Active Equalization of Loudspeakers

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Active equalization of a loudspeaker system is one of the simplest and yet most effective ways of extending low-frequency response of the system. If designed properly, it can provide good bass from small to moderate enclosure. However, since the extra low frequency output comes at the expense of the electrical power delivered to the system, it is vital to understand the design trade-offs and avoid pitfalls which may lead to mechanical damage to the driver.

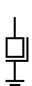
This article attempts to highlight some of the critical, but rarely discussed, issues associated with active equalization. First, the driver and enclosure are re-visited for acoustical impedance analogy representation. Next, an example of a simple active equalizer is given and finally, passive and active transfer functions of the combined system are reviewed.

Background

Transfer functions for several popular types of enclosures can be easily derived and computer programs can readily predict system response based on Small/Thiele parameters. However, this approach limits the user to a fixed set of functions built into the program. For example, if the program was designed to model sealed and vented enclosures, it will not model band-pass designs and so on. The approach presented in this article differs from the one described above. Currently available computer simulation software enables the user to utilize CAD methods and create acoustical impedance circuits containing all components of a physical enclosure and driver combination. Models created this way may be more complete than the simplified one, used for deriving fixed formulas, and offer fewer limitations to what can be modeled using CAD methods to obtain system responses. For the driver and the enclosure, a two-step process is proposed: (1) modeling of the physical enclosure using a mechanical mobility model and (2) transformation of the mechanical mobility model to acoustic impedance model, which is used as CAD circuit for obtaining all transfer system functions.

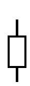
Mechanical Mobility

In this type of analogy velocity, 'u', corresponds to voltage and pressure, 'p', corresponds to current.



1. Mass Mm (kg): , second terminal of mass is the earth and has always zero velocity.

2. Compliance Cm (m/N): , compliant elements usually have two apparent terminals, which move with different velocities.



3. Resistance Rm (mks mechanical ohm): , resistive elements usually have two apparent terminals, which move with different velocities.

Mechanical mobility representation was chosen as a starting point because it is easier to identify individual mechanical components of the driver and enclosure. The following observations are useful:

1. all the symbols have two terminals, having velocities ua and ub.
2. symbol of the mass has one terminal connected to zero velocity, ub=0.
3. terminals vibrate in vertical direction.
4. horizontal, massless, rigid and losses line connects terminals with the same velocities.

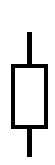
When faced with the problem of drawing an equivalent electrical circuit for a mechanical system, it is helpful to determine whether the velocity across different elements is the same or not. This could help to determine if the elements should be in series or parallel. In the loudspeaker's mechanical mobility circuit, velocity is represented by potential; so mass, compliance and resistance of the vibrating assembly must be in parallel, as they move with the same velocity. Also, it is beneficial to consider both pressure and volume velocity for each element. For example, enclosure and vent are subjected to the same pressure, so in the mobility circuit they will appear in series, as the pressure (current) flows through them. Once the mechanical mobility (sometimes called "inverse") representation is completed, the diagram is transformed to acoustical impedance, or "direct" circuit, using duality or "dot" method.

Acoustical Impedance

In this type of analogy pressure, 'p', (newtons per square meter) corresponds to voltage and volume velocity, 'U', (cubic meters per second) corresponds to current.

1. Mass Ma (kg/m): , mass of air accelerated by a force, but not compressed. A typical example is a tube with cross-sectional area S, filled with gas (vent). Ma=Mm/(S*S).

2. Compliance Ca (m/N): , compliance is associated with a volume of air compressed by a force, but not accelerated. Ca=Cm/(S*S) One terminal must always be at 'ground potential'.



3. Resistance Ra (mks acoustic ohm): , resistive elements usually have two apparent terminals. Elements are associated with dissapative losses of gas movement. Ra=Rm/(S*S).

Transfer functions

Sound pressure, P, at a distance, r, is proportional to frequency, f, and is governed by the following formula:

P=(f*1.18*|U|)/(2*r)

Quantity, U, represents volume velocity, r is the distance from the source in meters, f denotes frequency and 1.18kg/m3 is density of air.

Cone excursion X is described by the following formula:

$X=1/(s*Cas*Zat)$, $s=j*2*PI*f$, $PI=3.142$

Zat is the driving point impedance of the acoustical impedance representation and s is the complex angular frequency. Cas is the acoustic compliance of the driver.

It can be observed from the above that CAD systems involved must be able to handle not only solution of the circuit for nodal voltages and branch currents, but it also must be capable of accepting the above formulas for plotting final axial pressure and cone excursion curves.

The Driver

An arbitrary, but typical, 12 inch driver will be used for the model. Figure 1 shows all relevant data of the driver.

