Designing Loudspeaker Boxes

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Loudspeaker designers have traditionally calculated enclosures with the assistance of Thiele/Small (T/S) transducer model parameters. This is a technique that was under continuous development until approximately 1973; the methods have not significantly changed since then.

One may think this lack of further development is because the technique has been perfected, but this chapter will demonstrate the limitations and present a simple improvement to this technique demonstrated through a simulation. A method of evaluating the sonic quality of bass alignments is presented to foster further debate.

Historical overview

Most historical descriptions of loudspeakers (as we know them today) begin with Chester W. Rice and Edward W. Kellogg from General Electric [1]. They described what we know today as direct radiant transducers (included in a patent from 1924) and showed that speaker units in the mass-controlled range above their fundamental resonance can have a useful working range. This chapter is essentially describing the design of speaker boxes based on this concept.

One of the earlier books on speakers and acoustics is *Elements of Acoustical Engineering* by Harry F. Olson, published in 1940 (1st edition; 2nd edition in 1947) [2]. This book examines several interesting loudspeaker aspects, including the electrical equivalent circuit, the differential equation of motion, equations for nonlinear suspension, distortion and transient characteristics, cone vibrations and more. However, the book does not go into detail on how to proceed in designing an appropriately sized enclosure. A later expanded edition titled *Acoustical Engineering* was published in 1957.

Leo L. Beranek, a professor at the Massachusetts Institute of Technology (MIT), published *Acoustics* [3] in 1954. The book describes electrical equivalent circuits in detail and discusses open baffle, closed-box baffle, bass-reflex and other enclosures, drafting a methodology to reach a desired result. For example, bass-reflex enclosures are to be designed using an 8-step procedure that is repeated if the resulting performance is not satisfactory.

Furthermore, Beranek outlined when the bass-reflex tube behaves as a port (as intended) as well as when it instead behaves as a transmission line. This transition has been closely studied in recent times, and *Acoustics* is recommended reading material for any loudspeaker enthusiast.

In the 1959 *Journal of the Audio Engineering Society* (JAES), James F. Novak [4] described both the closed box and the bass-reflex box using an equivalent circuit. Novak derived *Q*-values and provided guidelines for designing enclosures. He claimed that the distortion in a bass-reflex box is less than that of a closed box. Novak asserted that Q_{TS} = 0.32 can provide a flat response; indeed, when applying his method to determine the enclosure's influence relative to the unit's stiffness, one does achieve "more bass" (lower resonance frequency). The theory is supported by examples, but there is no mention of any connection to filter theory or alignments.

Albert Neville Thiele committed a stroke of genius when he coordinated the speaker model parameters with the enclosure model to form a total system, tying it all together with well-known filter theory. By using both the transducer and enclosure parameters, one can predict the system's frequency and impulse responses. This method was presented at the 1961 AWA Convention and then published in the Proceedings of the IRE Australia [5] in the same year. Thiele mentioned that he chose this media to give Australians "a leading edge." There was little interest, however, and the concept was headed into oblivion until Richard H. Small of California arrived in Sydney to pursue his PhD thesis and expressed an interest in this work.

In January 1966, an article written by James F. Novak titled *Designing A Ducted-Port Bass-Reflex Enclosure* was published in *Electronics World* [6]. He explained a step-by-step procedure for what he called the "optimum volume design" to achieve the best transient performance and maximum efficiency. The procedure asserted that the port must always be tuned to the free air resonance of the transducer. Novak explained that the enclosure can be either too large or too small. The concept of "alignments" was not mentioned in the article; Neville Thiele's work was not common knowledge at this time.

From 1968 to 1972, Jeremy E. Benson wrote thorough and relatively complete articles on speaker systems that were published in the AWA Technical Reviews [7, 8]. These articles almost constitute an entire book on the subject at a high level; in fact, they were later published as a book titled *Theory & Design of Loudspeaker Enclosures*. These papers are exhaustive and pedantic, so the reader can easily get lost in less than critical details. They are useful for experts but of little use for the common hobbyist deciding how to size and build speaker enclosures.

