test frequency (300Hz). You may decide to use a combination of passive and active filtering.



Method for measuring voice-coil temperature of a loudspeaker.

Example:

1. Re = 6 ohm (from driver's data sheet)
2. DC voltage source = 20 VDC, in series with 1kohm resistor produces 20mA current.
3. Voltage Vcold = 6ohm*20mA = 120mV
4. Vhot = 157.7mV, and Rhot = 7.88ohm, as measured using substitution method.
5. Tvc = (7.88 - 6.0)/(6.0*0.00393)+20 = 79.9 deg + 20 deg = 100 deg C.
6. Tmag = 35 deg C, as measured using thermometer.
7. Pin = 60 watt.
8. Rvc = (100-35)/60 = 1.08 [deg/watt]
9. Rmt = (35-20)/60 = 0.25 [deg/watt]

Port Nonlinearities and Compression

Ported enclosures are known to extend the low frequency output of the loudspeaker system by exploiting Helmholtz resonator created by the compliance of the air inside the enclosure and inertance of the air in the port. Acoustic transformer created this way has its own resonant frequency, Fb, at which most (or all, if there was no losses) of the system acoustic output comes from the port. A fairly obvious implication of this is significant velocity associated with the air flow through the port. This in turn, causes all sorts of non-musical noises to be generated by the port and also distortion and acoustic compression. Depending on port geometry, and required low frequency SPL of the system, the issue can be quite significant.

The problem described above belongs to rather complex field of fluid flow. Assuming incompressible flow, some sophisticated FEM programs would be able to model air turbulence and associated vortex shedding in more detail[3], but this is well beyond the scope of my article. Fortunately, existing research results, design material and tests results enable us to formulate an approximate macro view of the problem and look at the acoustic impedance of the port under high air velocity. It needs to be stressed, that the approach taken here is a significant simplification of the physics of the problem. However, the resulting model is quite useful and finds confirmation in practical tests.

Intuitive approach to port non-linearity

We are all familiar with the need for good carpentry skills when building speaker boxes. Accuracy of joints and sealing the box is essential for proper operation of all types of boxes, be it sealed, vented, passive radiators and so on. Sealed box means exactly this – the air inside the box is trapped and sealed from the external world. There is no parasitic or accidental leakage from the box, so that mathematical model developed for the enclosure continues to be accurate. Consequently, the box Qb factor is controlled by the designer and NOT by sloppy workmanship. Vented enclosure also needs to be “air tight”, of course with the exception of the purposefully introduced opening – port. Just as for the sealed box, the cabinet needs to be sealed, so that no accidental air leakage from the box can occur. Properly executed design would include all sorts of seals and gaskets to ensure, that connector boards or drivers themselves do not cause air leakage. Assuming you have your perfect vented box build, you may expect that the SPL curve and input impedance curve will look as on Figure 4.33.

The impedance curve is very familiar and has two characteristic peaks with the dip between them. The dip is located exactly on the box tuning frequency, Fb = 25Hz. My design is a QB3 type with system parameters as shown on Figure 4.33, and we have also assumed, that Qb = Qp = 1000, so we have a very low-loss design. Port diameter in this example is 15cm or 6 inches and the input power to the system is 1watt. Now, assume, that we start reducing port diameter. Initially, the picture will not change by much. However, when the port eventually becomes very small, intuitively, there should be very little difference between the vented box with a “very small port” and sealed box with a “large leakage” problem. In this case, you would expect than the SPL curve will resemble that of a leaky sealed box and the input impedance curve will loose the lower peak and will become a “single peak” curve just like the sealed boxes have. Between those two extremes, you may expect problems commonly labeled as “port non-linearity”.

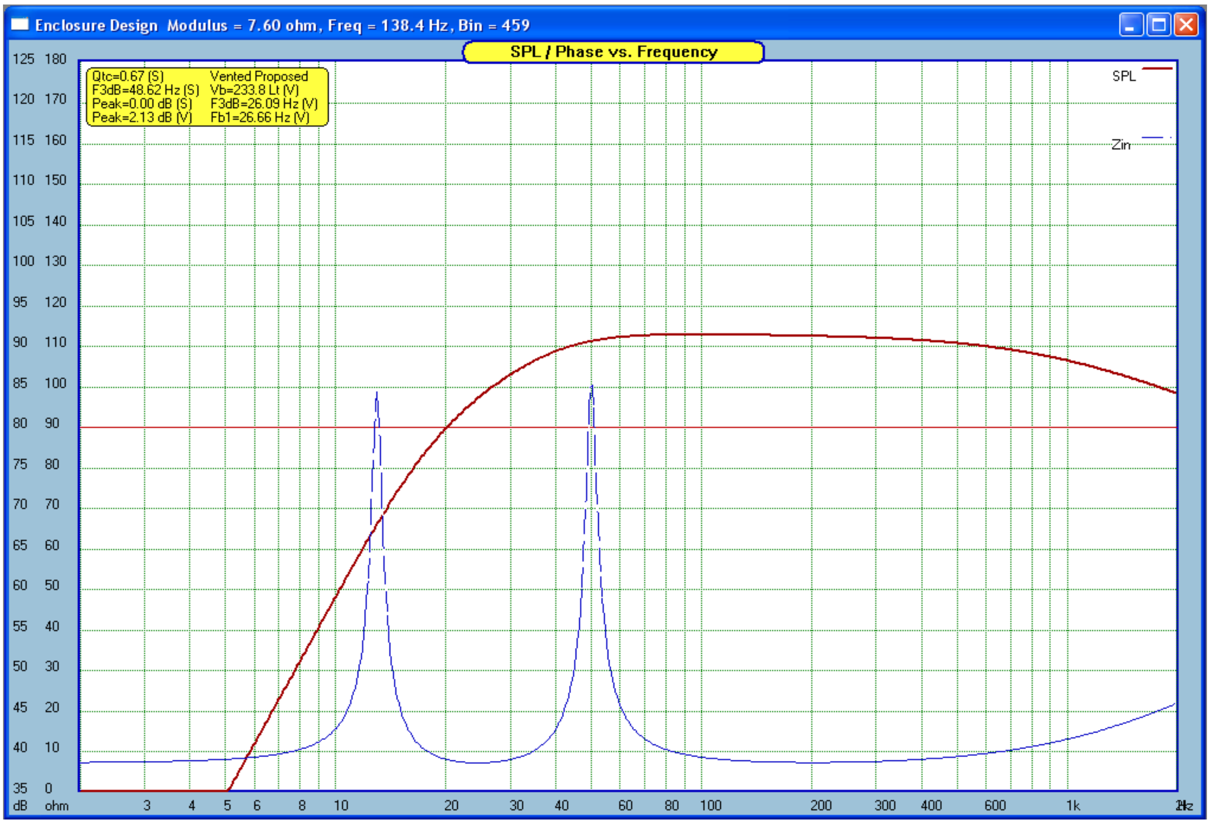


Figure 4.48 SPL and input impedance @ 1watt, port diameter = 15cm.

Now, assume, that we start reducing port diameter. Initially, the picture will not change by much. However, when the port eventually becomes very small, intuitively, there should be very little difference between the vented box with a “very small port” and sealed box with a “large leakage” problem. In this case, you would expect than the SPL curve will resemble that of a leaky sealed box and the input impedance curve will loose the lower peak and will become a “single peak” curve just like the sealed boxes have. Between those two extremes, you may expect problems commonly labeled as “port non-linearity”.

Historical research on fluids in tubes

Bies and Wilson actually experimented with an orifice (more like a decent port) 9cm (3.5”) in diameter and 2.9cm (1.13”) long. They found, that the acoustic resistance, R, of the tube varies with the particle velocity in a similar way as shown on Figure 4.34.Thurston, working with fluid flow through circular tubes found that analogous acoustic resistance, R, and inductance, M, again vary in similar way as shown on Figure 4.34. A landmark paper by Ingard offers an empirical formula for the non-linear component of the acoustic resistance of a tube. Later on, this work has been expanded by Ingard and Ising who offered more complete mathematical treatment of orifice behavior. Backman, experimenting with port non-linearity, plotted driver’s input impedance for various input voltages and determined, that the most sensitive to the amplitude variations is the magnitude of lower impedance peak. He pointed out, that the behavior of the vented enclosure approaches a sealed enclosure at high levels. The inner diameter of the port was 102mm (≈4”) and the length of the port was 168mm (≈6.6”). Vanderkooy measured and analyzed ports performance at high levels and proposed a simplified nonlinear model of a port. The comprehensive analysis included jet formation, acoustic compression, turbulence noise and distortion.

Simplified view of the problem

Typically, the area of the cone is many times larger than the area of the port. Since the volume of the air pushed via the port needs to be approximately equal to the volume displacement of the cone, the only way to accomplish this, is to push the air in the port much faster. At low SPL levels, the volume displacement of the cone is small and consequently, the cone velocity is low as well. This, in turn, stipulates low air velocity in the port or what is known – a laminar air flow. The process is characterized by a low Reynolds number, Re<2000, and there is no turbulence in the air flow.

$$Re = (r*V_{max}*\rho)/\mu$$

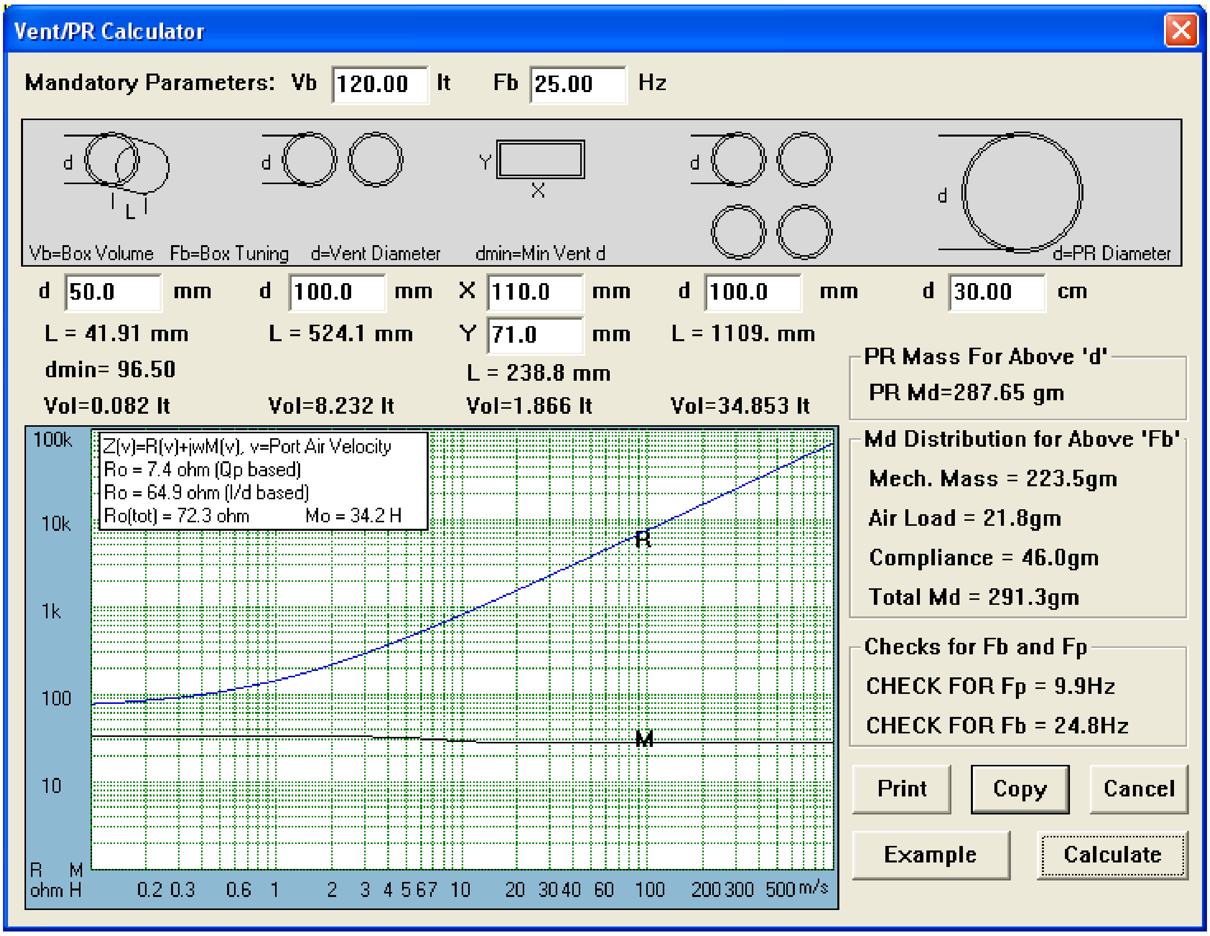


Figure 4.49. Acoustic impedance R+jωM of a short port.

Where V_{max} is the peak flow rate, ρ is the density of, μ is the viscosity of air, and r is the radius of the port. At this point of time, the acoustic resistance of the port, Rp, (port loss) is linear and is relatively low (50-200ohm). Also, the mass of air in the port, Mp, is constant. Remember, that mass of the air in the port resonates at Fb (box tuning frequency) with the compliance of the air inside the box, halting the movement of the cone. If there were no losses, at resonance, Fb, all acoustic output of the system would come from the port. Now, lets start reducing the port diameter. As we can guess, the volume displacement of the cone remains the same, so the air in the port must move faster and faster. Soon, the turbulence occurs in the air flow, jets of air are forming and the process is no longer linear. Reynolds number corresponding to the onset of turbulence is Re ≈ 20000. When the Reynolds number hits 50000, your vent is compressing. It is now more and more difficult to push the air in the vent. The acoustic output of the port is below the expected level (if the process continued to be linear) and you begin to experience "port compression" effect. Port impedance, Zp, has now additional component relating to turbulent air flow and depending on the air velocity, Rp(v). Also, the Mp has changed as well, drifting towards 60-70% of its original value and becoming velocity dependant. The Rp(v) is an interesting element. If plotted in frequency domain, it would resemble the curve depicting port air velocity, shown on Figure 4.35. This should be of no surprise, as Rp(v) is so heavily dependant on the port air velocity. In the frequency domain, the Rp(v) will quickly reach its peak not far away from the lower peak of the input impedance curve. Depending on the geometry of the port and the volume displacement of the driver, the air velocity in the port may reach 50-100m/s. Such a high values of the air flow result in Rp(v) reaching 5-10kohm levels.

The “double-peak” impedance curve is a clear result of the port action. The port air velocity curve plotted on Figure 3 clearly indicates, that the lower impedance peak will be much more affected by the port non-linearity than the upper impedance peak. Therefore, the Rp(v) needs to be incorporated in the driver’s model.

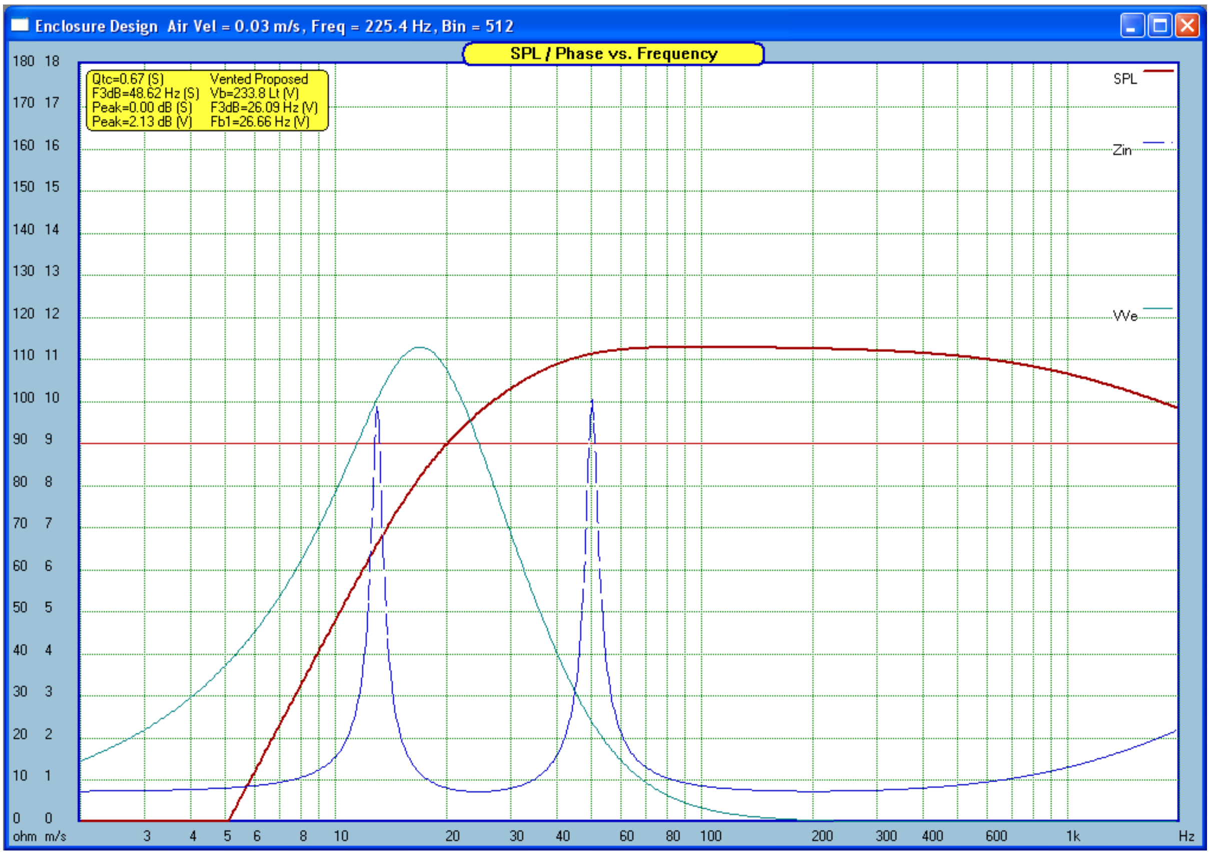


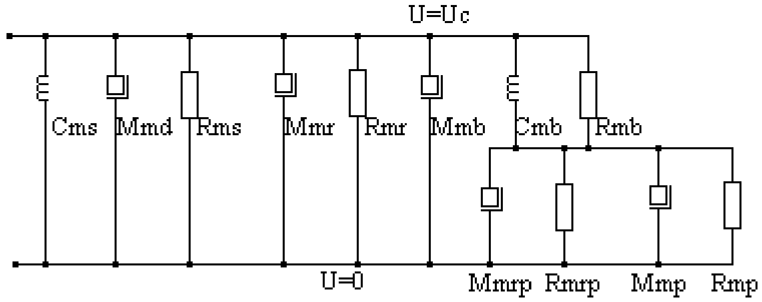
Figure 4.50. Port air velocity (curve 1) for port diameter = 5cm.