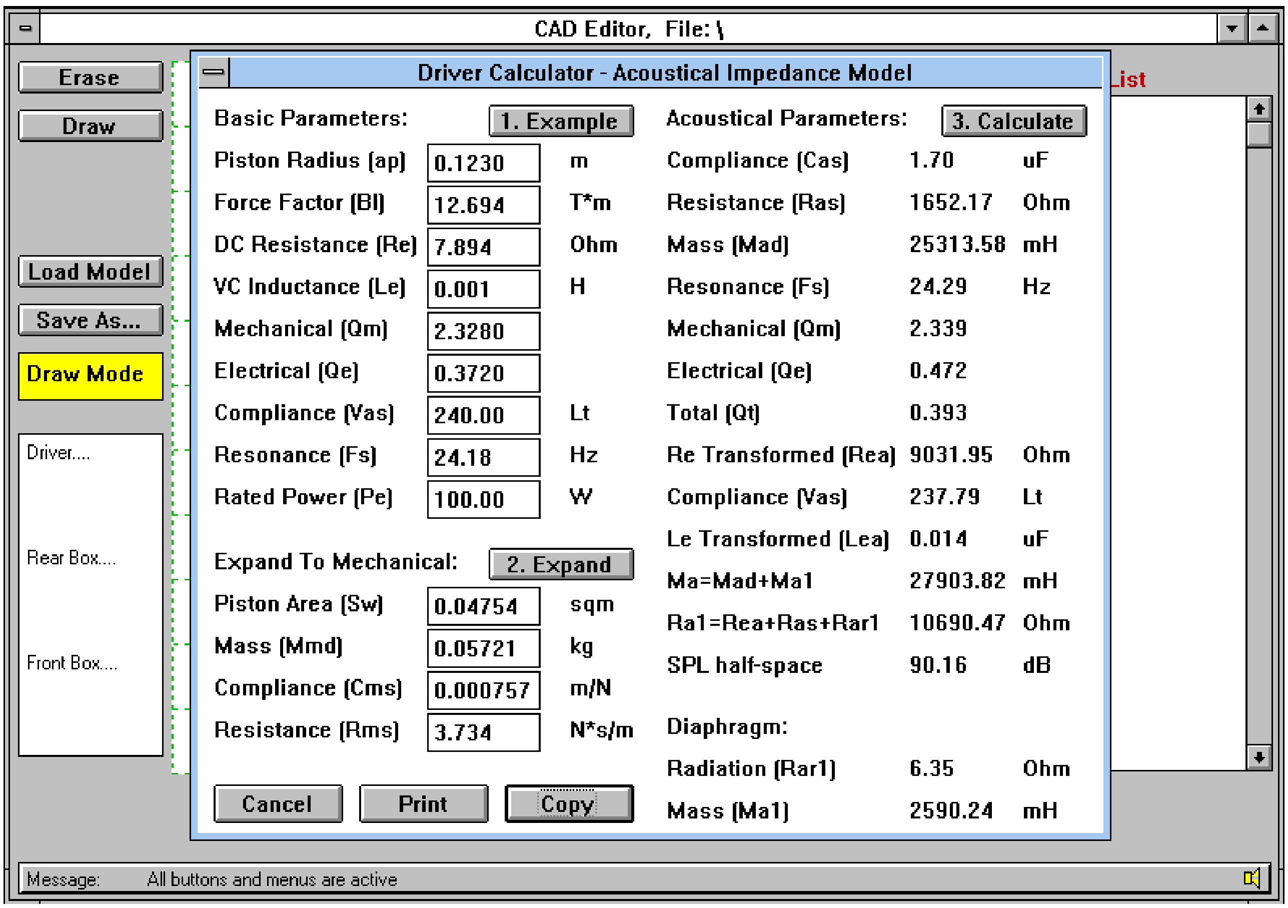


Figure 1. Small signal parameters of my test driver

Small/Thiele parameters are converted first to electromechanical, and then to acoustical impedance, parameters.

Equalisation example will be focused on 50W system with 7dB boost at low frequency. This would necessitate 250W peak power amplifier - 250W is 7dB more than 50W. It will also be assumed that 50W has been applied for some time to the driver, so that temperatures of the voice coil and magnet have stabilized and DC resistance, Re, (initially 6.5ohm) and force factor, BL, (initially 13.0 T*m) are now modified by their elevated temperature. Re will increase due to the positive thermal coefficient of aluminum and BL will drop due to a small loss of flux at elevated temperatures. Figure 1 shows actual new values for Re and BL that have been substituted for the original Re and BL and final acoustical impedance parameters were recalculated.

Figure 2 shows power compression parameters relevant for this driver.