Richard H. Small went on to publish his findings in the JAES from 1969 to 1974. In particular, his 1973 articles on bass-reflex boxes [9, 10, 11] proved to be crucial documents; interest in the method blossomed and it became a huge success. Today, approximately 40 years later, the method remains strong and is still taught to engineering students at universities around the world. Hobbyists also use the method as a basis for their practical work.

Compared to Thiele's articles, Small addressed the loss factors (which are also discussed in Benson's articles) and offered the practical simulation assessments of what's simulating correctly. This led to the frequently used approximation that leakage loss should initially be set to $Q_L = 7$, while other loss factors should be ignored (Beranek, Section 8.18, also showed calculations of box Q-values in the range of 7).

It thus became possible to simulate different enclosures for a given speaker unit and specify model parameters for a speaker unit based on the enclosure size and the desired total response. The predictions are quite accurate under "normal" conditions (within 1 dB or so) and unprecedented for the time (Small made calculations with pen and paper, as computers were not publicly available to the general public until the 1980s). Small's work advanced the method from theory to practical application.

The esteemed Knud Thorborg worked for Peerless (Denmark) when Thiele's paper was published. Peerless subscribed to the Proceedings of the IRE Australia and the engineers were quite interested. However, initial studies showed that the theory did not fit with experiments and the theory was rejected. It was later concluded that the reason was due to leakage losses in the transducers (probably from dust caps that were not air tight, which was not an unusual design at that time).

Observations on transducers and boxes

The traditional simulation model includes issues that anyone can observe when looking at things in detail. The measurement and determination of model parameters is most frequently accomplished with the added-mass approach, where a well-known change in the mechanical system moves the resonance frequency downwards; this is used to determine both the unit's moving mass and the force factor of the motor system (which converts between the mechanical and the electrical domain). All measurements are made in the electrical domain, and the change of mass in the mechanical domain reveals the force factor.

An appropriate enclosure is then calculated and the box is built. The unit is placed in the box, the resonance frequency of the system moves upward and the desired system response is expected to be achieved.

Let us plot the results of these efforts and compare the reality with simulations made using the Thiele/Small model. In this context, we only examine the impedance magnitude. The example is a Scan-Speak Revelator 18W/4531G00 in free air, with 20 grams of added-mass (free air) and with no added mass but mounted in a 15-liter sealed enclosure. The box is in this particular case built in aluminium, which is welded together, which provides a nearly theoretically ideal enclosure that possesses neither leakage loss nor internal damping (absorption). The simulations are therefore made without the inclusion of loss factors. See Fig. 1.