Changes in driver’s model

Vented enclosure shown on Figure below, provides different loading for the back of the diaphragm, as compared to the sealed box. The vibrating system and front loading of the diaphragm are represented on the mechanical mobility circuit the same way as for the sealed enclosure. Introduction of the vent adds several more components such as: (1) mass of the air in the port Mmp and its losses Rmp and (2) radiation impedance of the port represented by Rmrp and Mmrp. The air in the port is treated as a mass because of its small volume and more importantly, because it is incompressible. Particles of air will move on both sides of the vent with the same velocity. The air compressed in the box by the back side of the diaphragm has only one path to escape - pushing the air mass trough the vent. Therefore, the pressure path consists of series connection of Cmb, representing compliant air in the box and the four elements of the port. Since the air in the port is incompressible, the immediate layer of air in front of the box (radiation impedance) will be connected to the same velocity line as the entry to the port inside the box. The other end of the masses is connected to the U=0, or reference velocity as required in mechanical mobility circuits. The mechanics of the above process can be easily demonstrated on a physical model of a vented box. Connecting a small (1.5V) battery across vented box terminals, we can displace the cone in or out of the box. Small air-flow detecting device (candle) positioned in front of the port will show significant air movements, in the direction opposite to the diaphragm. The volume of air displaced by the cone should be similar to the volume of air leaving the port. If the difference is significant, than leakage losses are significant. The above experiment clearly shows the pressure (current) path in the mechanical mobility model, so it should now be easy to explain why the compliance of the box is connected in-series with port elements. It is observable, that Cmb and Mmp+Mmrp form series resonant circuit in the mechanical mobility representation. The circuit will act as a “selective short circuit” for the volume velocity Uc, shorting it to U=0 (ground) at the circuit resonant frequency. Because of the circuit losses, the short is not perfect, but velocity Uc will be much reduced. In the practical system this situation translates into much reduced cone excursion at the box resonant frequency.

Acoustical impedance representation shows Cas and Map+Marp forming parallel resonant circuit. Electrical circuit theory advocates that very little energy (current) needs to be fed into the circuit for it to resonate and for the current (volume velocity) in the resonant circuit to be still very high. Therefore, volume velocity in the “feeding” branch, which contains diaphragm output will be very small and volume velocity in the resonant circuit containing port will be high. This effect, although the strongest on the resonant frequency Fb, will extend over some narrow frequency range and on the low-end side creates extended system output. It is the enclosure/port resonance effect, that is being exploited here to augment system output at low frequency. Figure 4.29 shows mechanical mobility (top diagram) and acoustical impedance (bottom diagram) representation adopted for the vented enclosure model. The components are:

Cas,	equivalent compliance volume Vas transformed to acoustical side.
Mad,	mass of the vibrating system Mms transformed to acoustical side.
Ras,	vibrating assembly loss Rms transformed to acoustical side.
Mar+Mab,	air radiation of the front side of the diaphragm + air load of the back side of the diaphragm.
Rar,	air radiation of the front side of the diaphragm.
Cab,	enclosure compliance Vab transformed to acoustical side.
Rab,	absorption losses of the enclosure transformed to acoustical side.
Marp, Rarp	port radiation.
Map,	mass of the air in the port.
Rap,	frictional losses in the port.



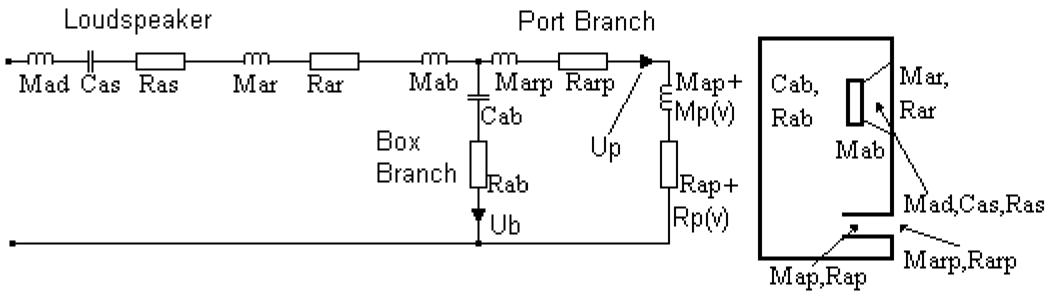


Figure 4.51. Modified model of the vented box including Zp(v) = Rp(v) +jωMp(v)

The non-linear port impedance was implemented as Zp(v) = Rp(v) +jωMp(v) in the port branch. Please note, that Map+Mp(v) will exhibit slight reduction in value as the air speed increases and Rap+Rp(v) will exhibit significant increase in value as the air velocity in the port increases.

Resulting performance

In order to gain some insight into system performance affected by port non-linearity problems, was plotted the SPL for a port of 5cm in diameter for 1W (curve 0), 10W (curve 1) and 100W (curve 2) input power – see Figure below. As we can see, with the increased input power, there is a sort of “saddle” developing on the SPL curve around the box tuning frequency of 25-30Hz. This is exactly the frequency range, where we would expect the port to contribute most to the system SPL. Our small port is clearly not performing as anticipated. Next, we modeled SPL for the same power levels, but this time, we used larger port, 15cm in diameter. The resulting plots on Figure 4.38 do not exhibit the “saddle” any more for 100W power level. Clearly, as the input power is increased, the SPL curves go up, maintaining the approximate shape acquired at 1W power level. It is also easy to observe, that the SPL curves now have resonant peaks above 200Hz, not seen on Figure 4.37. This is the result of enlarging the diameter of the port.

Figure 4.52. SPL for port diameter of 3cm, @1W, 10W and 100W input power

You may remember, that larger port must also be longer if tuned to the same frequency. Now the length of the port is such, that the self-resonances of the port tube fall into much lower frequency range – just to be displayed on the screen. In order to estimate "port compression" effect, you can plot the SPL of the two ports on the same screen – see Figure 4.39. Visual inspection of the graphs shows about 5dB of port compression at Fb = 25Hz. Well, we have just about lost our vented box performance gains if we was to use the smaller port. In the next step, it would be interesting to compare the input impedance plots to see if the lower impedance peak, characteristic for the vented enclosure, indeed disappeared from the plots. The answer is clearly evident on Figure 4.53, where the input impedance for three ports is plotted at 100W power level. The 15cm vent exhibits the expected "double peaks" curve, however the compression is still registered on the impedance plot. Ideally, the port should still be larger. The Reynolds number calculated for these conditions is Re = 68000. Therefore, the port is indeed compressing slightly. However, the 10cm port has the lower impedance peak significantly reduced. This is a sure sign, that this port is too small for the job. The severely undersized 5cm port produces "single peak" impedance curve for 100W input power. This port would also prove to be inadequate for 1 watt of input power. The Reynolds number calculated for 1W conditions is Re = 20000, which is a clear indication, that the port becomes turbulent. Indeed, the corresponding input impedance plot shown on Figure 4.41, is a clear indication of the non-linearity problem @ 1 watt for this port.

Remedies

First and foremost, the problem is related to air velocity in the port. As we know, air velocity in the port, Vp, depends on volume velocity Up, via the port branch divided by the port's area, Sp.

Vp = Up/Sp

Since the area Sp = πr^2, it is easy to observe, that air velocity in the port depends on the inverse squared of the port's radius. It seems rather obvious, that port radius should be kept as large as possible. Dickason recommends 15cm (6") ports for 15" woofer as a minimum and 10cm (4") ports as a minimum for 12" woofers. Salvatti offers a number of recommendations relating to port geometry. This includes balancing inlet and exit flows by using different tapers for inlet and exit. Roozen[3] advocates port geometry based on 6 degrees diverging contour towards the ends of the port. The inlet and outlet are rounded with relatively small curvature. This brief analysis of the vent performance clearly indicates, that nearly all practical size vents will compress at higher SPL levels. Loudspeakers intended for high power stage applications should be give very careful consideration regarding port non-linearity. For this type of application, port compression problem will tend to further degrade system SPL, already affected by the thermal compression during prolonged stage performance.

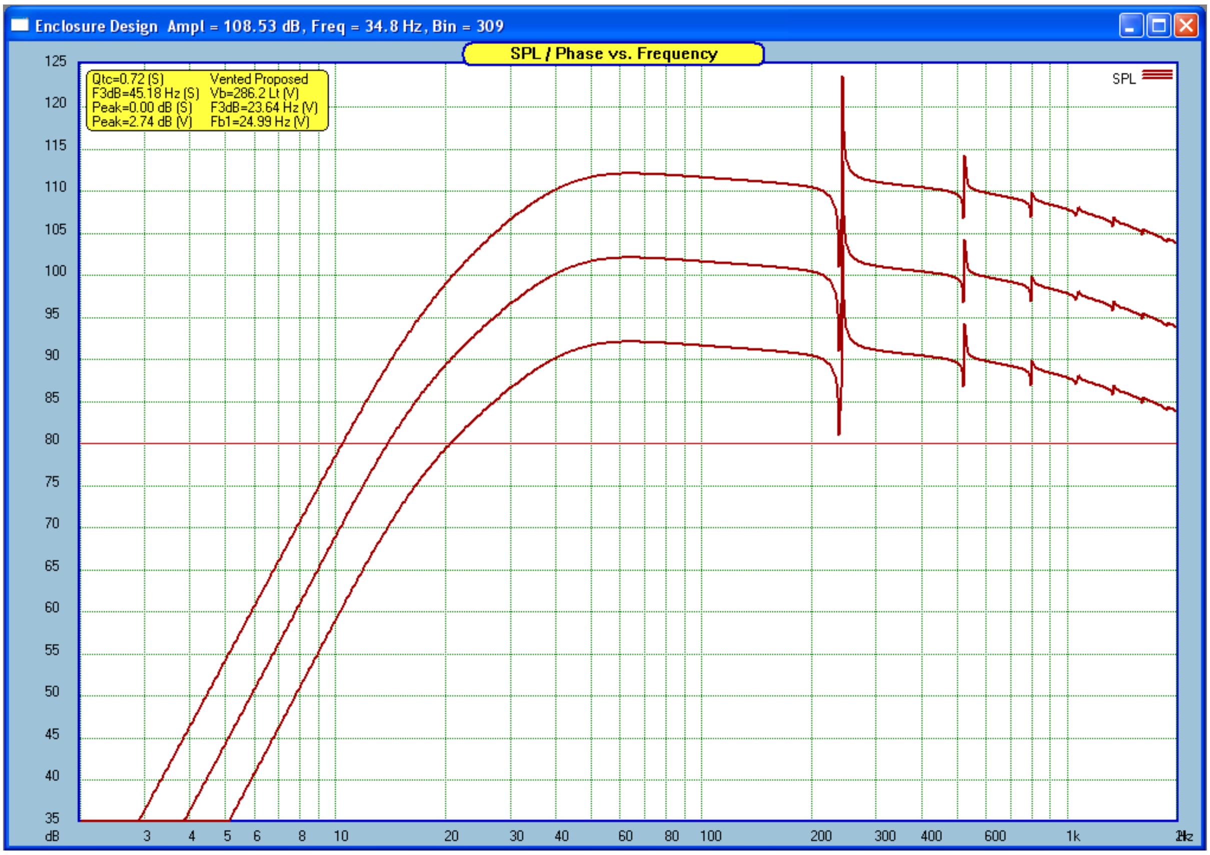


Figure 4.51. SPL for port diameter of 15cm, @1W, 10W and 100W input power

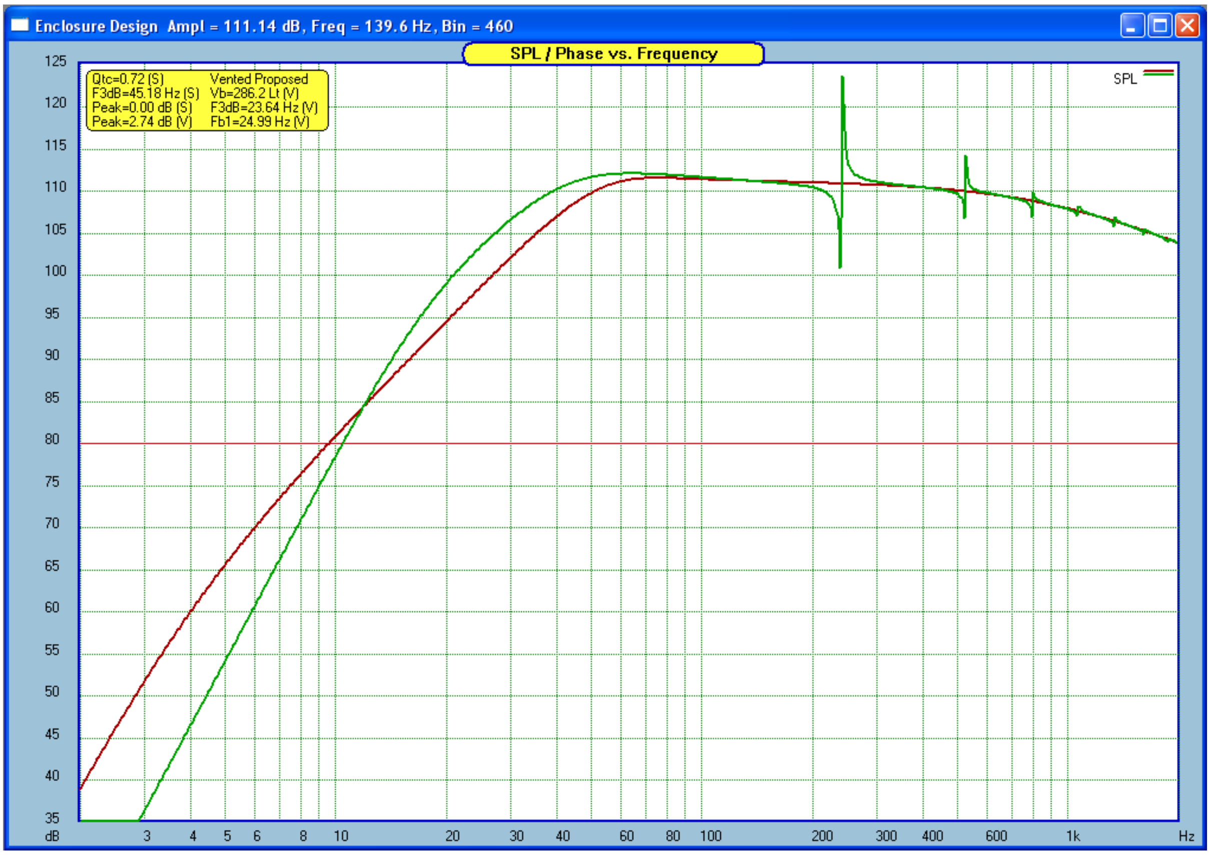


Figure 4.52 SPL for port diameter of 3cm (brown curve), and 15cm (green curve) @100W

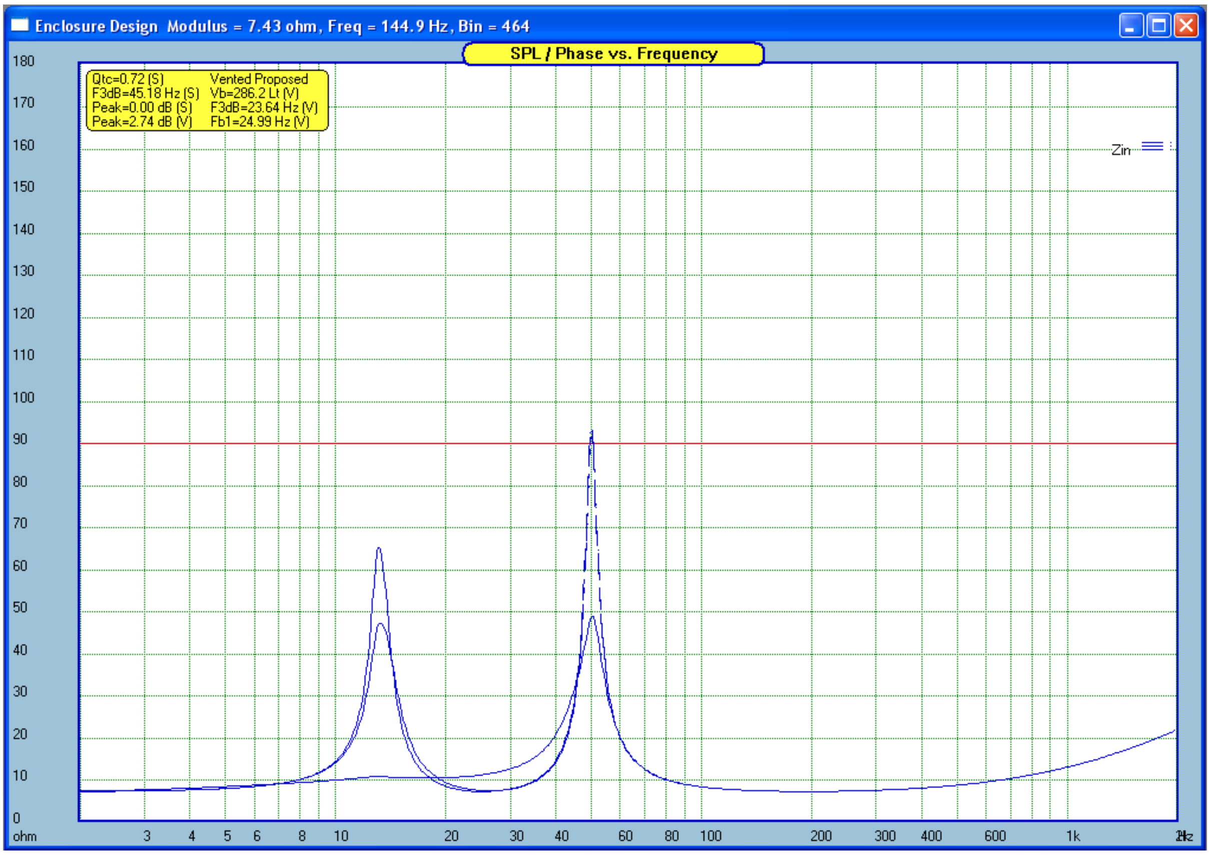


Figure 4.53. Impedance @100W for port diameter of 15cm,10cm and 3cm.