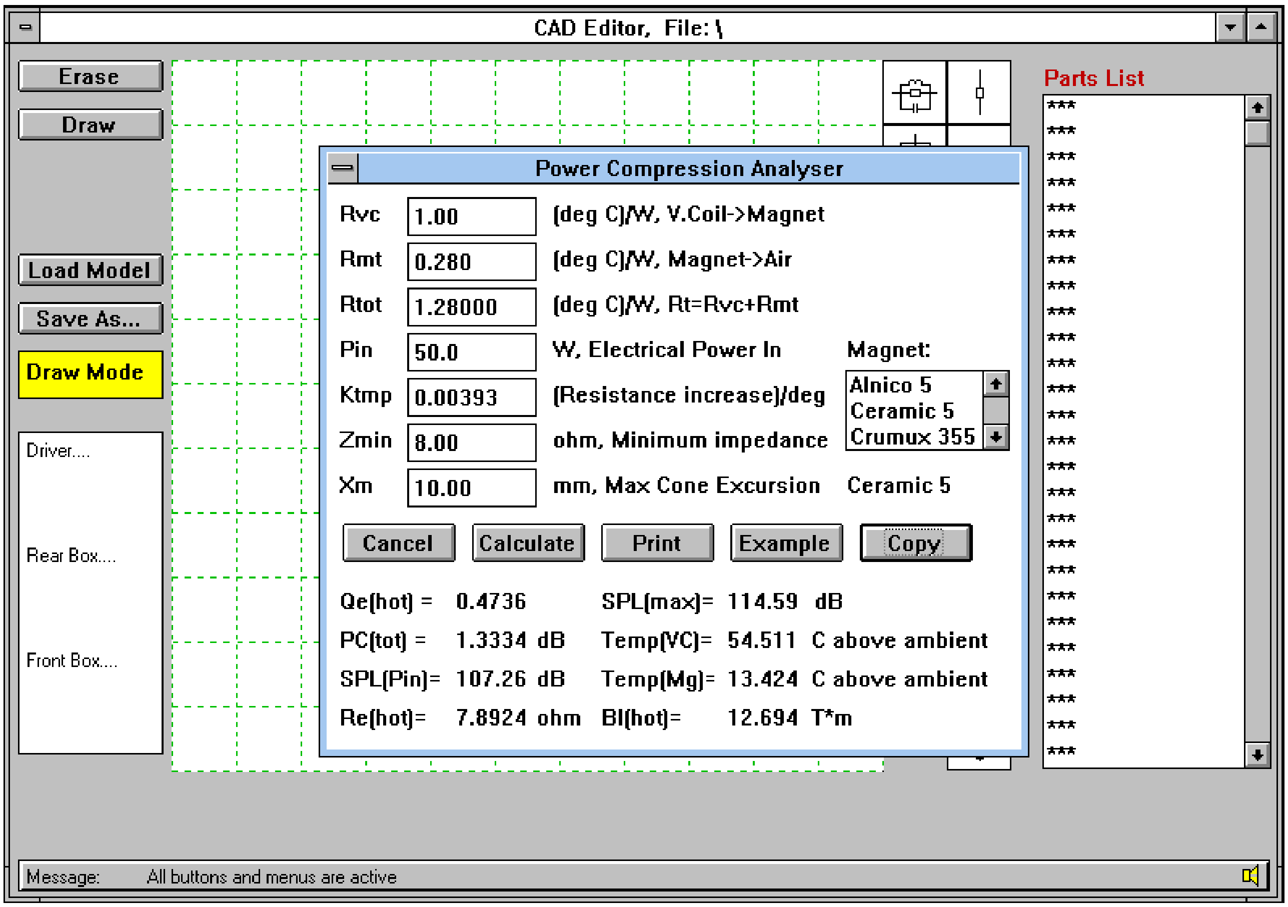


Figure 2. Power compression parameters relevant for this driver.

The final acoustical impedance model of the driver operating at 50W level is shown in Figure 4. Next we will look at the enclosure.

The Enclosure

Within the piston operating range, a loudspeaker vibrating assembly consists of mass, Mmd, resistance, Rms, and compliance of the suspension, Cms - Figure 3a. These three components move with the same velocity, Uc, in respect to the box, which has velocity equal to 0. The first step is to draw Mmd, Rms and Cms connected in parallel between the velocity line U=Uc and U=0.

The front of the diaphragm radiates into the air in front of the box and the radiation impedance is represented by Mmr and Rmr connected again in parallel between U=Uc and U=0 lines. In this case, connection to Uc reflects the fact that air immediately outside the box is excited by the diaphragm and connection to U=0 represents the fact that the energy will eventually be sinked into the "still air".

The back side of the diaphragm is loaded by a thin, non-compliant layer of air represented by Mmb and then by compliance of the box with its losses - Cmb and Rmb connected in parallel between velocity lines. Upper end of the mass, Mmb, vibrates with velocity, Uc, and the other end, as we remember, must be connected to ground when mechanical mobility circuit is considered. The air in the box is compressed between the diaphragm, vibrating with velocity, Uc, and the walls of the box, where velocity U=0 and this explains connection of Cmb and Rmb. 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.