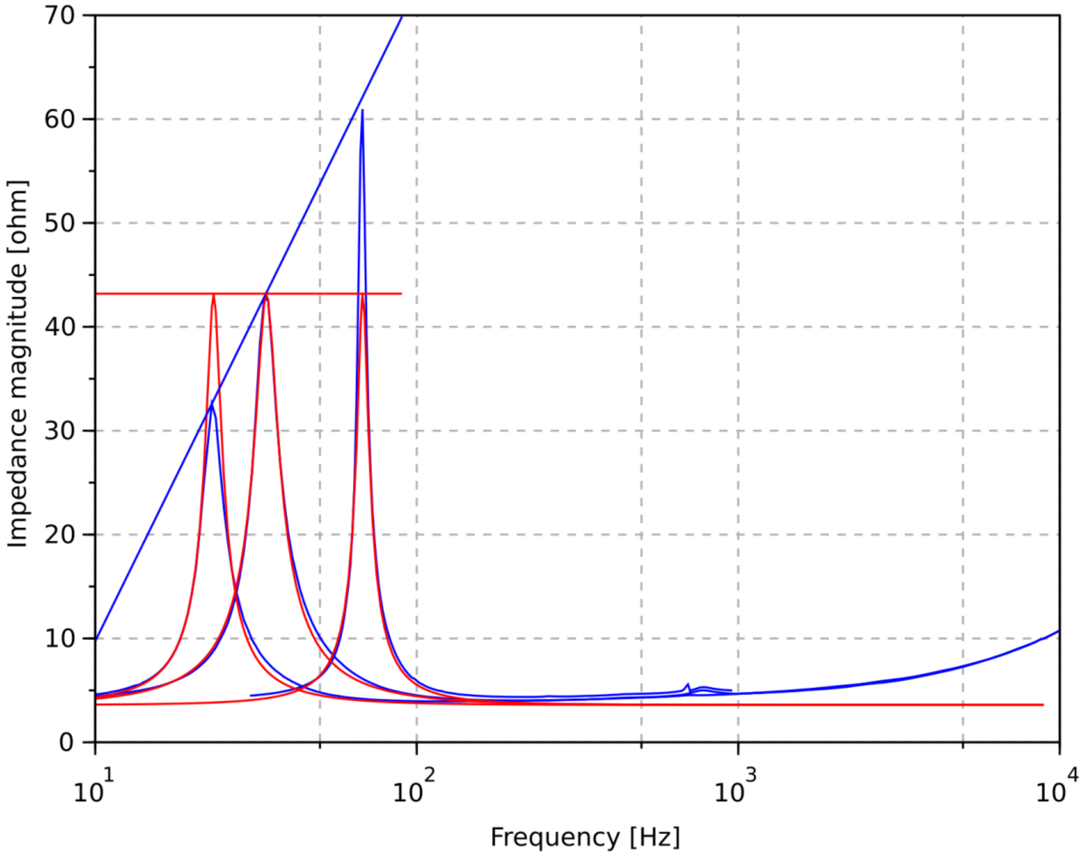

Fig. 1: Blue curve = measurements. Red curve = simulation.

Fig. 1 shows that the simulations do not match the experimental measurements. We choose to let the simulation and measurement fit for the device in free air, here we determine the T/S parameters, while both the added-mass and mounted box measurements disagree with the simulated measurements.

There are two approximated lines on the graph that respectively show (in red) that the simulation indicates an unchanged level of damping at all frequencies and (in blue) that in practice, the damping significantly changes with frequency. The damping rises at lower frequencies (the impedance peak drops) and reduces at higher frequencies. The simulation error increases with larger shifts in the system resonance frequency.

The difference between the simulation with the Thiele/Small model and experimental measurements suggests that the model is an oversimplification of reality. Because the differences are observed in free air (with and without added-mass), the responsibility of the divergent observations must lie with the transducer. Furthermore, it can be seen that the divergent observations are entirely unaffected when the unit is placed in the box, indicating that an improved transducer model will automatically lead to improved simulations of lossless enclosures.

A simple study of the above observations, as well as the traditional equivalent circuit shown in Fig. 2, indicates that the culprit is related to the damping in the mechanical domain. The component "R_{MS}" (or "R_{ES}" in the electrical equivalent circuit) is not constant, as has been assumed with the parameters in the Thiele/Small model; in practice, it changes with frequency. R_{MS} decreases (R_{ES} increase) with increasing frequency, which means that the damping is reduced at higher frequencies.

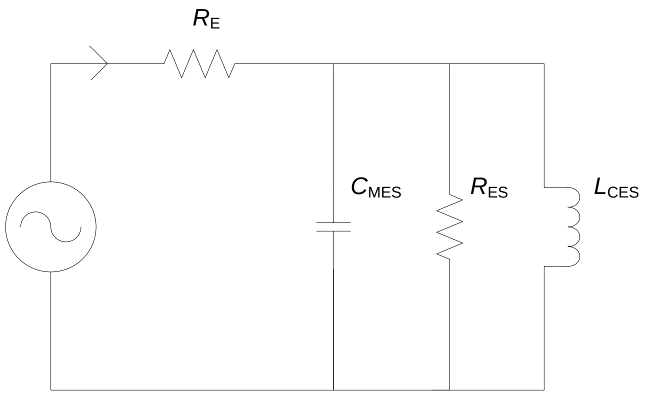

Fig. 2: Equivalent circuit diagram for the traditional model of the transducer. Induction is usually not included in the model and is therefore omitted. The coil is only represented electrically by a resistance value, R_E, which is typically the DC resistance.