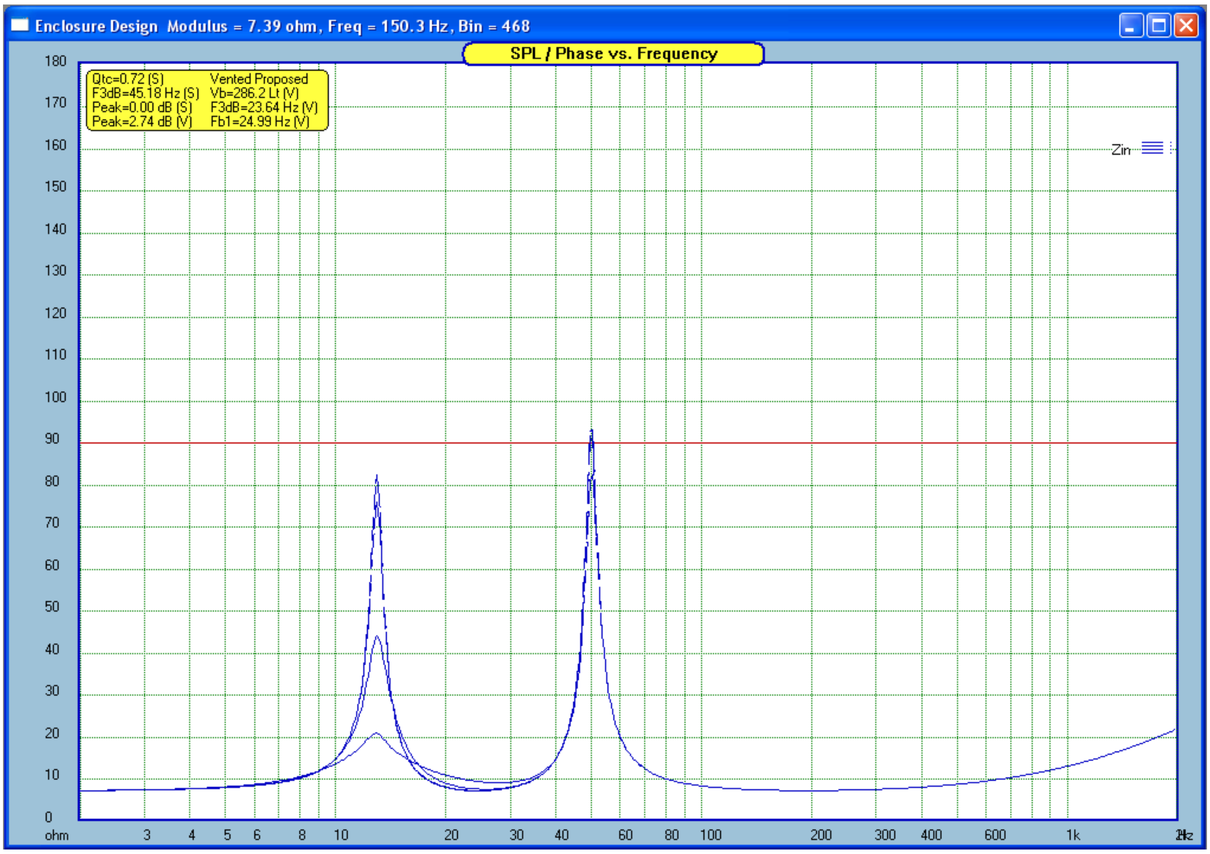


Figure 4.54. Input impedance @1W for port diameter of 15cm, 10cm, 5.0cm and 3.0cm.

BL(x) and Cas(x) analysis

Small-signal model of the loudspeaker and associated enclosure has been increasingly evolving to accommodate non-linear effects inherent in the design of the magnetic motor systems and cone suspension systems. It appears however, that we are in the early stages of the process, as the commercially available drivers are still not fully specified for their large-signal performance.

Indeed, it is very important to characterize the driver for small- and large-signal conditions, as two drivers having similar small-signal data will often perform quite differently under large-signal conditions. In our simplified approach to the problem, we assume, that the large-signal performance of the loudspeaker is mainly governed by non-constant "force factor" – BL(x) and non-constant compliance – Cas(x). Both these quantities are quite dependant on cone excursion – variable "x".

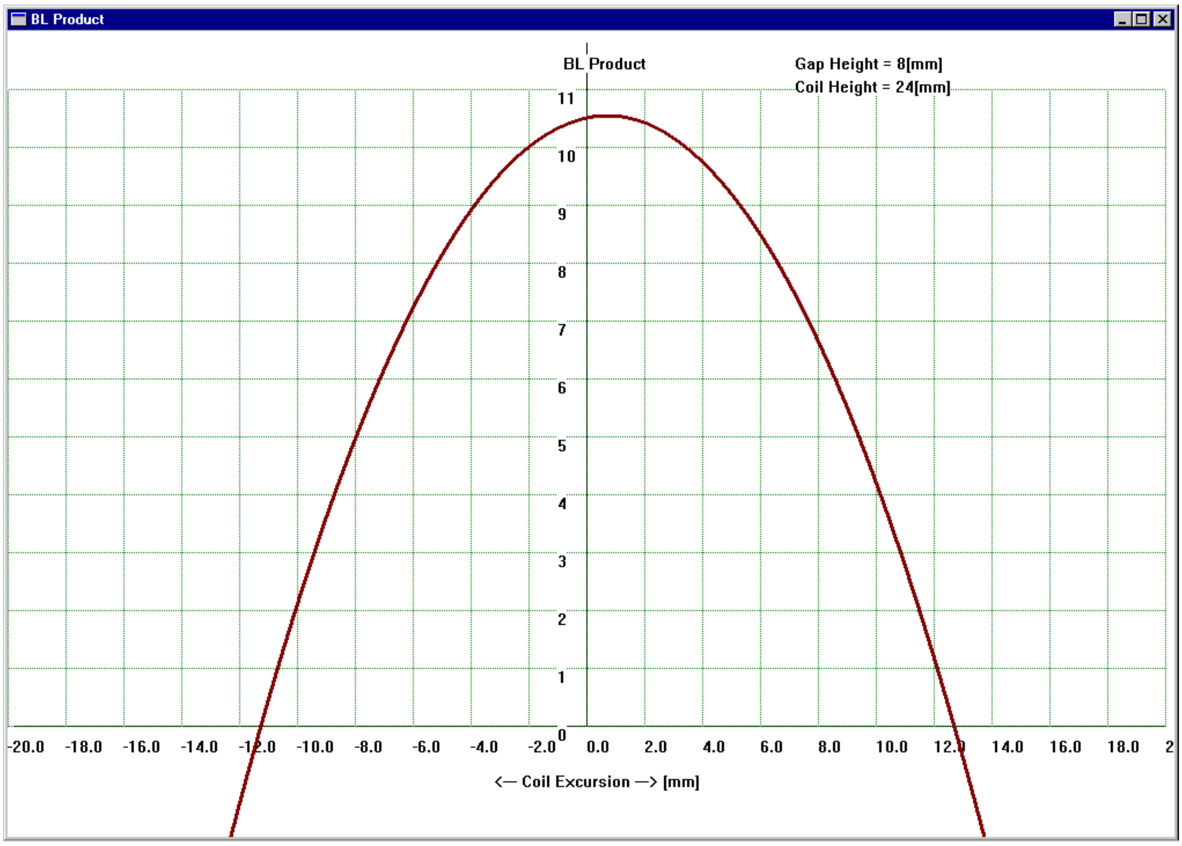


Figure 4.55. Truncated power series - BL(x) becoming negative for larger excursions, BL(0) =10.0

Recent research work focuses on measuring loudspeaker non-linear parameters and presenting them to the user as a power-series coefficients for the force-factor and stiffness. One can only hope, that loudspeaker manufacturing industry will finally embrace this idea and will offer non-linear parameters as a standard set of driver's data. Currently, we seem to some distance away from this point.

Truncated series of power coefficients (first and second order coefficients) approach suffers from several problems. For example: y = BL * (1.0 + 0.01*x - 0.007*x*x) plotted above shows that:

1. BL(x) does not fade gradually for larger excursions and can become negative.
2. Many magnetic "underhung" motor systems do actually have a flat (linear) section around the x=0 excursion and this can not be approximated by the "two coefficient", parabola approach.
3. Many loudspeakers have nearly constant stiffness (1/Cas(x)) over a wide range of excursions and then seep increase at larger excursions. Parabola does not approximate this very well.

The accuracy of power series coefficient method can be much increased by including higher order coefficients.

Possibly the biggest problem for you (and us) is to actually find out what the values of the coefficients are. Please refer to Chapter 17 for more information on large-signal measurement and analysis.

Is there "the next best thing"?.

There is a possibility extracting as much information from the visual inspection of the driver and data sheets as possible. We will end up with an estimation of the large-signal performance, but this is better than nothing.

Loudspeaker driver components are designed with a specific task in mind. Depending on the materials used and the geometry of the design, these components will perform differently in the final design. What we are looking for, is the relationship between design and the intended performance. For example: if you make the voice coil longer, you may expect larger maximum cone excursion, but also some drop in the SPL. If you also use the spider with more corrugation rolls, you would expect higher compliance and again, larger maximum cone excursion – this would be the "long throw woofer".

Now, these are some of the clues to which we need to assign some numerical values.

Estimating Xmax due to BL(x)

Xmax is usually termed as the **"maximum linear excursion". Practical measurements have shown, that this excursion corresponds to 3 – 10% harmonic distortion or 65 – 75% of the drop in BL(x) from its maximum value.** Please keep these figures in mind, as they are my "sanity check" reference.

Formulas for Xmax given below are due to Small and also Gender found good correlation between Xmax and distortion. The formulas link the voice coil length and magnetic gap height and offer an approximated expression for Xmax. The Xmax is expressed in mm.

For "overhung" voice coils:

$$BL(x) = BL_0 \frac{1.0 - \dfrac{0.005}{X\max} x^{\frac{5.0}{\sqrt{X\max + 10.0}}}}{1.0 + \dfrac{0.005}{X\max} x^{\frac{18.0}{\sqrt{X\max + 10.0}}}}$$

$$where \quad X\max = \frac{CoilHeight - MagneticGap}{2.0}$$

For "underhung" and "equal" voice coils:

$$BL(x) = BL_0 \frac{1.0 - \dfrac{0.000005}{X\max} x^{\frac{5.0}{\sqrt{X\max + 10.0}}}}{1.0 + \dfrac{0.005}{X\max} x^{\frac{12.0}{\sqrt{X\max + 10.0}}}}$$

$$where \quad X\max = \frac{MagneticGap - CoilHeight}{2.0} + 0.15 CoilHeight$$

Formulas for BL(x) are empirical. The formulas are "rough" and could be improved by adjusting the constants but as you will see later, they offer some insight into the difficult issues of driver's non-linearity. More importantly, they link the well known Xmax formulas with my goal - to generate BL(x) curves. Please note, that BL(x) curves shown on Figure 4.43 and Figure 4.44, do not suffer from the problems associated with the "two coefficient" power expansion approach. Also, these curves are symmetrical , and therefore will approximate odd order non-linearities better than even order. You would expect a well designed driver to be the odd order limited, particularly in the suspension area.

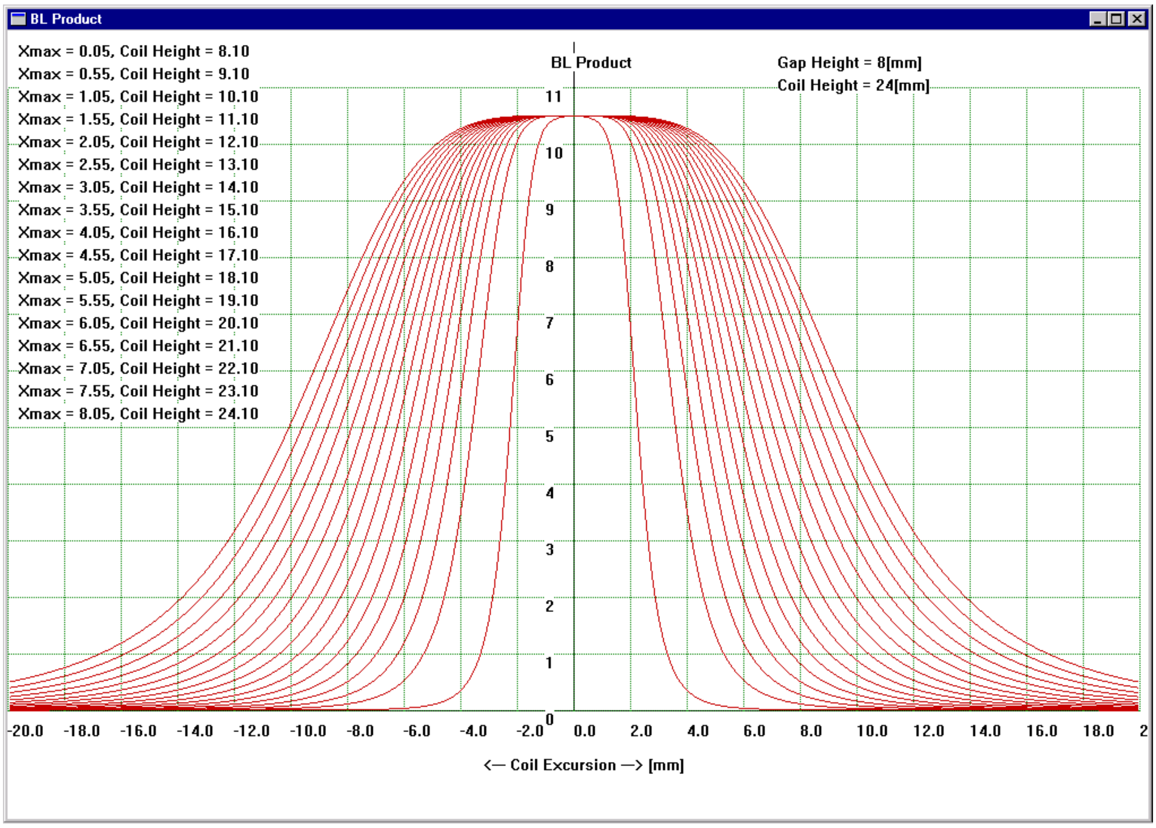


Figure 4.56. Example of the family of overhung Voice Coils

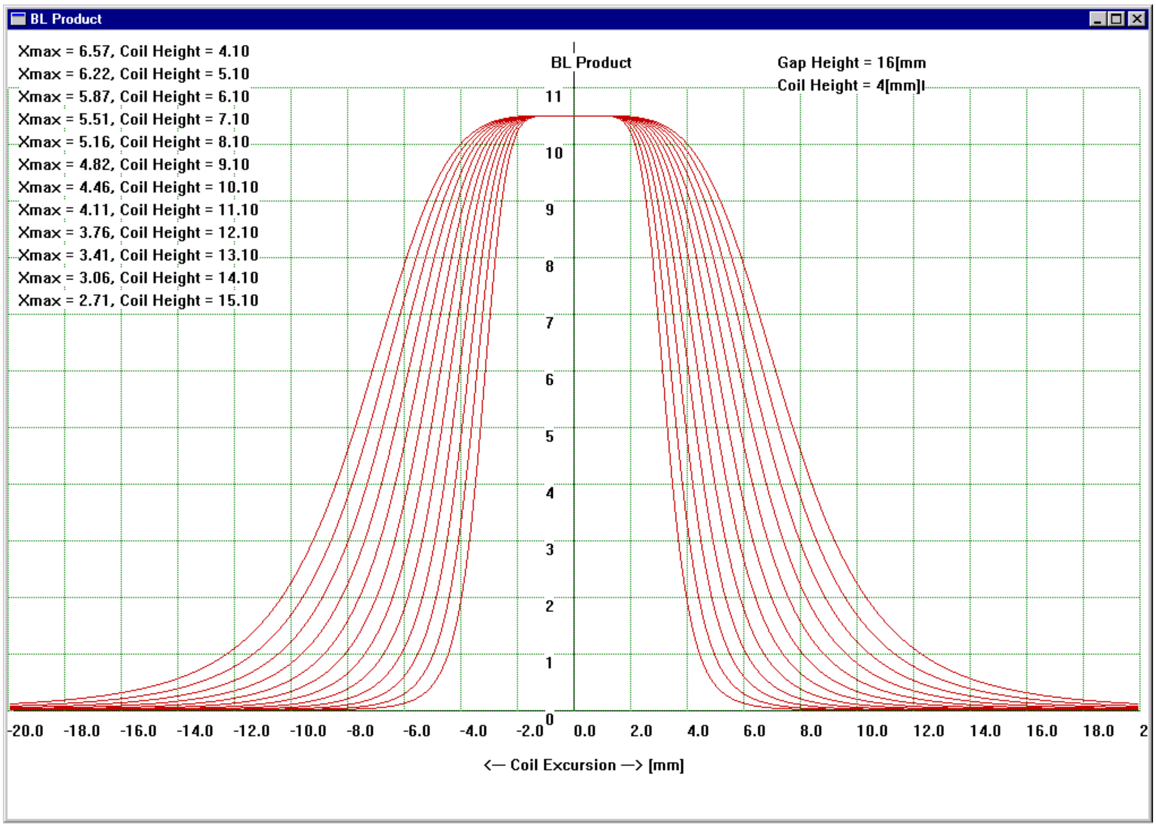


Figure 4.57. Example of the family of underhung Voice Coils

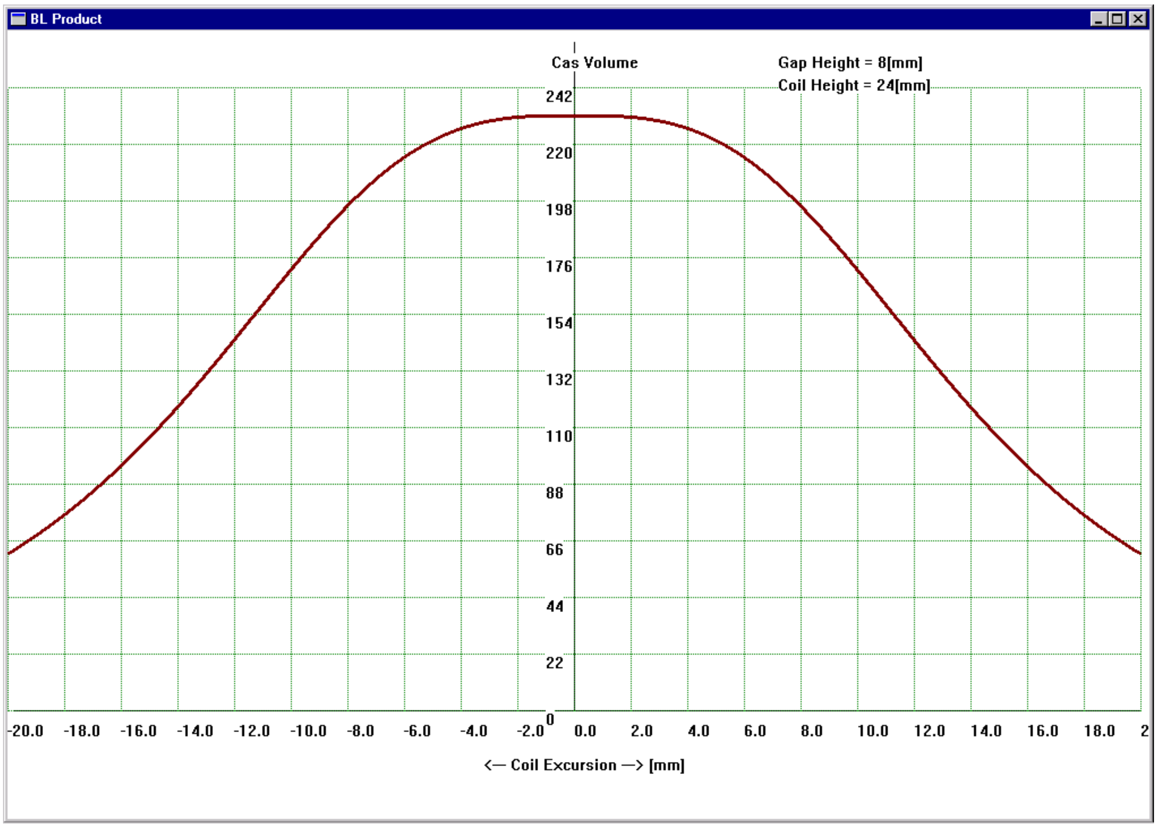


Figure 4.58. Estimating Cas(x) non-linearity. Assumed Xmax = 10.0mm

Estimating Xmax due to Cas(x)

Expression for Xmax due to the spider was developed in Chapter 2.