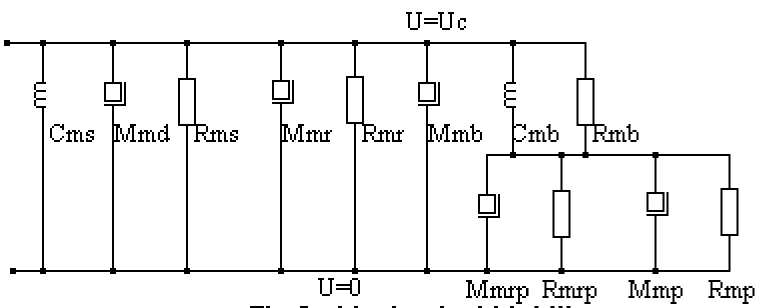


Fig 3a Mechanical Mobility

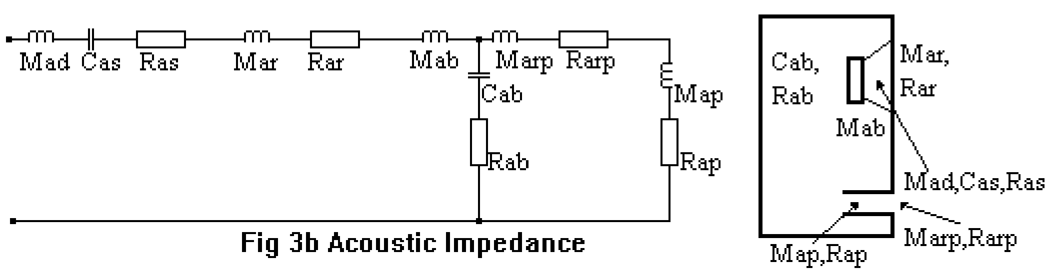


Fig 3b Acoustic Impedance

The air compressed in the box by the back side of the diaphragm has only one path to escape - pushing the air mass through the vent. Therefore, the pressure path consists of series connection of Cmb, representing compliant air in the box and the 4 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 ends of the masses are connected to the U=0, or reference velocity, as required in mechanical mobility circuits.

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. Figure 3b shows mechanical mobility circuit from Figure 3a transformed to acoustical impedance representation. Masses Mar and Mab constitute what is known as "air load" Mat=Mar+Mas. The air load increases total moving mass by up to 10% above the weight of the vibrating assembly. Assuming that the walls of the enclosure are rigid, one would expect that absorption losses are small and the dominant enclosure losses are due to air leaks. There are many possible paths for the air to escape, such as: back plate connectors, basket seal and - probably the worst offenders - porous dust caps. These losses will affect enclosure Q-factor. It is more convenient to lump all possible losses due to various leaks into one element, Rl, and represent it on the acoustic impedance circuit as a resistor connected in parallel to Rab+Cab. This would provide an alternative, parasitic path for volume velocity (current - in acoustic impedance circuits) to trickle away from the circuit to the ground. Low Rl resistance would provide lower resistance path for volume velocity to be diverted from the radiating component (port), effectively reducing total system output. The acoustical impedance circuit is therefore more refined than the mechanical mobility representation and offers the user separate entries for absorption and leakage losses.

Acoustical impedance representation shows Cab 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. The acoustical impedance model created using the "dot method" can be again enhanced by adding enclosure leakage losses represented by Rl and connected in parallel to Rab+Cab. Figure 4 shows acoustical impedance representation adopted for the vented enclosure model. The components are:

R0 = Rea; electrical DC resistance, Re, transformed to acoustical side.
C1 = Lea; voice coil inductance, Le, transformed to acoustical side.
C2 = Cas; equivalent compliance volume, Vas, transformed to acoustical side.
L3 = Mad; mass of the vibrating system, Mms, transformed to acoustical side.
R4 = Ras; vibrating assembly loss, Rms, transformed to acoustical side.
L5 = Mar+Mab; air radiation of the front side of the diaphragm + air load of the back side of the diaphragm.
R6 = Rar; air radiation of the front side of the diaphragm.
C7 = Cab; enclosure compliance, Vab, transformed to acoustical side.
R8 = Rab; absorption losses of the enclosure transformed to acoustical side.
R9 = Rl; leakage losses.
L10 = Marp; port radiation.
R11 = Rarp; port radiation.
L12 = Map; mass of the air in the port.
R13 = Rap; frictional losses in the port.

The enclosure used in this model is grossly de-tuned. Optimum tuning frequency would be around 30Hz, but the implementation of this model calls for extended bass response and active equalization. It can be observed that cone excursion curve has a pronounced dip at the tuning frequency of 20Hz. The curve is plotted in linear-log scale to relate cone excursion to the maximum linear displacement range of 10mm (see Figure 1). The amplitude response is down by 6.5-7dB at 20Hz and this is the ideal frequency to peak the active compensation network, discussed next.