Richard Small did observe in his 1973 AES article [9] that speaker system damping is not constant with frequency, but he did not indicate that this relates to the speaker unit. Small discovered the misconception and the correct context shortly after the article was published.

For Richard Small to have reached this conclusion, we can assume that his efforts to build a nearly lossless box showed that the impedance peak was higher than the traditional Thiele/Small model can predict. A *Q*-value greater than infinity is not possible (negative loss is nonsensical), and therefore Small arrived at his conclusion concerning frequency-dependent damping.

Today there are many options that include frequency-dependent losses in transducer modeling. A straightforward process includes mechanical viscoelasticity by replacing L_{CES} with a 3-parameter solid model, often called SLS (standard linear solid); this is described in many textbooks and scientific papers on the mechanics of materials [12, 13, 14], but better models do exist. The most successful model relating to speakers is the LOG model of M. H. Knudsen and J. G. Jensen [15], which was partially adopted by W. Klippel [16]. While the model is sound, it is mathematically oriented and not easily represented by the well-known electrical equivalent circuit (see Fig. 2). Further improvements have been suggested to the LOG model, e.g., by F. T. Agerkvist and T. Ritter [17], to improve the behavior at frequencies well above the driver resonant frequency.

Electrical equivalent circuits and the representation of transducer model parameters in the so-called "small signal" domain (where the nonlinear model parameters are assumed to be linear) is a technique that has been plagued by problems. The fact that this domain does not exist (as a practical matter) for normal speakers, means that focus has turned towards describing speaker units with nonlinear mathematics, which is valid for small and large signals with so-called "large signal" models. This has been addressed by D. L. Clark [18], E. S. Olsen [19], M. H. Knudsen [20], W. Klippel [21] and others, but little work has been done since 1973 to improve on the concept of small-signal models.

At the same time C. N. Strahm [22] established his own transducer model and developed software for calculating speaker systems. LinearX produces software called *LEAP Enclosure Shop*, now in its 5$^{\text{th}}$ generation; this software is used by several respected professionals and speaker manufacturers in the industry. The model utilizes a total of 53 model parameters to describe the transducer, but the software is proprietary and the model is not publicly available.

Transducer model with electrical induction and mechanical damping

This chapter proposes a simple extension of the conventional equivalent circuit diagram of the transducer to incorporate frequency-dependent damping. The model includes a total of five electrical components to accurately describe the transducer's electrical resistance and induction (see Thorborg et al.) [23, 24].

The model, though not perfect, is the simplest possible model that can describe the transducer in practice. The author strongly favors simple models; their direct purpose is to simplify reality yet still operate as intended. Moreover, models that reflect reality, particularly where components of the model reflect physical phenomena, are desired.