Sanity check

Having estimated Xmax due to BL motor strength and also Xmax due to stiffness of the suspension, it would be sensible to check if the computer model based on these assumptions was indeed of any practical use. Fortunately, we have a good reference point provided by the relationship between the cone excursion Xpeak, and SPL of the driver, widely developed in the literature:

$$X_{peak} = \frac{(1.18*10^3)10^{SPL/20}}{f^2 a^2}$$

Where: “f” is the test frequency in Hz and “a” is the radius of the driver in mm.

In this example a 25cm driver (125mm radius) at 52Watt input power and 50Hz test frequency, develops 105dB SPL, and cone excursion is ~5.6mm – the theoretical formula gives 5.37mm. We are **within 5%** from the desired value.

We would like to point out, that we are now in a good position to discard the option of not performing the large-signal analysis at all, as we do have a crude model, working with 15% accuracy. In summary:

1. We have examined physical design of the driver, looking at the voice coil length and magnetic gap height. These parameters are likely to be included in your driver specifications.
2. With the help of some crude mathematics, we have built my BL(x) curve, taking into account available and measurable references, eg: BL drops to 65-75% at Xmax (due to BL).
3. We have also visually inspected the design of the spider and from there, we calculated my Xmax due to the spider. Then, we developed my Cas(x) curve again, assuming, that just like for BL(x) curve, the 65-75% drop in Cas(x) value would correspond to cone excursion = Xmax (due to Spider), as shown on Figure 4.45.

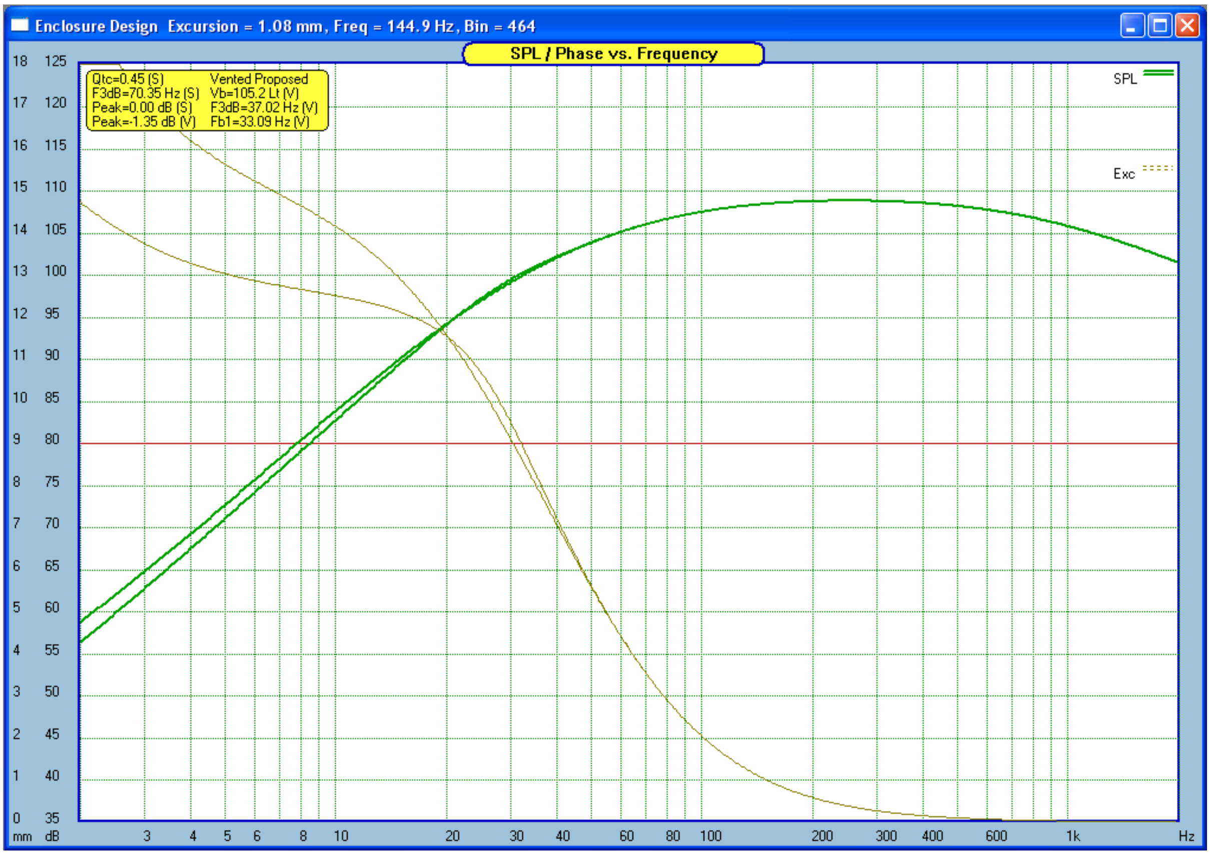


Figure 4.59. Checking the model of the example driver for accuracy.

How useful is this simple model ?. In order to evaluate this, we would like to take a look **at “suspension instability” issue and associated with it, “soft clipping”**. This problem would be more related to vented enclosures rather than sealed boxes, because sealed enclosure automatically offers “stiffer” suspension due to the air trapped inside the box. We would like to start with a quick remainder about spiders.

Suspension Instability

When a driver is operated at input power levels leading to excessive cone excursion (cone excursion > Xmax) there is a distinct possibility, that the voice coil will eventually “jump out” of the magnetic gap when approaching system resonance. When the coils is moved into much reduced BL(x) region, there is a dramatic increase in Qes of the loudspeaker. Remember, that electrical dumping (Qes) is the primary means that the amplifier uses to control the driver. Drivers with low Qes will be tightly controlled, while drivers with high Qes will exhibit very poor and “ringing” behavior in their impulse response. So we are now in a situation, where the amplifier looses the control over the driver excursion and if something is not done to prevent it, the voice coil will leave the magnetic motor system permanently. This situation is depicted on the Figure 4.47 below. The driver has 24mm long voice coil and 8mm magnetic gap. This gives us:

Xmax = (24mm – 8mm)/2 = 8mm due to BL(x)

It is also easy to observe, that the voice coil will be leaving the magnetic motor system at X > 16mm. There is some magnetic field extending above and below the top plate, so at X>16mm, the coil is above the top plate, but is still in some residual magnetic field. You would expect, that for high input power applied to the speaker, this field may still be able to push the coil a mm or so further out. Ideally, our simple model should be able to show this behavior. Now please consider two cases:

Case 1

The suspension system and motor systems are designed with the characteristics as shown on Figure 4.48. Here, the BL(x) is much more non-linear than Cas(x). In fact, this design would be characterised by distortions generated mainly due to BL(x) non-linearity.

The Cas(x) curve is much wider than BL(x) curve. It is likely, that a “long throw” woofer would be designed along these lines and chosen for this application.

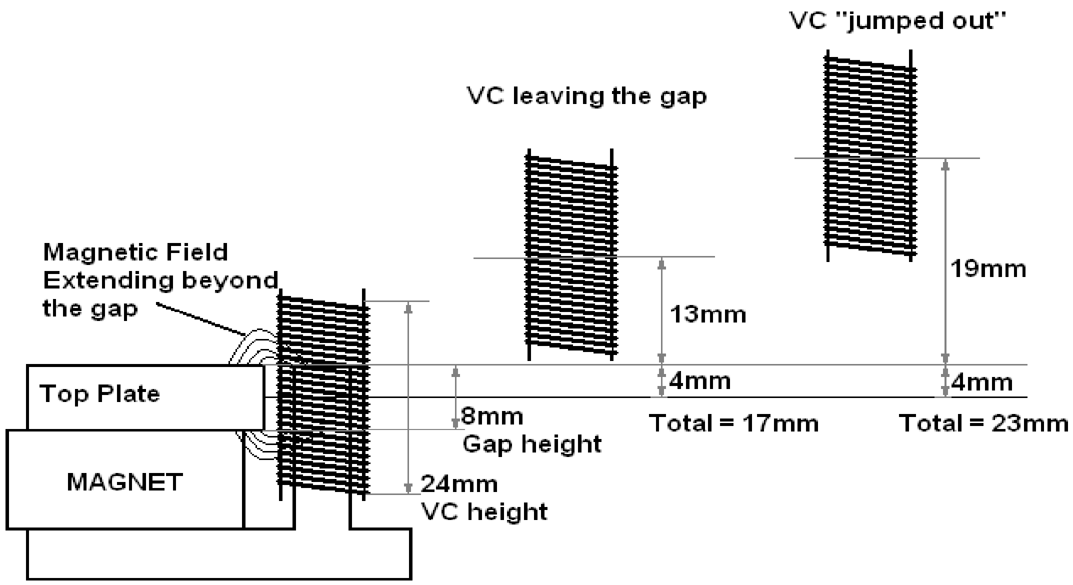


Figure 4.60. Excessive VC excursion leading to suspension instability

Case 2

The suspension system and motor systems are designed with the characteristics as shown on Figure 4.62. Here, the Cas(x) is more non-linear than BL(x). In fact, this design would be characterized by distortions generated mainly due to Cas(x) non-linearity. The Cas(x) curve is providing “soft clipping”.

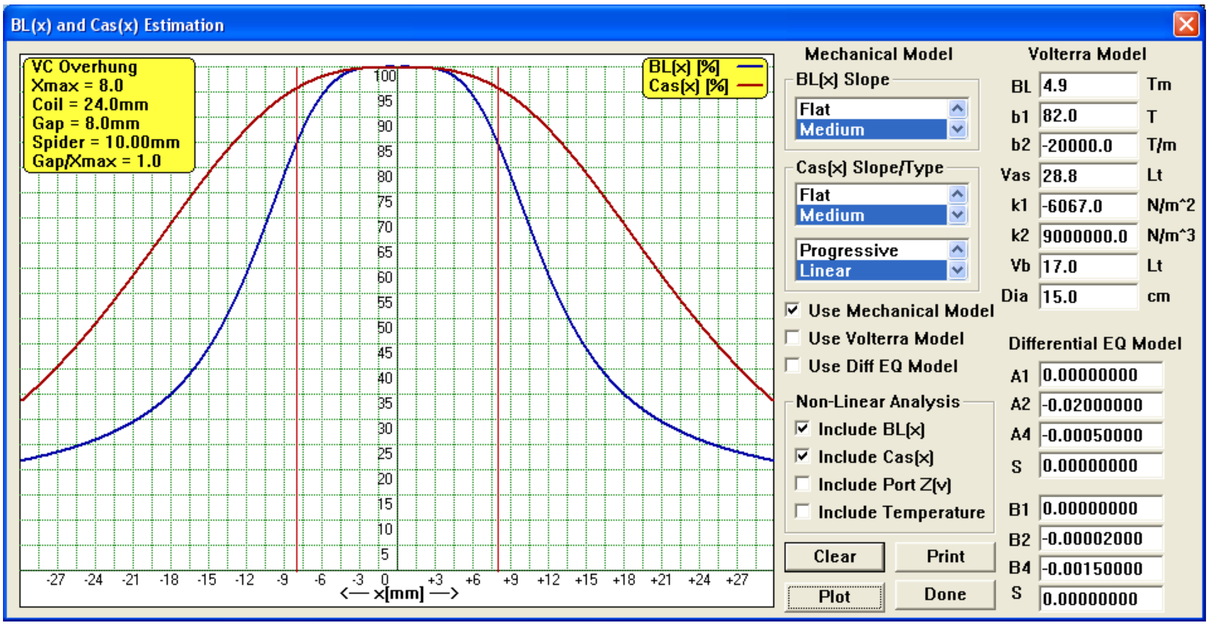


Figure 4.61. Very linear Cas(x) characteristics.

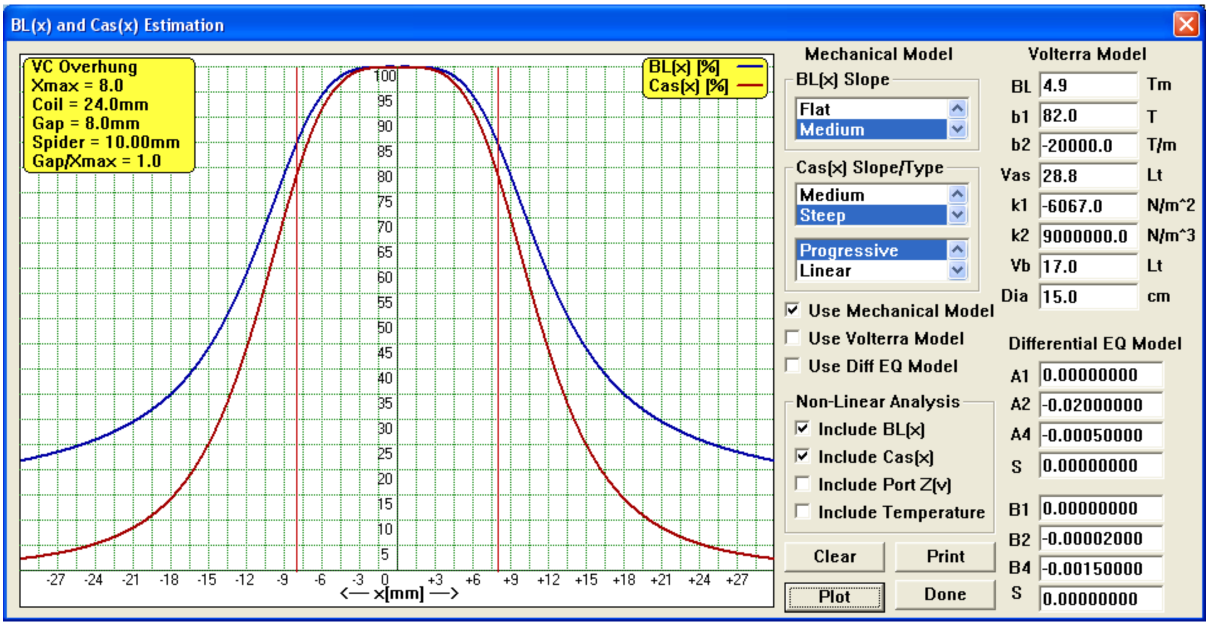


Figure 4.62. Cas(x) implemented as “soft clipping”.

Soft Clipping

Figure 4.61 shows the “high compliance” and very linear suspension case, leading to voice coil “lock out”. Initially, (curve “light brown”) the voice coil moves only as much as the combined effect of the relaxed suspension and BL(x) non-linearity allows for. At frequencies around 15-18Hz, the cone is approaching the lower resonant peak of the impedance (one of the system resonances) and the movement begins to be controlled by the mechanical losses in the system. The control is much poorer and this leads to further increase of excursions and the voice coil lands outside of the magnetic system. Figure 4.63 shows this situation, where curve labeled “light brown” is reaching 23mm of excursion and corresponds to Figure 4.60 – VC “jumped out” of the motor system.

As a remedy to this problem, the “progressive” suspension has been proposed. Again, Figure 4.63 shows clearly the effect of “soft clipping” provided by the “progressive” suspension system. This design adopted different characteristics for Cas(x). Now the spider does not allows the cone to move too far. The Cas(x) curve is much narrower and has stepper slopes designed to prevent the voice coil from traveling beyond the magnetic gap. Indeed, if a design shown on Figure 4.49 was implemented in the same driver, the cone excursion would be quite flat, as the voice coil is now heavily restricted from leaving the magnetic gap at any frequency. There is only a small rise at 15-18Hz, as opposed to the previous peak. It is easy to observe, that 12-13mm is just about as far as the spider and the rim can stretch for this input power level. Acoustic output monitored by a microphone would show symmetrical clipping.

Interesting observation can be made about the shape of the cone excursion curves shown on Figure 4.63. Cone excursion curve due to BL(x) looks quite different from the cone excursion due to the Cas(x) non-linearity. The BL(x) exhibits a sharp and pronounced peak around the system resonance frequency for the reason explained above. The cone excursion curve controlled mostly by the stiffness or Cas(x) is significantly flatter. This was to be anticipated, as the “progressive” spider with the characteristics shown on Figure 4.49 continues to be the dominant, movement restrictive element in the driver design.

Figure 4.63. Cone excursions. "light brown"- Cas(x) nearly linear, "green" – Cas(x) soft clipping, "red" – BL(x) and Cas(x) completely linear..

BL(x) and Cas(x) Dialog Box

As discussed previously, current release of the program provides you with an alternative method of estimating BL(x) and Cas(x) curves. Only three parameters are required to enable the large-signal evaluation:

1. Voice coil height
2. Magnetic gap height and
3. Estimation of the spider linearity – Xmax due to spider.

(1) and (2) are likely to be specified in the driver's data sheets. The spider's linearity, (3), can be estimated from the geometry of the spider. However, there can be additional information available, which provides you with more accurate estimation of the non-linear behavior of the driver.

For example: if the center pole of the motor system is extended above the top plate, it is likely, that linearity Xmax, due to BL(x) will be improved. The amount of the center pole extension will determine the improvement in linearity. One way of reflecting this additional data in the analysis, is to use provided dialogue box – "BL(x) and Cas(x) Estimation".

This box allows you to "expand" or "shrink" the BL(x) and Cas(x) curves. Option "Medium" from the "BL Slope" and Cas(x) list boxes will typically correspond to the BL(x) and Cas(x) curves dropping to 65-75% at x = Xmax.

An example of the suspension system and motor systems are shown on Figure 4.48, and Figure 4.49 The width of the BL(x) curve is calculated from coil and motor parameters. Therefore, another simple way of adjusting the width of the BL(x) curve is to simply modify the voice coil height and magnetic gap height. These two values are used only for calculating the Xmax due to magnetic system and can be easily tuned to obtain the best approximation of your BL(x) curve. The controls on the dialogue box are really self-explanatory. Simply select the settings that approximate your BL(x) and Cas(x) curves and continue with the analysis.

Harmonic Distortion

Total Harmonic Distortion (THD) coefficient is calculated as per formula given below. The nominator contains less summation factors, because we do not include the fundamental frequency (F_1) in the sum. Summation is performed over the first 64 harmonics. The Fi, denotes the level of each harmonic.

$$THD = \sqrt{\frac{\sum_{i=2}^{64} F_i^2}{\sum_{i=1}^{64} F_i^2}} * 100\%$$

The dialogue box simulating the THD is very simple and self-explanatory. All you need to do, is to **enter the desired power** level, Pin (Watts) and select enclosure type from the provided list box. Then click on the "Plot" button and you can watch the THD curve being plotted together with the harmonic content of the distorted signal.