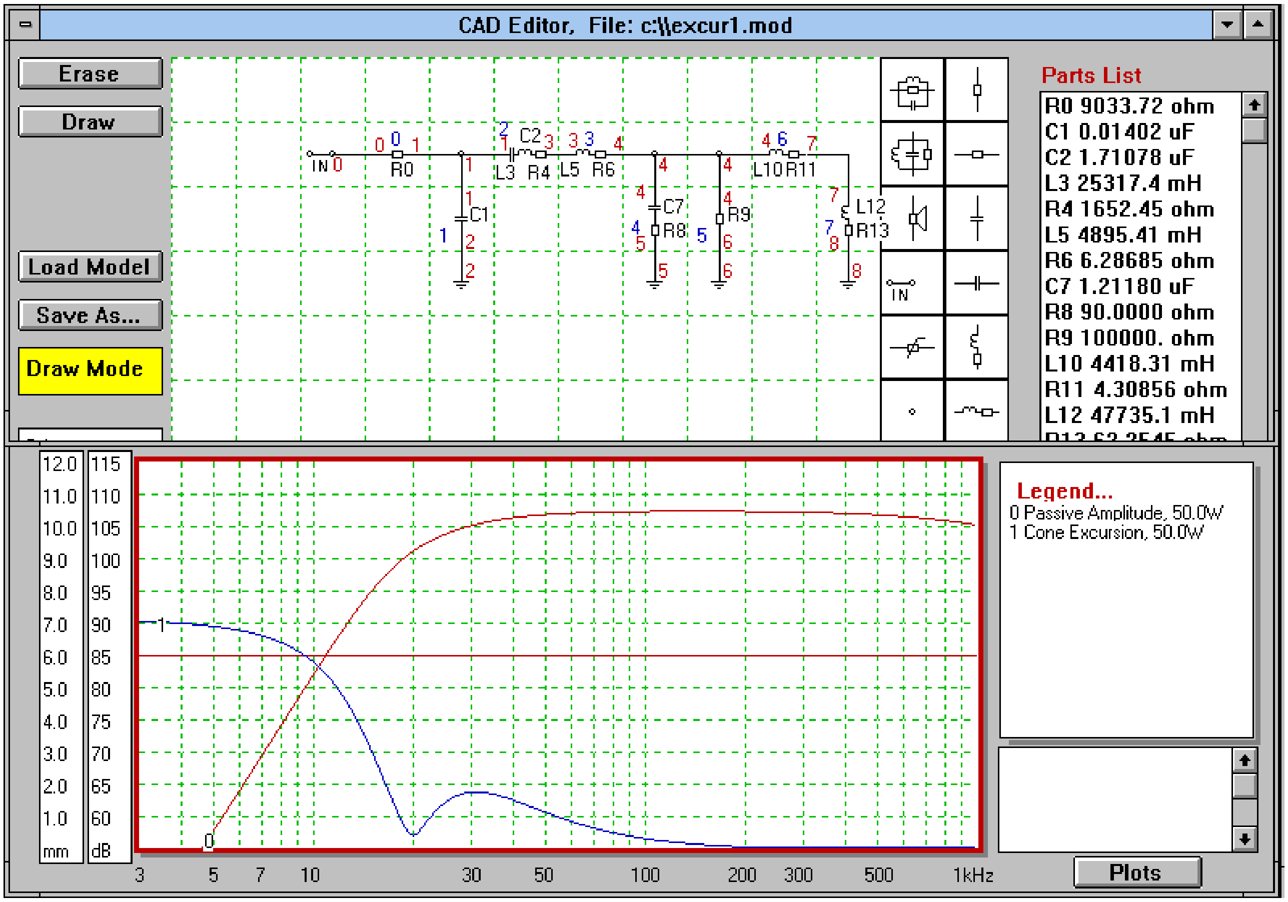


Figure 4. SPL and cone excursion

Active Section

In a generalized case of tunable enclosures, it can be observed that the system's frequency response is already down by 3-4dB at the enclosure tuning frequency. Without equalization, this would be the specified system -3dB cut-off frequency. Also, cone excursion curve superimposed on the amplitude response plot reveals that cone excursion has a dip at the enclosure tuning frequency. One can take advantage of this and feed more power into the driver while still avoiding mechanical failure and any significant increase in distortion. Obviously, the driver must be able to handle increased electrical power. One can take this concept even further and deliberately tune the enclosure to lower frequency than that which the optimum flat response would require. Any irregularities introduced in the frequency response this way can be electronically equalized with the added advantage that the cut-off 3dB frequency would be still lower. It is this particular advantage of tuned enclosures that makes them particularly suitable for active equalization.

Here, one must assure the peak of the equalization matches the enclosure tuning frequency, where the dip of the cone excursion resides. The second factor taken into account is a need for "mirror image" curves for active boost and passive driver frequency roll-off. The third factor is matching power handling of the driver with electrical peak power required. Securing the above three conditions protects the driver and provides optimum bass extension.The active equalizer used in the model is a second order high-pass filter (Sallen and Key type). Rather than selecting a more complex and sophisticated equalizer, this (possibly the simplest) active equalizer has been chosen deliberately to illustrate the main issue at hand - avoiding pitfalls of active equalization. Filter values and schematic are shown on Figure 5. The filter has the following main characteristics:

1. The amount of boost is around 7dB at 20Hz.
2. For 50W active subwoofer, the system would require 250W peak electrical power.
3. 12dB/oct high-pass filter protects the system against excessive infrasonic frequencies.
4. Amplifier voltage gain is equal to 1.
5. The filter can be built using any popular, low noise operational amplifier and can be inserted in the audio path just prior the main power amplifier.