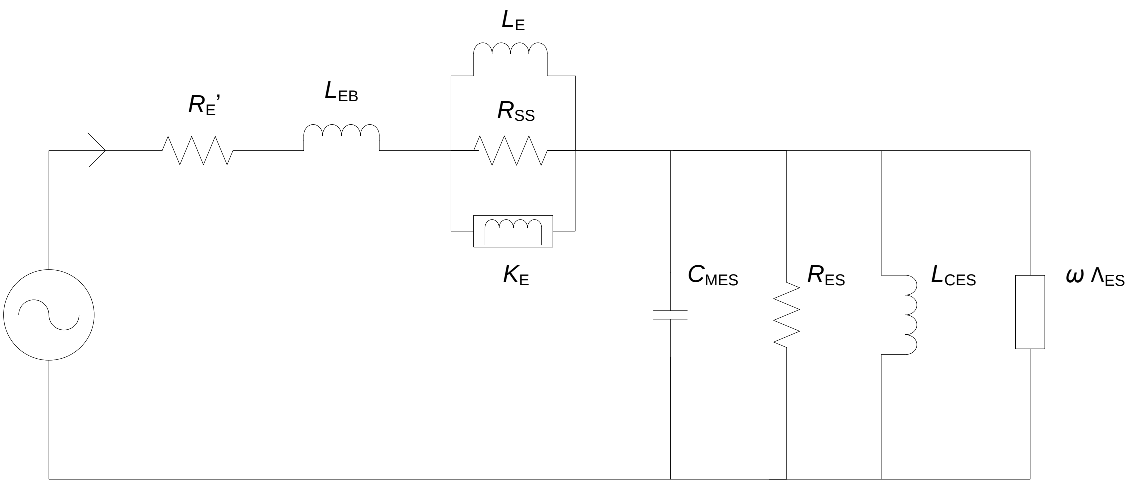

Fig. 3: Extended equivalent diagram, including frequency-dependent damping and induction. The FDD model, second generation [24].

Note that the FDD model, with its frequency-dependent damping, is not a viscoelastic (creep) model–it only includes the (linear) damping in the audio range. It is possible with additional mathematics for the FDD model to behave similarly to the LOG model, but this would be an unnecessary complication and not relevant for simple linear box simulations.

Compared to previous work by Thorborg et al. [23], it can be seen in Fig. 3 that the frequency-dependent damping resistor, $\omega \cdot \Lambda_{ES}$, is in parallel with the other mechanical components (C_{MES}, R_{ES} and L_{CES}). Earlier work placed the frequency-dependent damping resistor in series with L_{CES}; it makes sense to consider the frequency-dependent damping as a loss in the speaker's mechanical suspension, which is fully consistent with the LOG model of Knudsen and Jensen [15].

This proposal (Fig. 3) changes the layout to a purely parallel connection of the mechanical circuit components; while the change of the circuit is not particularly significant, it does make the values of Λ_{ES} and L_{CES} independent of each other, which is convenient when determining the transducer model parameters (the variables become orthogonal). The results of these efforts are plotted and the simulations performed with the FDD model are compared to the experimental measurements in Fig. 4.