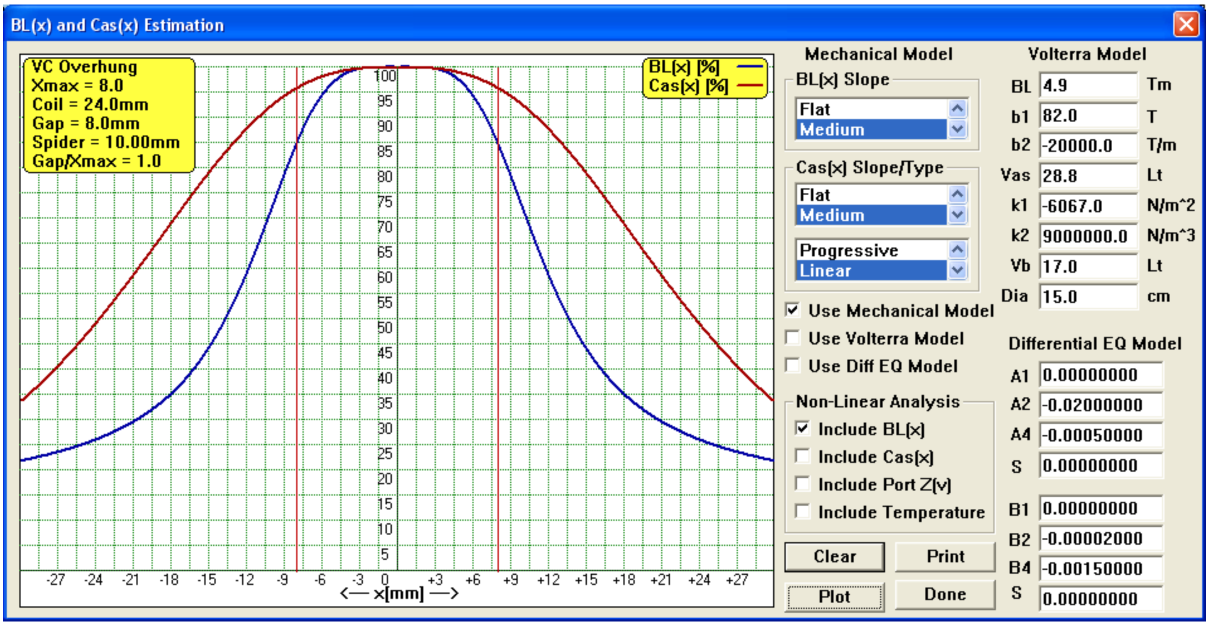


Figure 4.64. Example of BL(x) curve highly non-linear at low cone excursions.

The THD performance of the driver is highly dependant on the characteristics of the BL(x) and Cas(x) curves. The BL(x) curve exhibiting high non-linearity at low cone excursions and dropping more gradually (see Figure above) will produce higher level of THD at low to moderate cone excursions (10% at 42Hz) and lower THD levels at higher cone excursions (70% at 2Hz) – see Figure below. Conversely, the BL(x) curve as shown on Figure 4.61, will produce lower level of THD at low to moderate cone excursions (5% at 42Hz) and higher THD levels at higher cone excursions (83% at 2Hz) – see Figure 4.54. Quite similar analysis could be conducted for Cas(x) curves. The "soft clipping" issue discussed previously can only implemented at the expense of significant increase of harmonic distortion.

The THD dialogue box has also **"Fourier Analyser"** built-in. This unique function allows you to observe the level of the first 32 harmonics of the sweeping sine wave as it is reproduced by the non-linear driver. The test sine wave will be distroted accordingly to the BL(x) and Cas(x) nonlinearities. The symmetric nature of the BL(x) and Cas(x) curves will produce dominant odd-harmonics.

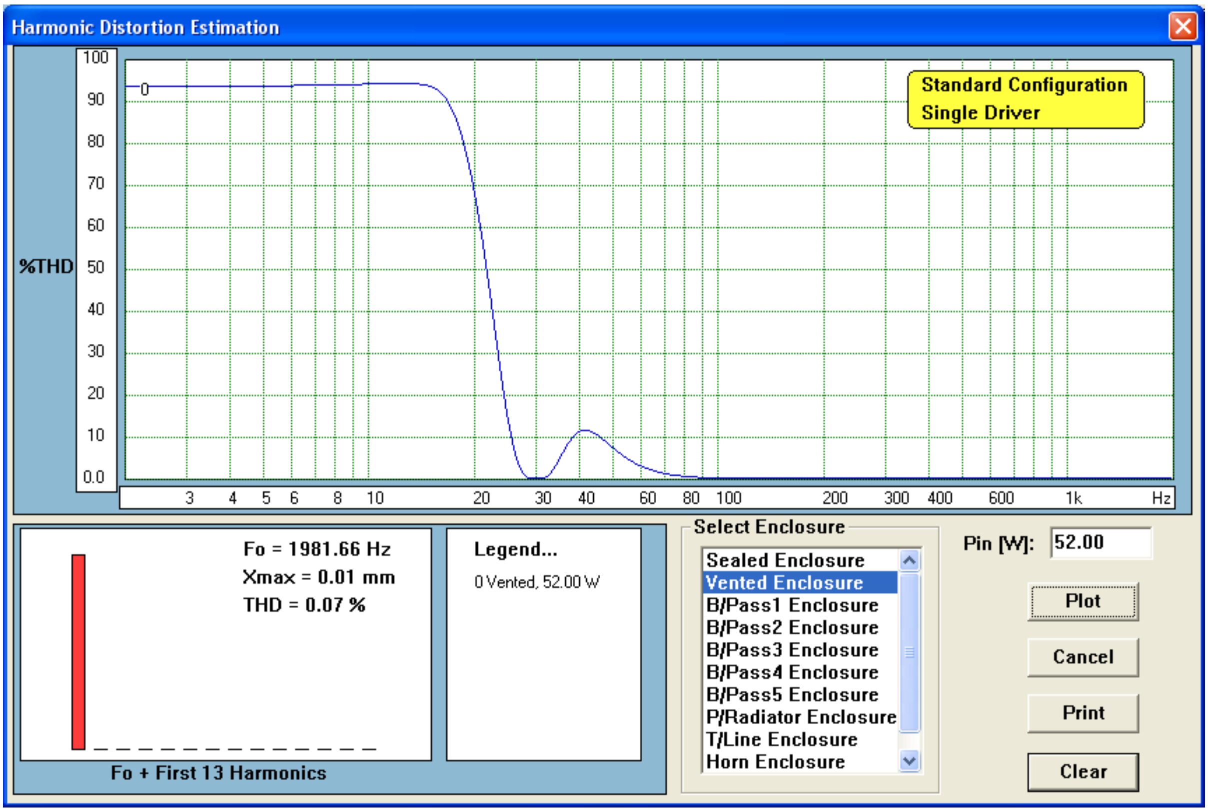


Figure 4.65. THD corresponding the BL(x) design shown on Figure 4.57

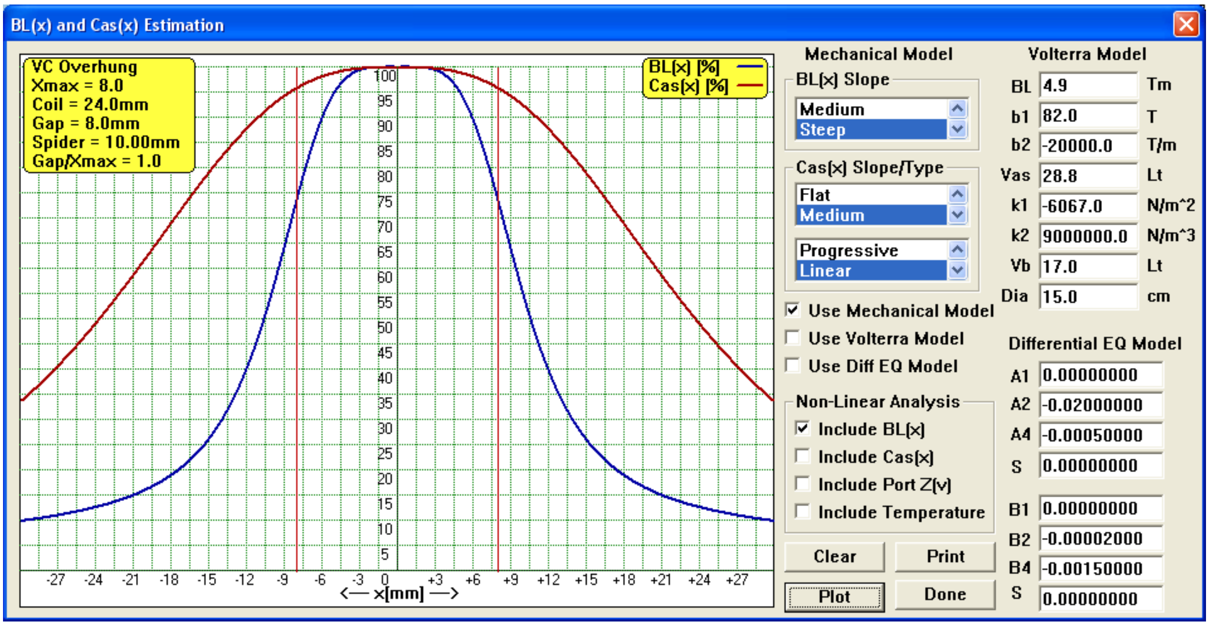


Figure 4.66. Example of BL(x) curve highly non-linear at high cone excursions.

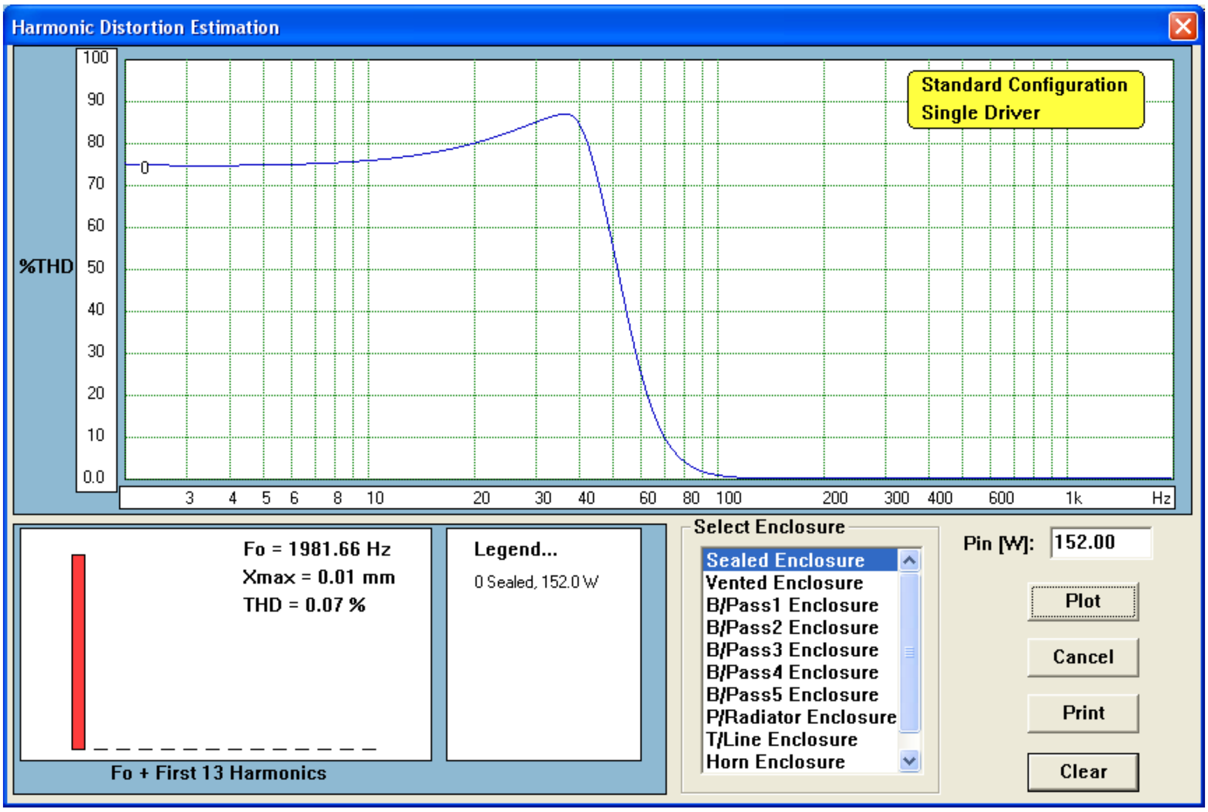


Figure 4.67. Example of harmonic distortion plot for 152W and sealed enclosure

Measuring BL(x) curve

The BL(x) curve can be measured quite simply and with minimum hardware. Here is the basic idea:

1. Place your woofer vertically up on a flat surface.
2. Apply mass M1, sufficient to displace the cone by -10mm to –12mm
3. Apply DC current I1 to the driver, to return the cone back to x = -1mm.
4. Apply different mass M2 (<M1) to displace the cone by -7mm to –9mm.
5. Apply DC current I2 to the driver, to return the cone back to x = –1mm.
6. Calculate:

$$BL(x) = BL(x = -1mm) = g\frac{M_1 - M_2}{I_1 - I_2}, \qquad g = 9.81\frac{m}{s^2}$$

You have now your first data point for BL(x = -1mm). The idea of returning the cone back to the same deflection x, allows you to nullify the influence of the stiffness in the BL(x) formula. Numerical value of stiffness will be the same for both masses, only if the cone returns to the same x-position.

Re-apply mass M1, but now apply enough DC current I1, to return the cone to x = -2mm. Then again, re-apply mass M2 and only enough DC current I2, to return the cone to x = -2mm. Once again, apply formula given in step (6) to calculate BL(x = - 2mm).

Continue on, until the BL(x) is completed within say, +/-10mm. You may have to turn the speaker upside-down and suspend the weights from the speaker to obtain BL curve for positive displacements. The deflections for M1 and M2 were given only as a guide. For “long-throw” woofers you may opt to use larger displacements – simply apply common sense here. After plotting all calculated BL(x) values, you can easily extrapolate the actual BL(x) curve. The next step is to select appropriate parameters in SoundEasy that approximate the measured BL(x) curve best.

Unfortunately, the same procedure can not be applied to stiffness (or 1/Cas(x)) curve. Unless, you have a hard copy of the Cas(x) curve (or stiffness plot) you will need to rely on the Cas(x) estimation from spider’s geometry. You may also be able to measure Cas(x) curve – as shown in Chapter 16.

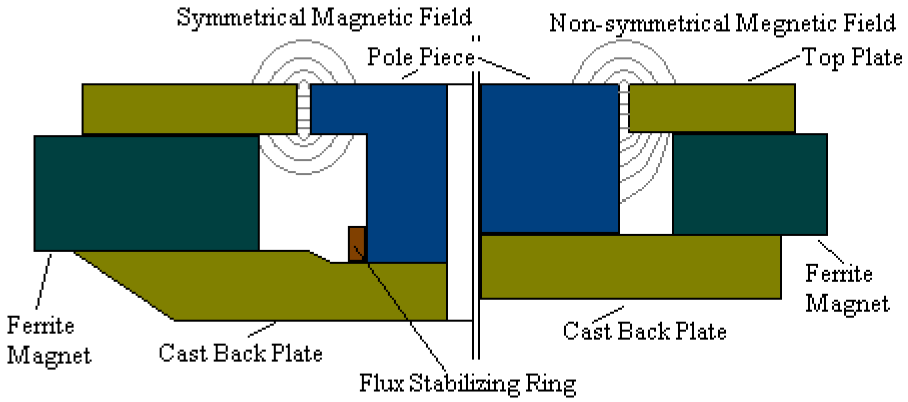


Figure 4.68. Example of a symmetrical magnetic field motor design.

Mechanical Model and Volterra Model

Measuring and modeling nonlinear behavior of a loudspeaker is a difficult process. We have taken the "step-by-step" approach to introducing these features into the program.

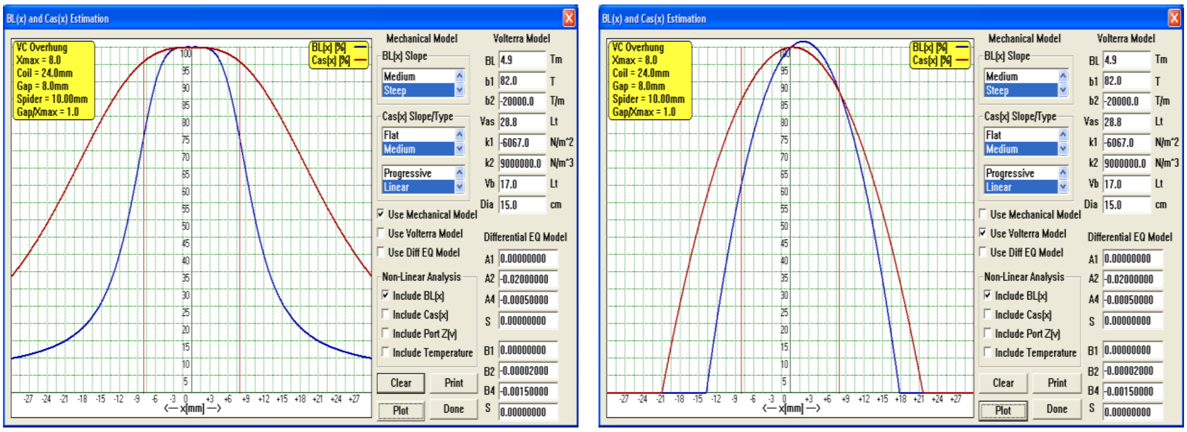


Figure 4.69. Mechanical and Volterra BL(x) and Cas(x) models.

Mechanical model of loudspeaker was our "first step" into the area of nonlinear operation of the loudspeaker. The model was simple and was able to relate easily visible and measurable mechanical parameters of the loudspeaker to it's performance at high input power. In this program release, we have implemented measurement system (refer to Chapter 16) for extracting non-linear parameters suitable for Volterra model of loudspeaker non-linearity. It can be seen clearly, that the two models produce different BL(x) and Cas(x) curves, particularly for larger cone excursion. This will have impact on all performance curves plotted on the "Enclosure design" screen for higher power levels.

The built-in truncated Volterra model has the advantage of providing a set of coefficients extracted and calculated from loudspeaker under actual operating conditions. This is important and would indicate, that at least the initial section of BL(x), Cas(x) and Le(x) curves is approximated reasonably well for -Xmax < x < +Xmax. For cone excursion values higher than Xmax, the truncation becomes quite evident. Also, loudspeakers with flatter section of the BL(x) and Cas(x) curves around -Xmax < x < +Xmax are not approximated well by the parabola-type of curves. The mechanical model holds a little better in this area and also provides more gradual and realistic "decay" of the BL(x) and Cas(x) curves for higher cone deflections. On the disadvantage side, this model is purely based on the mechanical properties of the loudspeaker. Also, the model is symmetrical for positive and negative cone deflections. There are several other non-linear models and measurement techniques proposed over the years by the scientific community. Some of them exploit MLS properties and others stay in time domain and propose to solve a set of differential equations in order to obtain a set of non-linear coefficients. We will continue to evaluate these methods and implement other techniques gradually into the program as well.