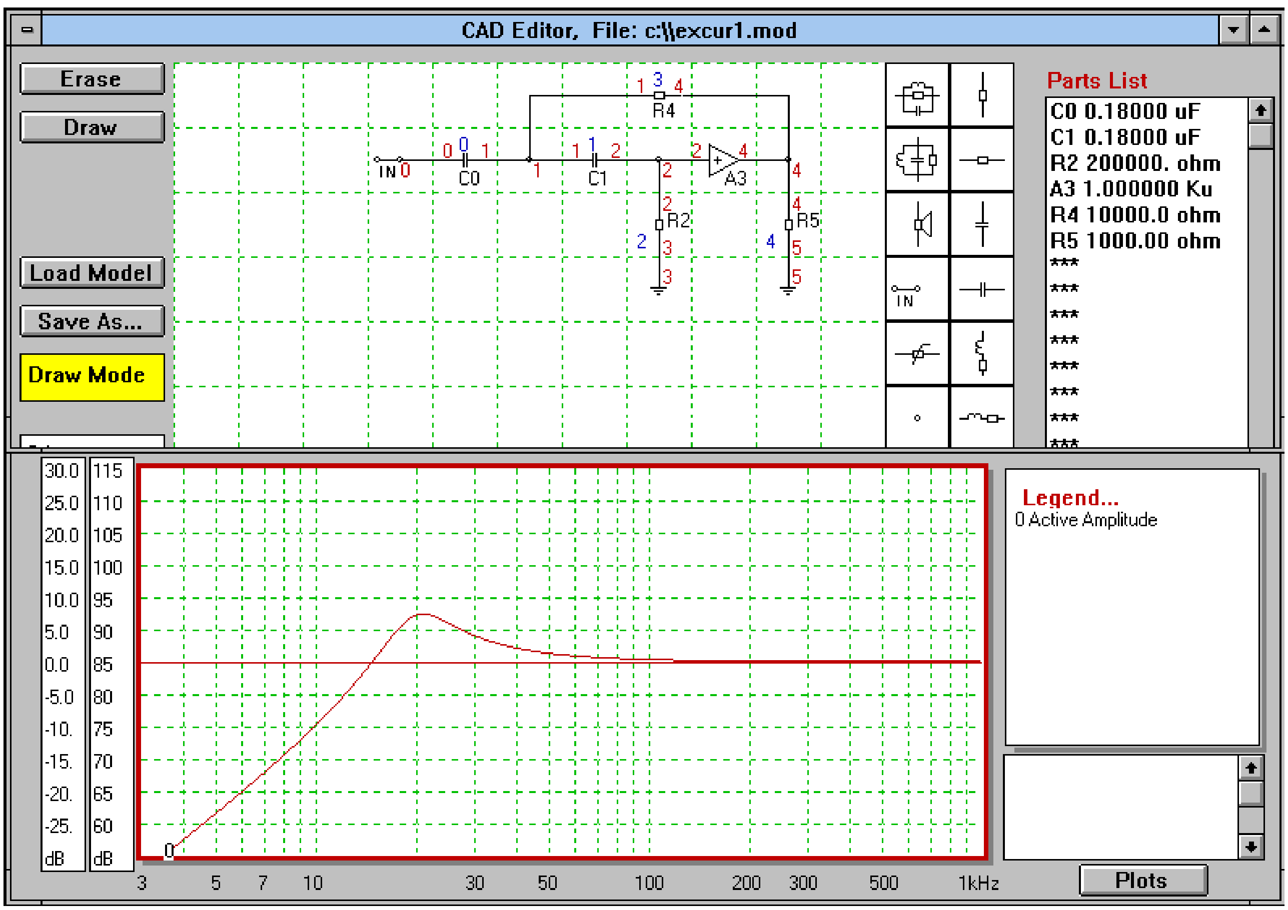


Figure 5. Active section filter

Combined System

Finally, the all-important system responses are evaluated at 50W input power fed into the loudspeaker. As previously explained, plotted curves represent system performance after the temperature of all components has stabilised. Figure 6 shows the relevant transfer functions: (1) frequency response of the passive system at 50W - curve 0, (2) cone excursion of the passive system for 50W input power - curve 1, (3) frequency response of the combined (passive and active) system - curve 2 and finally (4) cone excursion of the combined system at 50W - curve 3.