You can switch between the models by selecting **"Use Mechanical Model" or "Use Volterra Model" check boxes**. Volterra model parameters, that are directly affecting the shape of the curves are highlighted in red.

Differential Equation Model.

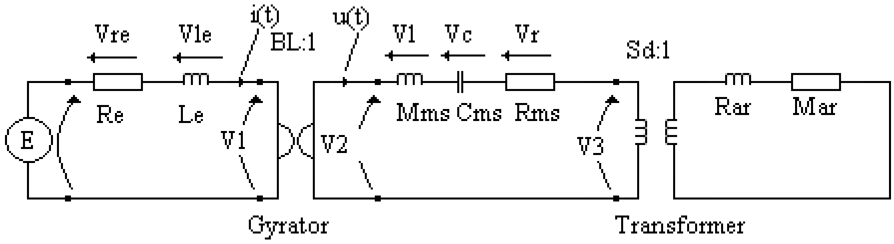


Figure 4.70 **Mechanical impedance** representation of a loudspeaker.

System equations can be derived starting with the basic of the electro-mechanical transduction.

$$F = Bli \quad and \quad e = Blv$$

Where: v = velocity of the conductor and i = current through the conductor

We can then introduce the nonlinear functions for BL and Cms. As far as the shape of the BL(x) and Cms(x) functions is concerned, one can employ an exponential function of the following prescription:

$$BL(x) = BL_0 e^{A4(x-S)^4 + A3(x-S)^3 + A2(x-S)^2 + A1(x-S)}$$

$$Cms(x) = Cms_0 e^{A4(x-S)^4 + A3(x-S)^3 + A2(x-S)^2 + A1(x-S)}$$

BLo = Static force factor
Cms_0 = Static mechanical compliance
A1 = Asymmetry about x = 0
A2 = Flat (low value) or Peaked (high value) around x = 0
A3 = Not used, but can be useful as absolute value, A3 = 0
A4 = Wide (low value) or narrow (high value) of the curve as it falls to y = 0
S = Left / Right shift of the whole curve

There are several advantages of using this function rather than simple quadratic approximation of the BL and Cms curves. The quadratic approximation (or polynomial approximation) tends to positive or negative infinite value for large excursions. The BL factor will clearly go to zero as the excursion increases. The exponential approximation is more realistic in following the BL(x) and Cms(x) shape of real loudspeakers.

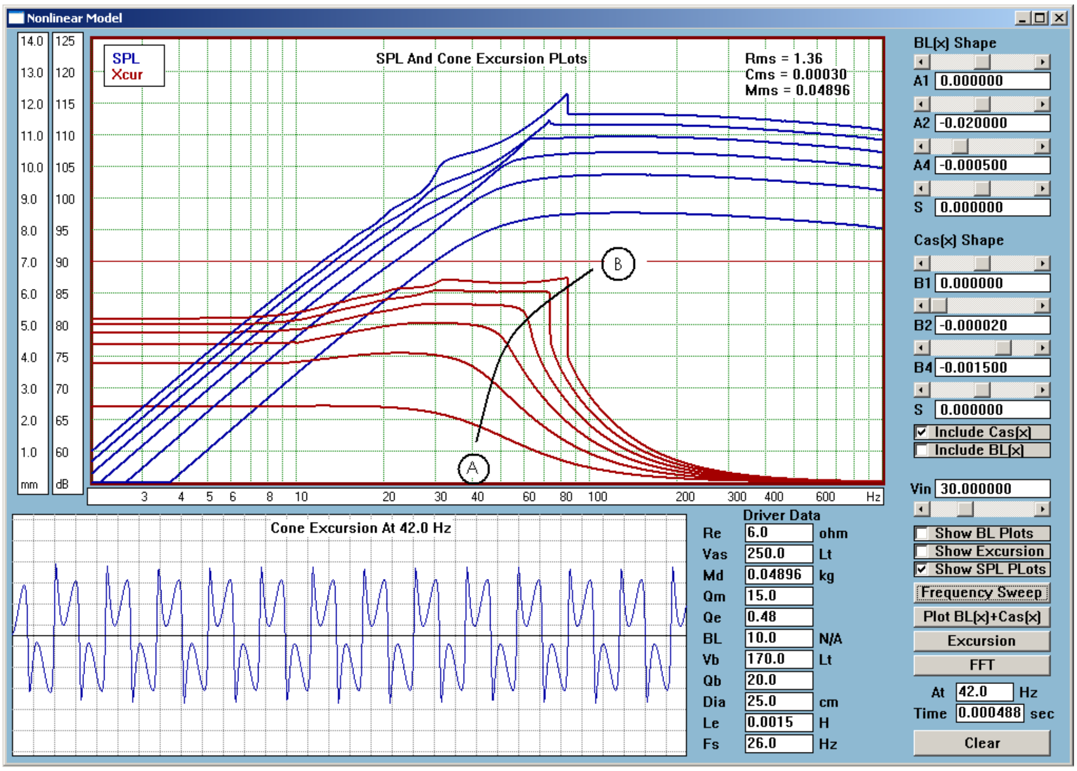


Figure 4.71. SPL and cone excursions for different input voltages – Cas(x) non-linearity only.

We can now consider the SPL and cone excursion of the loudspeaker for different input voltage levels in the frequency domain. It is clearly observable, that the resonant frequency of the system increases with the applied amplitude of the input voltage. This phenomenon is the result of increasing stiffness of the practical suspension systems, as the cone moves further and further away from the central position. As in the example above, at the extreme input voltages, one can observe the system resonance shift (moving from A to B) and finally, the vibrating system "jumping into another state" – the vertical portion of the cone excursion curve.

Differential Equation Model seems to be the easiest to understand and use. We recommend this model for your modeling work.

Enclosure Self-Resonance

The Thiele-Small approach to electro-mechanical behavior of a loudspeaker is based on "equivalent circuit" consisting solely of resistors, capacitors and inductors. This is also known as the "lumped element" model. The sound produced by the loudspeaker can then be obtained via a relatively simple circuit analysis. In the lumped element model, it is assumed, that driver's enclosure is represented by pure capacitor (or acoustic compliance Cab).

In real life, this is unfortunately not the case. Loudspeaker enclosure – just like any other enclosed volume of air, will exhibit it's own resonant frequencies – or modes. For the purpose of increasing the accuracy of modeling of the loudspeaker system, it would be very beneficial to incorporate modal behavior of the enclosure into the standard model.

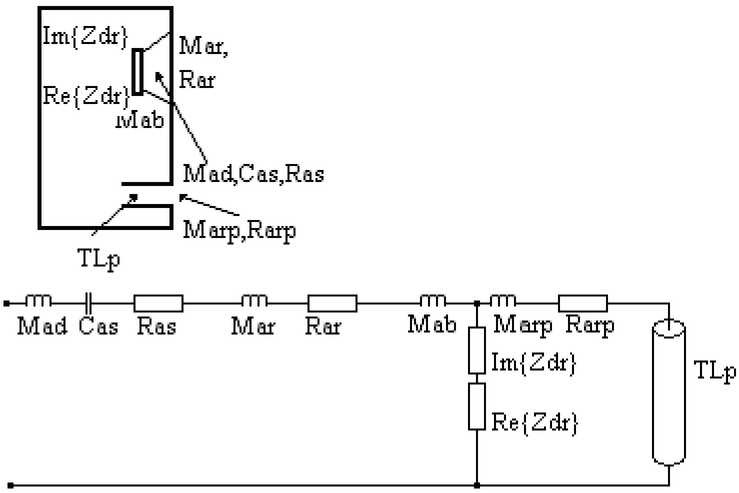


Fig 4.72 New acoustical impedance representation adopted for vented enclosure model.

The components are:

Cas = equivalent compliance volume Vas transformed to acoustical side.
Mad = mass of the vibrating system Mms transformed to acoustical side.
Ras = vibrating assembly loss Rms transformed to acoustical side.
Mar+Mab = air radiation of the front side of the diaphragm + air load of the back side of the diaphragm.
Rar = air radiation of the front side of the diaphragm.
Im{Zrd} = enclosure reactance (former compliance Vab) transformed to acoustical side.
Re{Zdr} = enclosure resistance (former absorption losses of the enclosure) transformed to acoustical side.
Marp = port radiation.
Rarp = port radiation.
TLp = short transmission line tube representing the port.

Possibly the best mathematical tool for evaluating modal behavior of fluids (air too) is **Finite Element Method.** More precisely, the **FEM** will be used to calculate driving point impedance, **Zdr**, that the driver's cone is exerting on the air in the enclosure.

First thing about FEM that one will notice, will be the long duration of these processes. Therefore, it is sensible to strive for a reasonable compromise between the accuracy of the analysis (number of nodes involved) and the length of the process. You will notice, that number of nodes in a vicinity of ~400, will give you acceptable accuracy up to 600Hz.

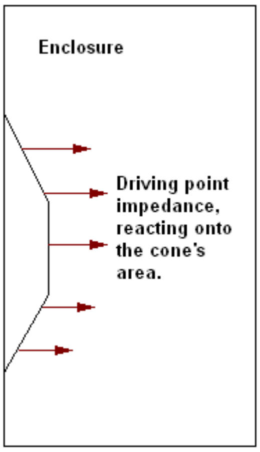


Figure 4.73 FEM allows you to replace enclosure reactance (**ω*Vb**) with **Zdr = A(ω) +jB(ω).**

Note: Before we proceed with explaining the FEM approach, please note, that the loudspeaker model implemented into SoundEasy incorporates the following:

1. Enclosure T/S analysis
2. Port resonance and non-linearity (compression)
3. Diffraction effects
4. BL(x), Cms(x) and Le(x) non-linearity
5. Enclosure modes (Zdr) analysis
6. Thermal power compression analysis

Diffraction effects and Enclosure modal analysis are position-dependent phenomena, therefore, a high level of attention must be paid to make sure, that the overall model is coherent, and all parameters are in agreement with each other. For instance, baffle location of the driver for Diffraction Analysis must correspond to the location of nodes in Enclosure Modal Analysis.

In this software release, you have a choice of using FEM with “brick elements” or FEM with Delaunay triangulation (or wedge elements) for sealed and vented enclosures. Future releases of the program will implement this method for dual-chamber enclosures as well, with brick elements for one chamber and wedge elements for the other. Wedge elements are often a better choice for modeling tilted enclosures, or enclosures with sharp edges.

A brief over view of the process is given first.

To implement either the brick or Delaunay method the following windows must be accessed: First, the driver file must be opened or conventional T/S data must be entered in the Driver Editor screen. Then the Preferences screen should be opened to set the frequency range for the Box and FEM analysis. Then, you must set-up the enclosure parameters in the FEM system. Now, go back to the Enclosure Design system and select the desired FEM option and enable the port resonance, if it is a ported box is to be computed. Then you must select the correct enclosure type, sealed or vented, and be sure System SPL is checked. Only one enclosure type should be checked. Then enter the desired Qb and port properties for a vented box and click plot. The process takes a long time. If your computer has minimum memory and/or speed the screen may appear to lock up. However, allow the code to run and when completed the screen will refresh correctly. After the frequency response has been plotted you can change Qb and/or port properties and produce a second frequency response plot without returning to the FEM screens. If you which to compare the result to the conventional response plot open the Vent/PR Calculator screen and uncheck the “Use FEM for Enclosure Modes” check box. Be sure that the correct box volume is entered in the Enclosure Design Dialogue and then click plot. First, we are going to take a look at the FEM with brick elements.

Enclosure Modes in FEM Brick-Element Model

Here is a simple description of how to use the built-in enclosure model.

1. Load a test driver file. The example driver will use 170Lt box volume and Rear Box Qb=20.

2. Go to **Preferences** screen and select frequency range for Box – type of screens. For example 1Hz – 1000Hz. This will set the frequency range of the FEM analysis.

3. Open **Room/Car 3D** FEM system – FEM Process Control TAB box will appear..

4. Select “**Fr/Response**” TAB and click on “**My Enclosure**” check box. This will ensure two things: (1) the wire mesh will display nodal numbers on the front panel, so you can easily select which nodes will represent driver’s cone area, and (2) the enclosure modeling process will use brick element FEM representation of the enclosure.

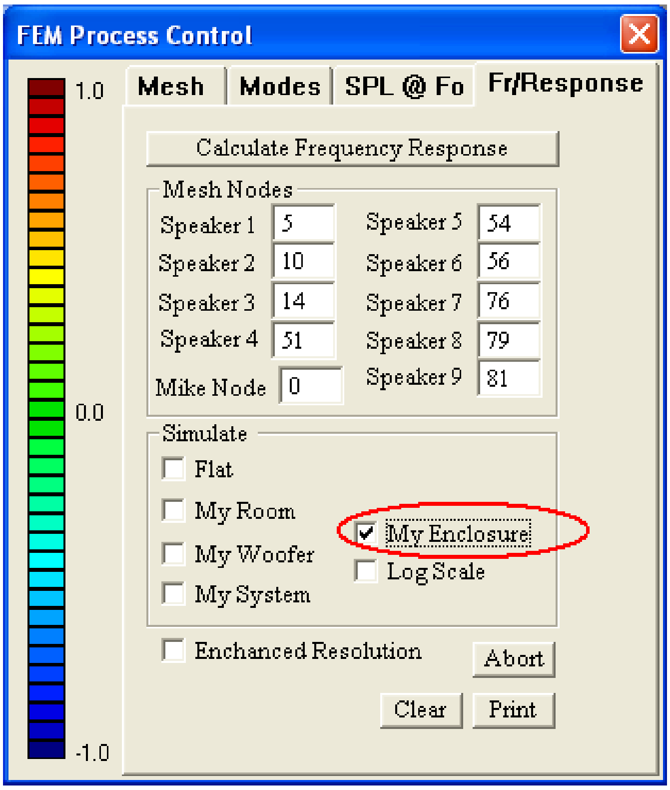


Fig 4.74 FEM brick element control box

5. Go to “**Mesh**” TAB and select “**Loudspeaker 275 nodes**” built-in option from “**Load Models**” list box. This is done for your convenience at this stage. Normally, you would built your enclosure from “brick elements”, and then allocate nodes mimicking driver location. You can have 4 OR 9-nodes ONLY to represent the driver.

This model is a simple 173Lt rectangular enclosure. The size of the brick element is X=12cm, Y=12cm and Z=7.5cm. There is 160 elements in this model and 275 nodes. The footprint of the enclosure is 4 x 4 elements, and there are 10 layers vertically.

This model will suffice till about 500Hz, but will run faster than more complicated model. If you need more accurate analysis at higher frequencies, you will need to built a more dense model. In this case, the footprint may be as big as 6 x 6 elements or similar.

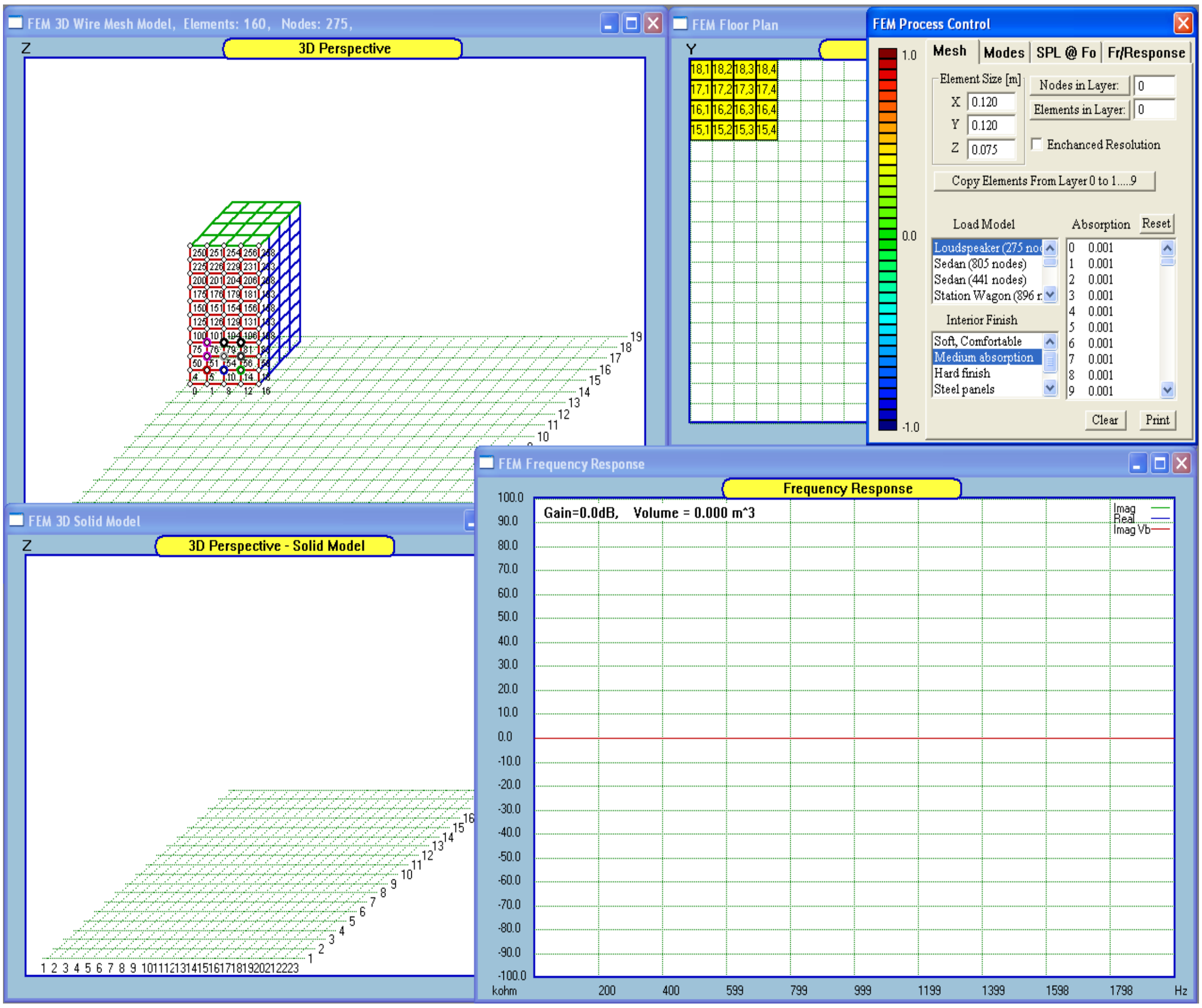


Fig 4.75 Example of a rectangular enclosure with front panel nodes shown.

6. Go back to “**Fr/Response**” TAB and enter node numbers you wish to use. For the built-in model, all nodes are already entered for you. Also, box volume will be calculated, based on brick element size. As you can see, the example enclosure volume is 0.173m^3 = 173Lt. This is close enough.

   If you need to change the enclosure volume or dimensions, at this stage, just go back to “**Mesh**” TAB and enter slightly different Element Size (X, Y or Z). When you go back to “**Fr/Response**” TAB, you will see the new enclosure volume re-calculated.

7. **IMPORTANT**: “**My Enclosure**” check box will enable FEM brick Element model during enclosure frequency response calculation.

8. In the next step, go to “**Enclosure Design Control**” and select on the “**Enclosure Parameters**” TAB:

   “**Sealed**” enclosure,
   “**Standard**” check box,
   “**Rear Box Qb**” – enter desired value
   “**FEM For Box Modes**” check box

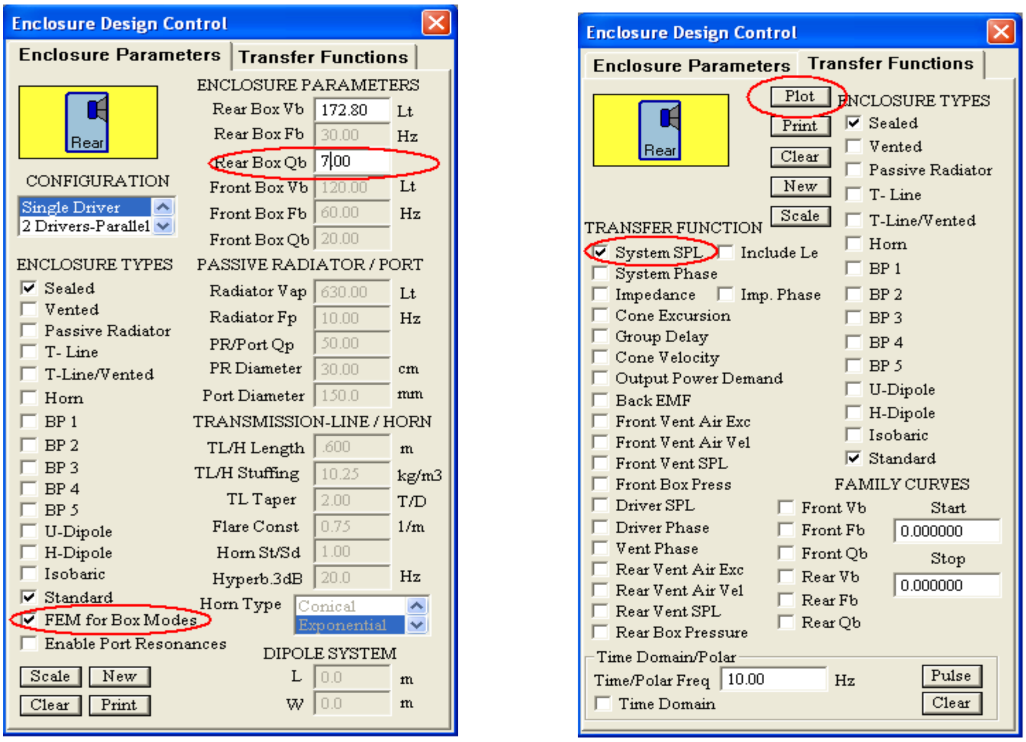


Fig 4.76 Enclosure Design box with marked controls for FEM analysis

9. Now select “**Transfer Functions**” TAB and check the “**System SPL**” box only. Then press “**Plot**” button. You will notice a progress bar will pop-up and indicate status. The process is LONG.

You will notice, that the enclosure volume calculated by the FEM screen will be transferred automatically to the driver’s file.

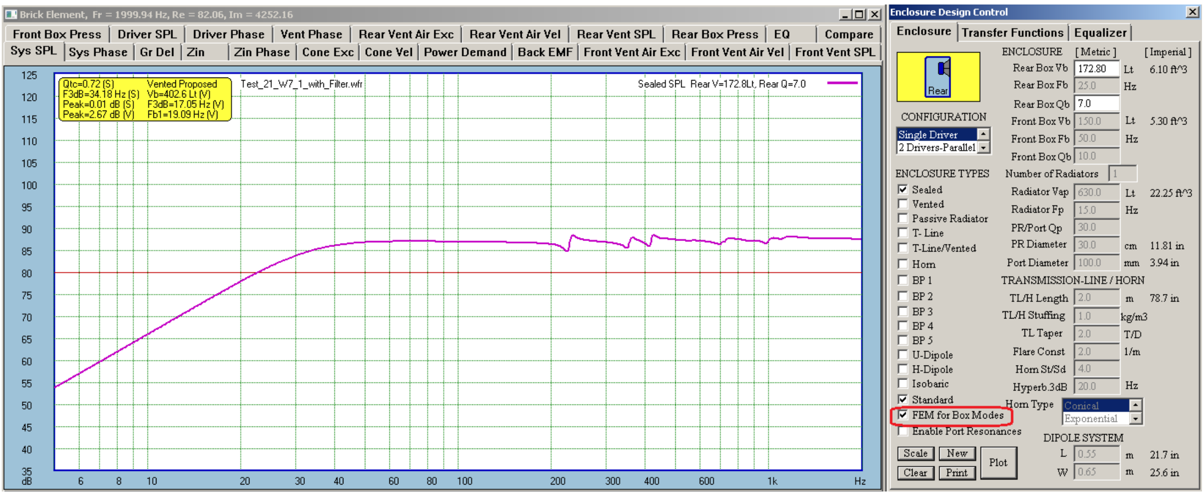


Fig 4.77 Example of a sealed enclosure SPL with FEM modal analysis

10. Finally, you can go back to the FEM system, and on the “**Fr/Response**” TAB press “**Calculate Frequency Response**” button to calculate the driving point impedance of the enclosure. Picture below was plotted for Qb=20. Simulation of “**My Enclosure**” on the FEM system predicts driver’s driving point acoustical impedance, Zdr, looking into the enclosure. The Q-factor (Qb) is selected from the Enclosure Design Control box.

Blue plot = Real part of Zdr.
Green plot = Imaginary part of Zdr (reactance).
Red plot = w*Cab, reactance, if the box was pure compliance (as in the standard T/S model).

It is easily observable, that Zdr deviates from the T/S model well below the first modal frequency of the enclosure.

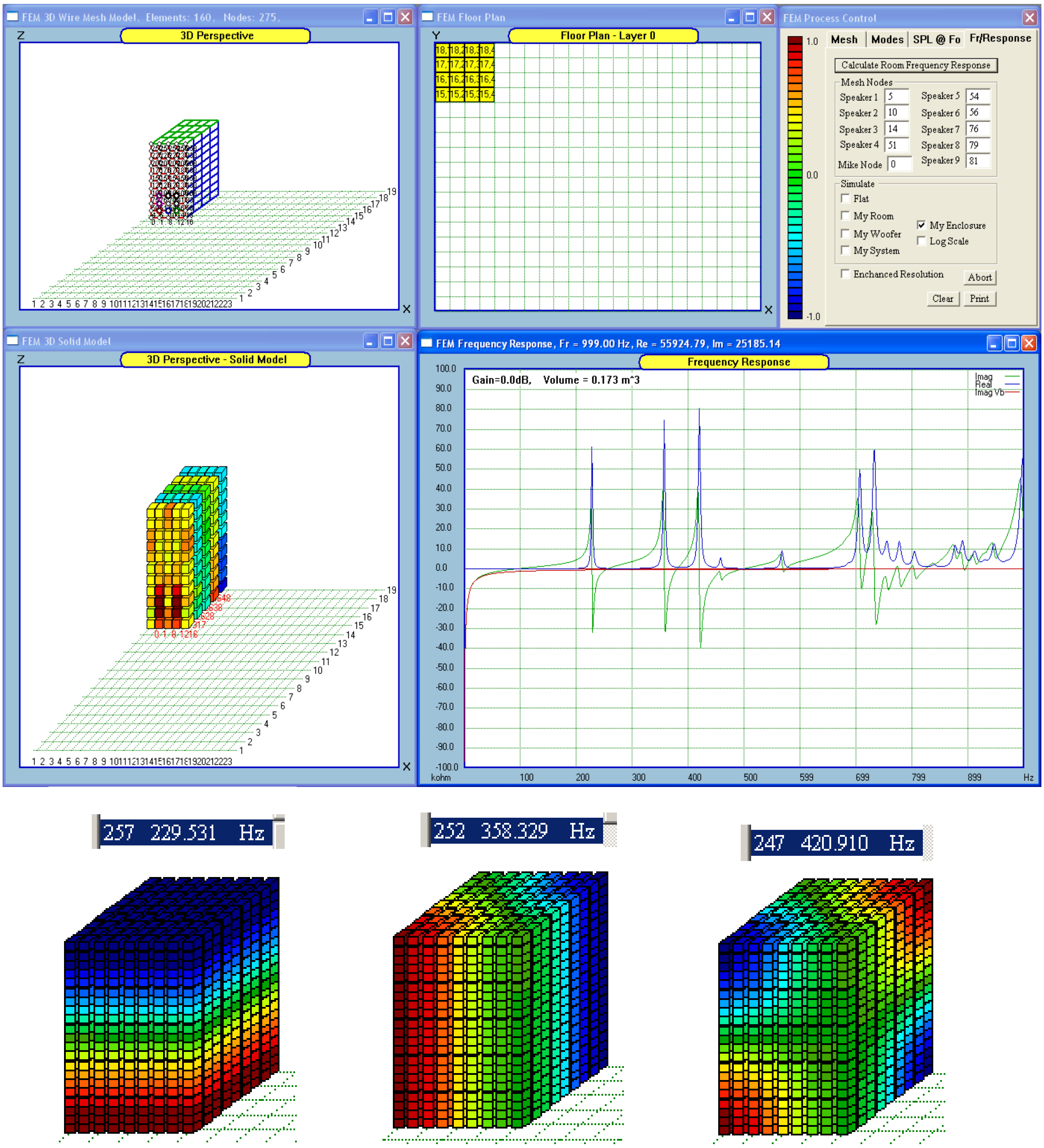


Fig 4.78 Driving point impedance and some box resonances of a simple rectangular enclosure

There is a need to locate the driver on the front baffle for diffraction effects and select corresponding nodes in the FEM box model. Also, the size and shape of the front baffle, used for diffraction analysis needs to be the same as the front baffle resulting from element dimensions and used for Fem modal analysis. Currently, it is user's responsibility to make sure, driver location and node locations fit together.

The built-in enclosure is actually very useful. You can adjust the volume and shape of the enclosure by selecting different element X, Y, and Z dimensions. Certainly, if more extended frequency range is desired, or perhaps mode accurate determination of enclosure modes is necessary, then you will need to construct more accurate (finer) mesh. As a consequence, you will experience longer calculation times.

Constructing the mesh from "brick" and "wedge" elements, isn't complicated, is given in Chapter 15, therefore will not be introduced here.

Example of an SPL plot for a sealed enclosure, for Qb = 7 is shown below. During the run of the FEM enclosure modal analysis, you will notice, that the plotting window title bar shows progressive calculations of Zdr, and is indicative, that this rather complex process is running properly.

Important Note: The FEM analysis involves long and complex mathematical processes. In order to speed-up calculation time, some sections of the program were written in assembler (machine code) language. Therefore, when running the FEM analysis, the CPU usage will be 100%. We strongly suggest, that you let the process run to completion, without attempting to use your PC for anything else.

Port Resonance and Enclosure Modes

The SPL plot below, were generated for the same enclosure (Vb=170Lt, Qb=7, Fb=30Hz) , the only difference being – the second picture has port diameter set to 150mm (instead of 100mm, as on the first picture).

It is observable, that resonating port, interacts significantly with self-resonating enclosure, and produce a substantial level of SPL. Since the port subtracts from the cone’s output, the higher the level from the port, the more irregular the system SPL becomes.

Secondly, larger ports will be much longer, therefore, the fundamental resonant frequency of the port, will easily fall into the audio range designated for the woofer driver – causing undesirable SPL variations. This is quite unfortunate, since it is desirable, to have as large port as possible, in order to minimize undesirable port compression and non-linearity at higher port output. It seems, that there are conflicting requirements for the port dimensions. Therefore, it may be beneficial to evaluate vented woofer systems not only for the “piston” range of the driver, but also taking into account resonant modes of the port and the enclosure.

For instance, Figure 4.66 below clearly shows, that enclosure reactance become positive above ~96Hz (green curve), and the resistive part of Zdr is not a flat (enclosure loss, or pure resistor in T/S model) curve either. However the standard “lumped element” model obviously continues to behave as the capacitive reactance (red curve). This discrepancy would suggest, that the system designer needs to be aware of the possible weaknesses, when working with the standard “lumped element” model.

Guided by the plots below, one could draw the conclusion, that larger ports will require lower cut-off frequencies of their LP system filters, so that the driver is not forced into the frequency range, where the port-enclosure interaction begins to affect the smoothness of the SPL.

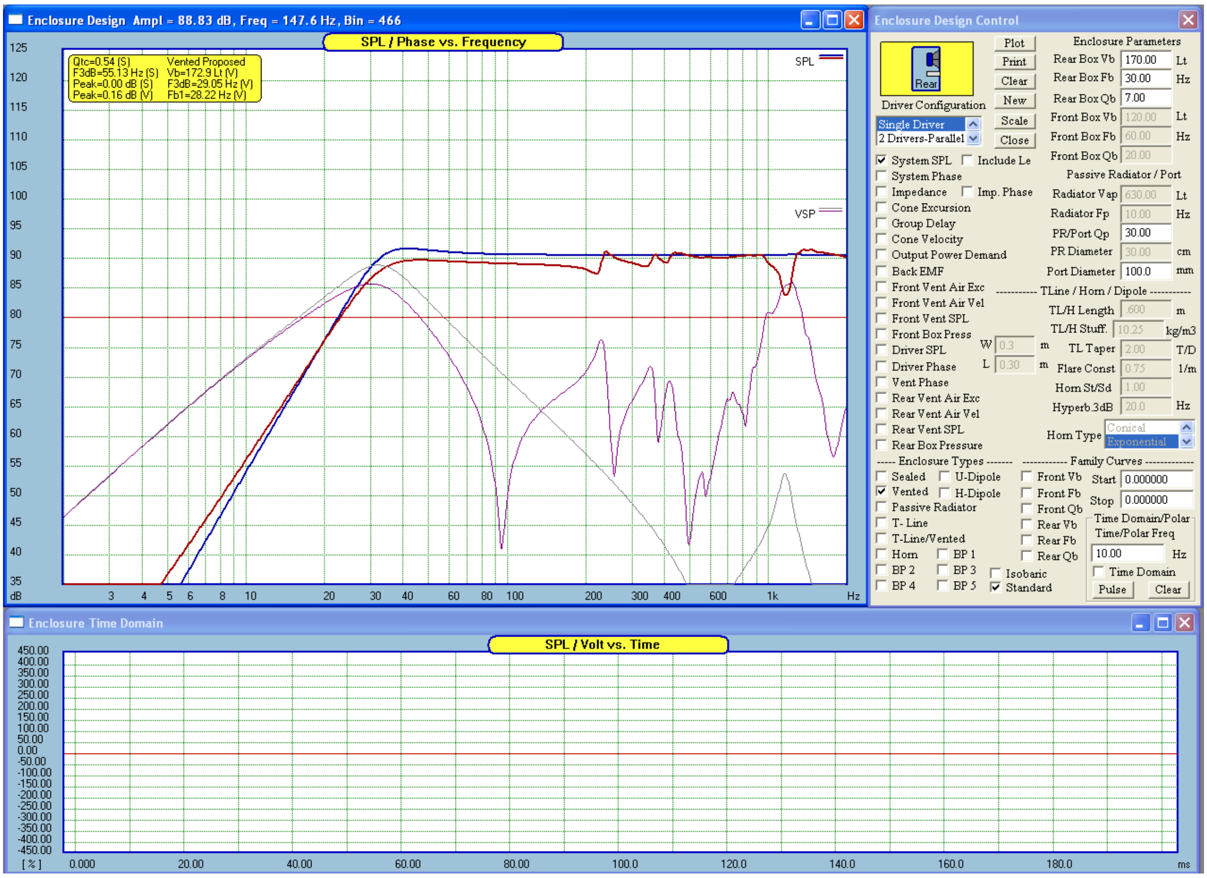


Fig 4.79 Vented enclosure SPL with Qb = 7 and port diameter = 100mm.

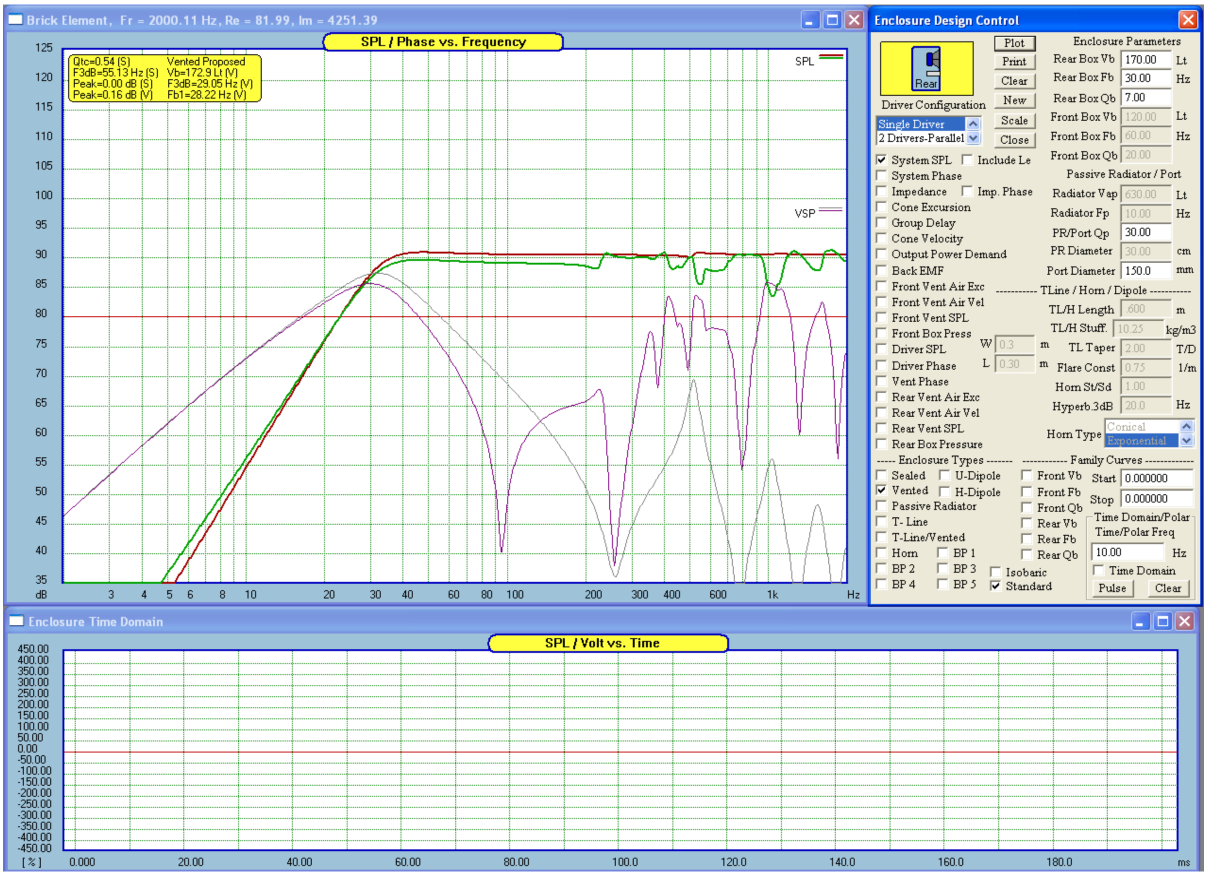


Fig 4.80 Vented enclosure SPL with Qb = 7 and port diameter = 150mm.