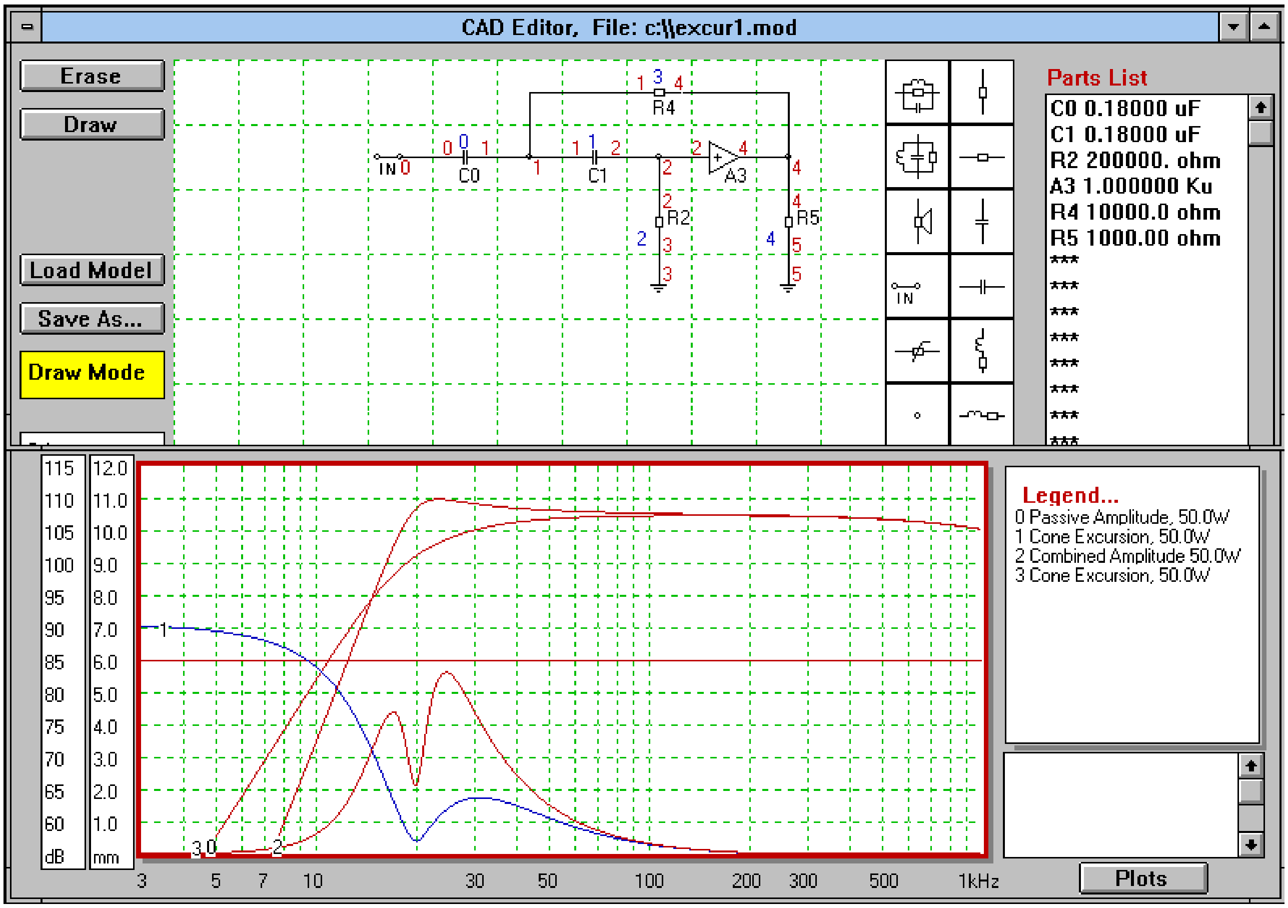


Figure 6. Combined response

It is immediately observable that even at the equalization peak, the cone excursion is well within the safe operating range, at around 56% of the maximum linear displacement (curve 3). The -3dB low-end cut-off point of the combined system moved down to 18Hz, which is better than was expected (curve 2). Usage of the high-pass filter has visibly dramatic effect on the cone excursion at very low frequencies, providing excellent protection for the driver. Generally, this system would pass as a very good subwoofer. Adding a second driver in "isobarik" configuration would reduce box size to 85 liters and would double electrical power handling. With appropriate equalization, one could easily design a 100W active subwoofer system with 15Hz low-end -3dB frequency. Systems exhibiting this level of performance are commercially available.

As far as cone excursion is concerned it would be possible to operate the model system at even higher power (75W system or 375W peak). It may be useful to note that several well known manufacturers offer home Hi-Fi class loudspeaker drivers capable of 1000W and more electrical impulse power. At this stage, careful analysis of driver's electrical power ratings and thermal analysis are recommended. If thermal resistances of voice coil-to-magnet and magnet-to-air are known, it will be possible to determine power compression factors for any level of continuous input power (see Figure 2). For comparison, Fig 7 shows the same system with the vent blocked. This would convert vented enclosure to a sealed box. The plots are: (1) frequency response of the passive system at 50W - curve 0, (2) cone excursion of the passive system for 50W input power - curve 1, (3) frequency response of the combined (passive and active) system - curve 2 and finally (4) cone excursion of the combined system at 50W - curve 3. Although the cone excursion of the passive system is significantly lower below 20Hz, the active equalization causes cone excursion to exceed the 10mm limit at 20Hz by up to 40% (the curve is clipped at the top).