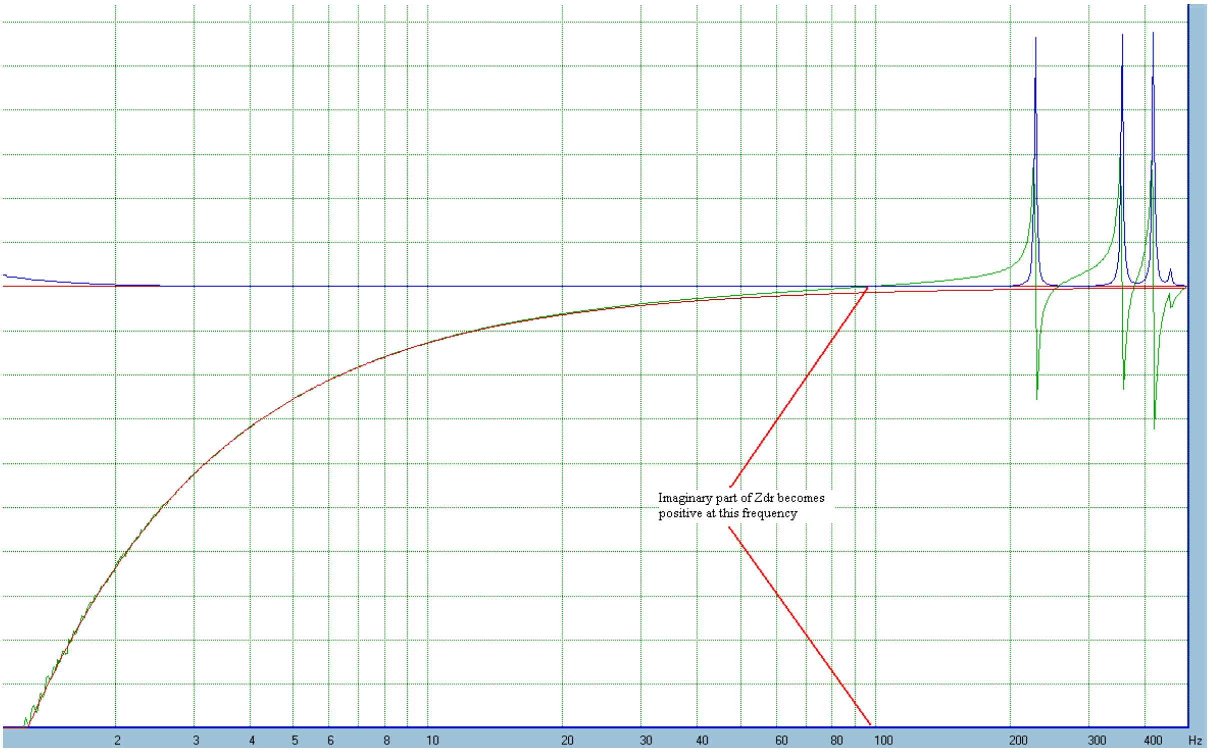


Fig 4.81 Transition frequency above which the “lumped element” model deviates from FEM enclosure model.

Enclosure Modes in FEM Wedge-Element Model

Here is another simple description of using built-in enclosure model. This time, it is "wedge element" based FEM, and we will only calculate the Zdr.

1. Load a test driver file. Our example driver will use 170Lt box volume and Rear Box Qb=20.
2. Go to **Preferences** screen and select frequency range for Box – type of screens. For example 1Hz – 1000Hz. This will set the frequency range of the FEM analysis.
3. Open **Room/House DEL 3D** FEM system.

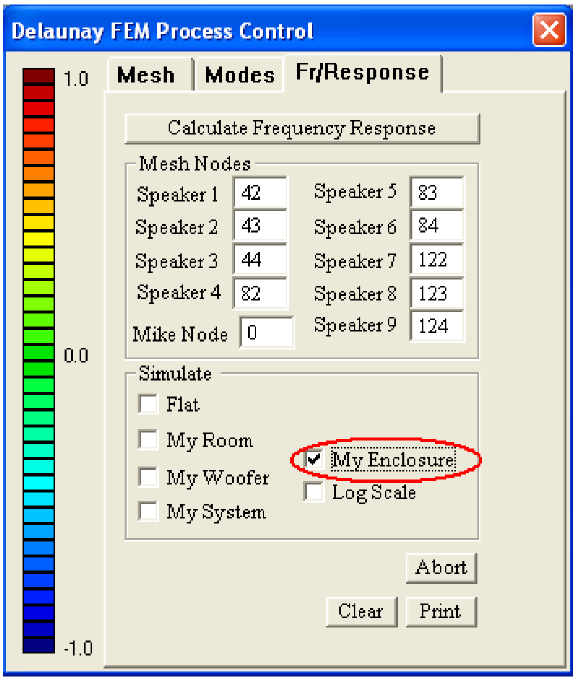


Fig 4.82 "Wedge" element control box

Select "**Fr/Response**" TAB and click on "**My Enclosure**" check box. This will ensure two things: (1) the wire mesh will display nodal numbers, so you can easily select which nodes will represent driver's cone area, and (2) the enclosure modeling process will use wedge element FEM representation of the enclosure.

4. Select "**Mesh**" TAB.

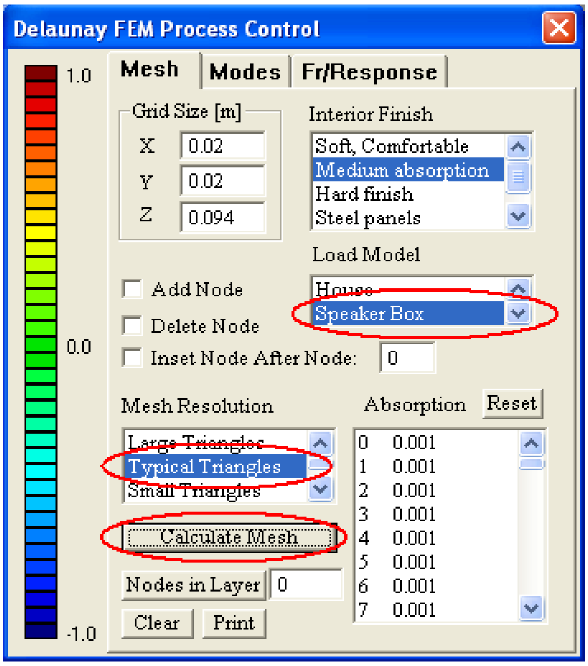


Fig 4.83 "Wedge" element control box – Mesh TAB with marked controls

Select: “**Speaker Box**” – to use built-in model
“**Typical Triangles**” – to obtain about 360 Nodes in FEM mesh.
“**Calculate Mesh**” – button to generate mesh.

Following, you should now see something like the picture below. One can review Nodes layer-by-layer to determine which nodes should be used to mimic the loudspeaker’s cone.

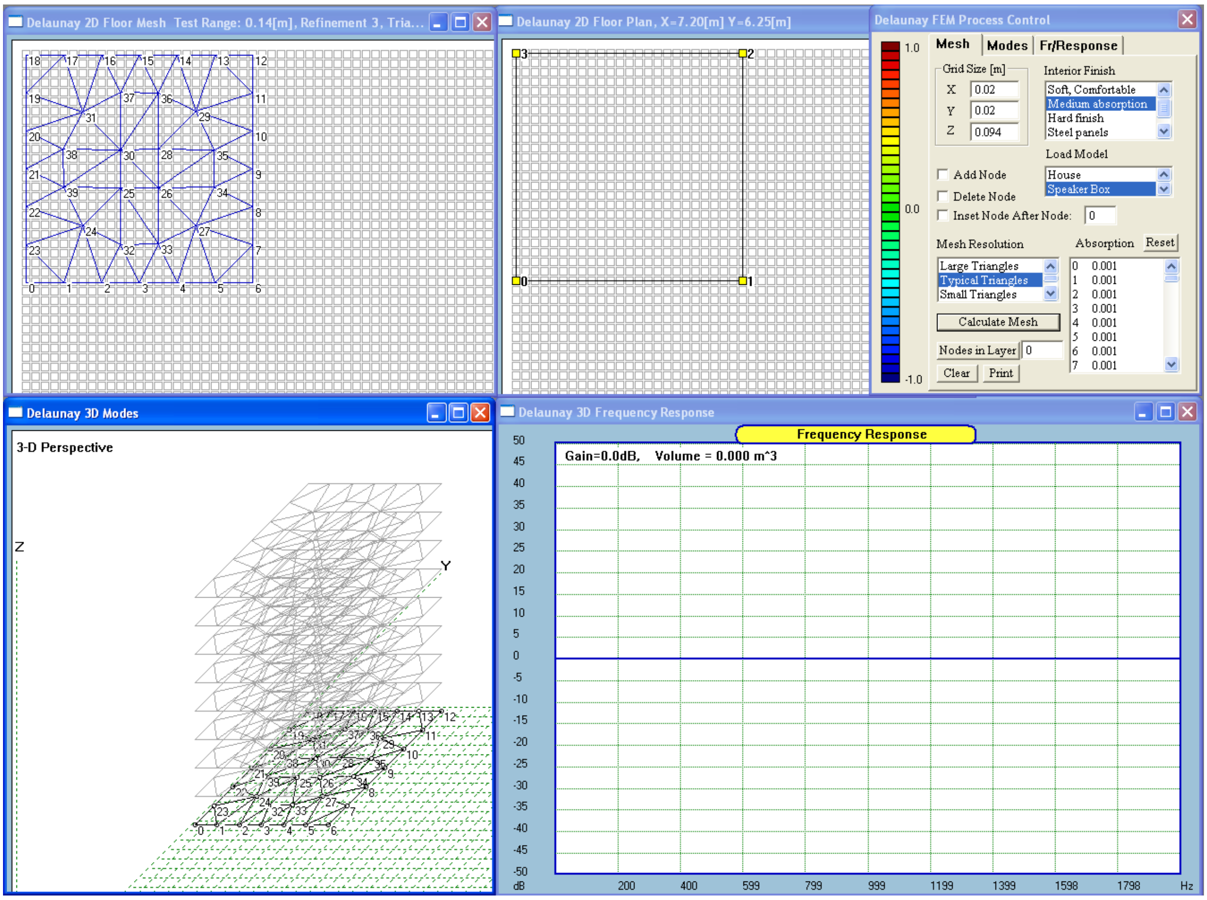


Fig 4.84 Example of “Speaker Enclosure” with node numbers shown in Layer 0.

5. Go back to “**Fr/Response**” TAB and enter node numbers you wish to use. For the built-in model, all nodes are already entered for you. Also, box volume will be calculated, based on brick element size. As you can see, the example enclosure volume is 0.173m^3 = 173Lt. This is close enough.
   If you need to change the enclosure volume or dimensions, at this stage, just go back to “**Mesh**” TAB and enter slightly different Element Size (X, Y or Z). When you go back to “**Fr/Response**” TAB, you will see the new enclosure volume re-calculated.

6. **IMPORTANT**: “**My Enclosure**” check box will enable FEM wedge Element model during enclosure frequency response calculation.

7. For now, press “**Calculate Frequency Response**” button to calculate the driving point impedance of the enclosure. Picture below was plotted for Qb=20. Simulation of “**My Enclosure**” on the FEM system predicts driver’s driving point acoustical impedance, Zdr, looking into the enclosure. The Q-factor (Qb) is selected from the Enclosure Design Control box.

Blue plot = Real part of Zdr.
Green plot = Imaginary part of Zdr (reactance).
Red plot = w*Cab, reactance, if the box was pure compliance (as in the standard T/S model). It is easily observable, that Zdr deviates from the T/S model well below the first modal frequency of the enclosure.

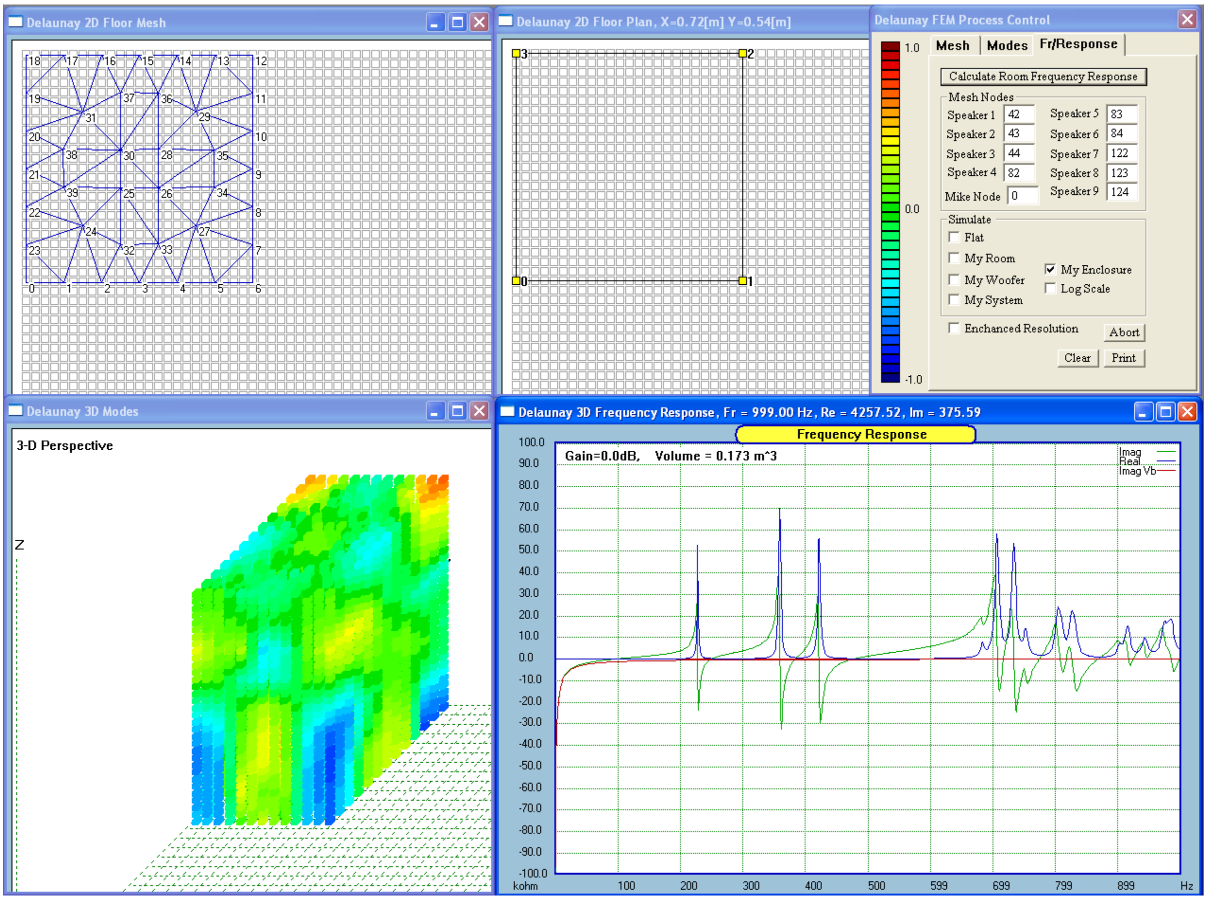


Fig 4.85 Driving point impedance of a simple rectangular enclosure

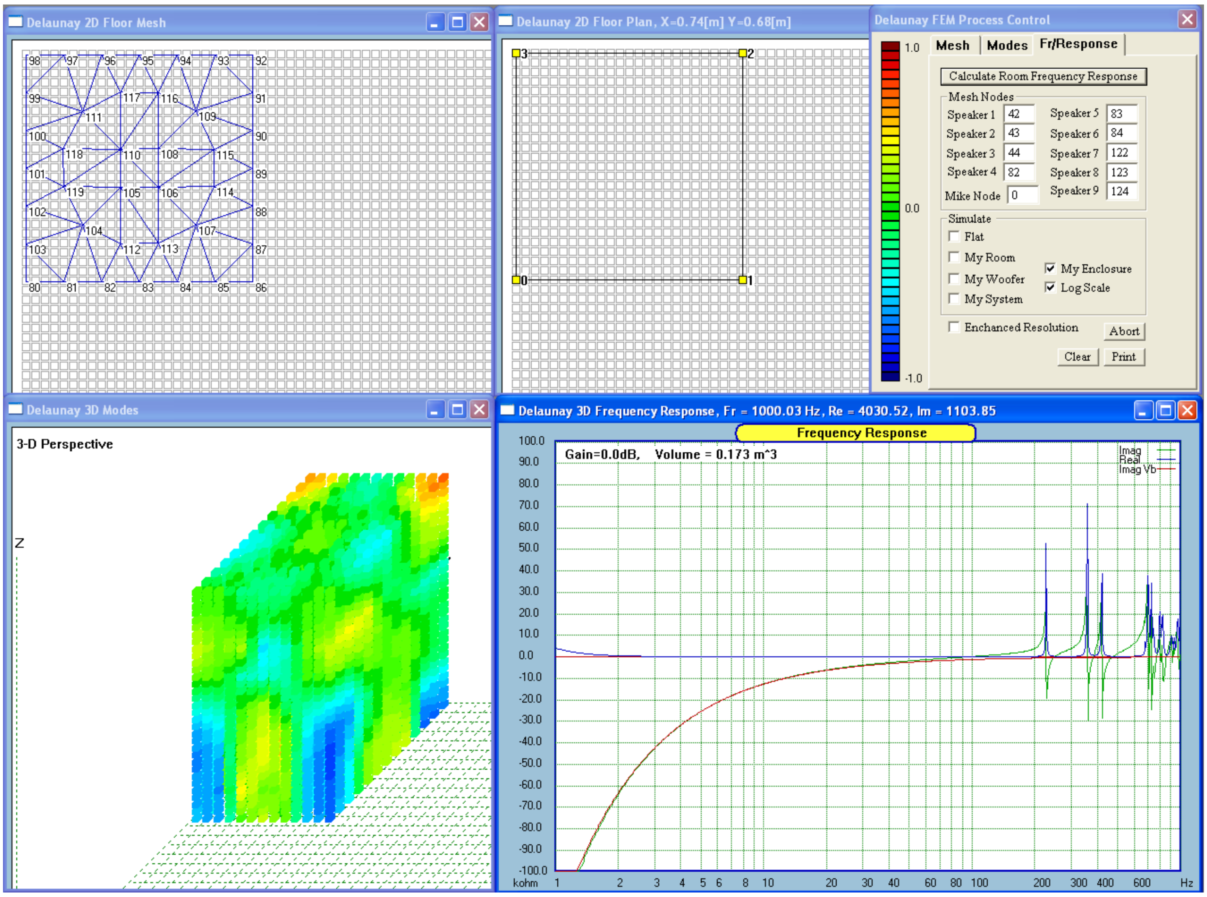


Fig 4.86 Driving point impedance of a simple rectangular enclosure – LOG scale.