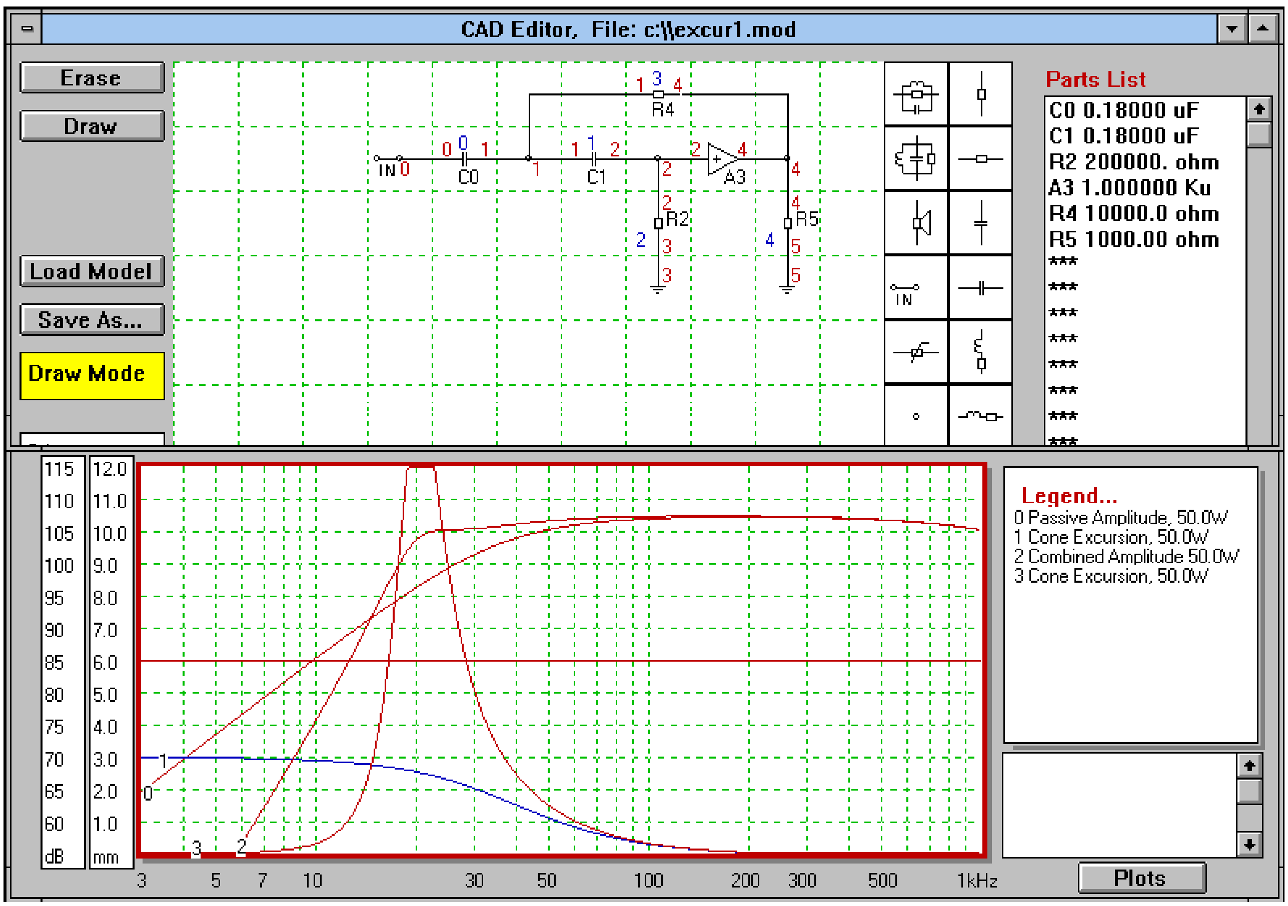


Figure 7. Combined response for sealed box.

Also, despite the 7dB of equalization, the amplitude response of the sealed system is 5.5dB below vented system at 20Hz and began to roll-off at 100Hz. Another 2.5dB of active boost would be required, which would make the cone excursion problem even worse. It needs to be noted that sealed enclosure of the same size as vented enclosure is not suitable for this driver. However, it does illustrate cone excursion problems when a sealed boxes is actively equalized. Analysis of the two cases would indicate that tunable enclosure would be the preferred choice when active equalization is considered.

The combined vented system would provide excellent low frequency extension, and, if used with properly designed high-pass filter, mechanical integrity of the driver can be assured. Combined vented system is sensitive to misalignment between the following parameters: (1) enclosure tuning, (2) amount of low frequency boost required, (3) frequency of maximum boost. Mismatch in the above conditions seems to be the most commonly encountered difficulty when attempting active equalization.

Another View of Tuning

In the context of active equalization, tuning of the vented enclosure could be viewed from a different angle. Traditionally, box tuning frequency, Fb, would be chosen with the optimally flat frequency response of the passive system in mind. With the help of an active network, box tuning frequency can be selected as the lowest frequency for which the "detuned" vented enclosure can still be equalized, taking into account all power and cone displacement related issues.Evaluation of several transfer functions, cone excursion being one of the most important, is needed for successful design outcome. Overlapping all relevant curves greatly improves understanding of correlation between all vital parameters and trade-offs being made. The accuracy of the analysis was further enhanced by including thermal/power compression effects. Therefore analytical software tools, such as the one used for this model, are highly recommended.

References

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