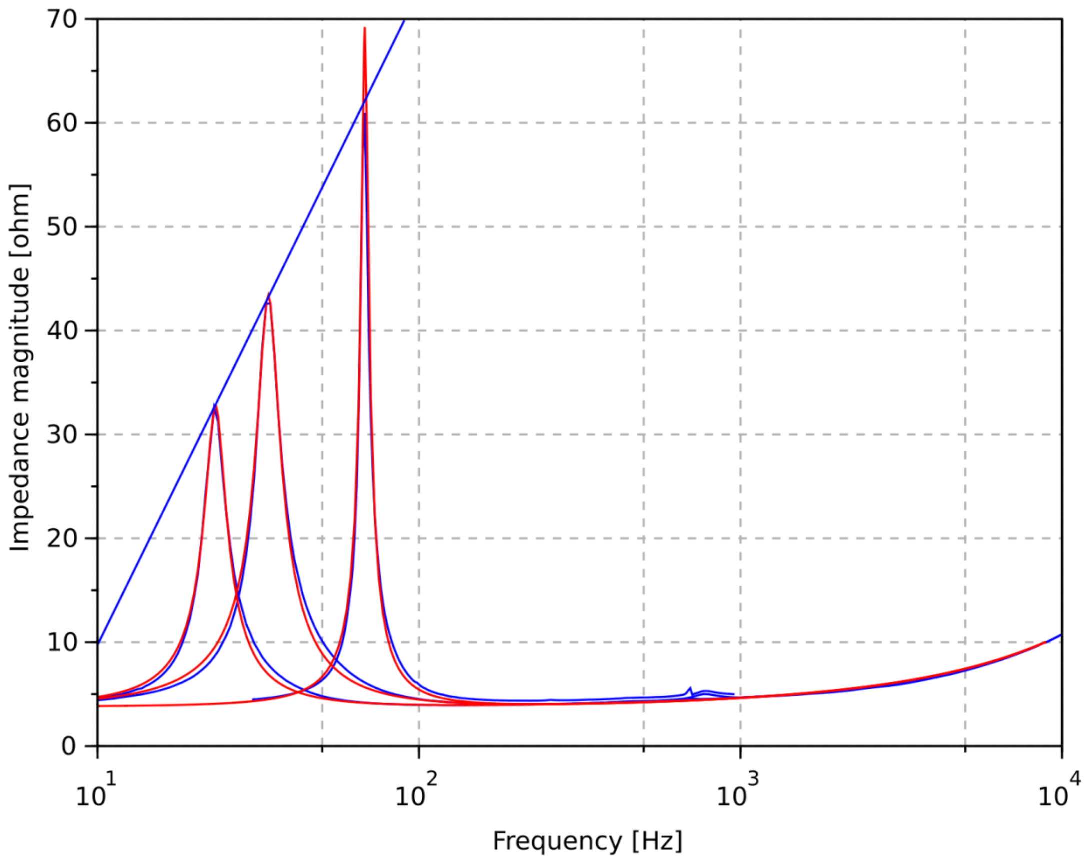

Fig. 4: Blue curve = measurements (identical to Fig. 1). Red curve = simulation.

As shown in Fig. 4, the free air simulations align almost perfectly with the actual measurements. The model has, despite its simplifications and small-signal modeling, made it possible to simulate the actual speaker in the given measurement setup (a stepped-sine response with a voltage generator). In the enclosure simulation, the red curve does slightly exceed the measurement; this is because in practice, there is a loss stemming from the mounting of the actual speaker unit. This minimal additional loss (in this case, $Q_B = 109$) is not incorporated into the simulation. We can conclude that, in practice, the simulations should be executed with $Q_B \approx 100$ for low-loss boxes.

Simple model, including damping and absorption

To address the design of enclosures, it is now necessary to study Small's AES paper from 1973 [9] in some detail. Through his work with the traditional loudspeaker model, Small arrived at the following recommendations:

Port loss	Q_P – between 50 and 100
Absorption loss	Q_A – typically 100 or more
Leakage loss	Q_L – between 5 and 20

Small writes in his paper (Quote): "*The last result is surprising, because the enclosure tested well built and appeared to be leak free*" and Small then arrived at the explanation that the damping is not independent of frequency.

In the previous section, we achieved an understanding that the frequency-dependent damping occurs in the transducer. Small observed a combination of frequency-dependent damping in the transducer with absorption losses in the box. Because the frequency-dependent damping in the transducer increased with decreasing frequency, and because the absorption losses in the enclosure rise with increasing frequency, the two phenomena counteracted with each other and Small essentially observed the effect of the combined losses crop up as leakage loss in his analysis of the total system. Small later confirmed to me in a private conversation that this is true.

When the model is extended to include frequency-dependent damping in the transducer, we can no longer use the old recommendations from Richard Small. We must instead use the correct leakage loss (which can be very small, resulting in a very high Q_L-value, 50-100 or higher). The absorption loss will now be dominant, perhaps 5-20 or higher. This is unknown because the loss comes into play with the frequency-dependent damping in the transducer.

In reality, the enclosure absorption is highly dependent on the damping material in the box. Many high-end speaker manufacturers choose to reduce the damping material to an absolute minimum because low damping enhances crisp and clear details. The purpose of damping material in the box is not to manipulate the low-end but to absorb internal reflections and prevent (acoustic) vibration-modes in the midrange frequency response. Low-loss is a common philosophy behind the products from various high-end manufacturers.

The solution is not to estimate a Q_A-value but to build a model where one can directly specify the amount of damping material, determine the absorption and study the system response.

We now present the model proposed by the author at the 130th AES Convention, held in London, May 2011 [25]. Fig. 5 shows a simple model operating in the "small-signal" domain (similar to the above-described transducer model), which ignores nonlinearities for ease of understanding. The model is based on a model of fibrous material proposed by W. Marshall Leach, 1989 [26] and the work of Gavin R. Putland, 1994-1998 [27, 28, 29] and Viggo Tarnow, 1996-2002 [30, 31].

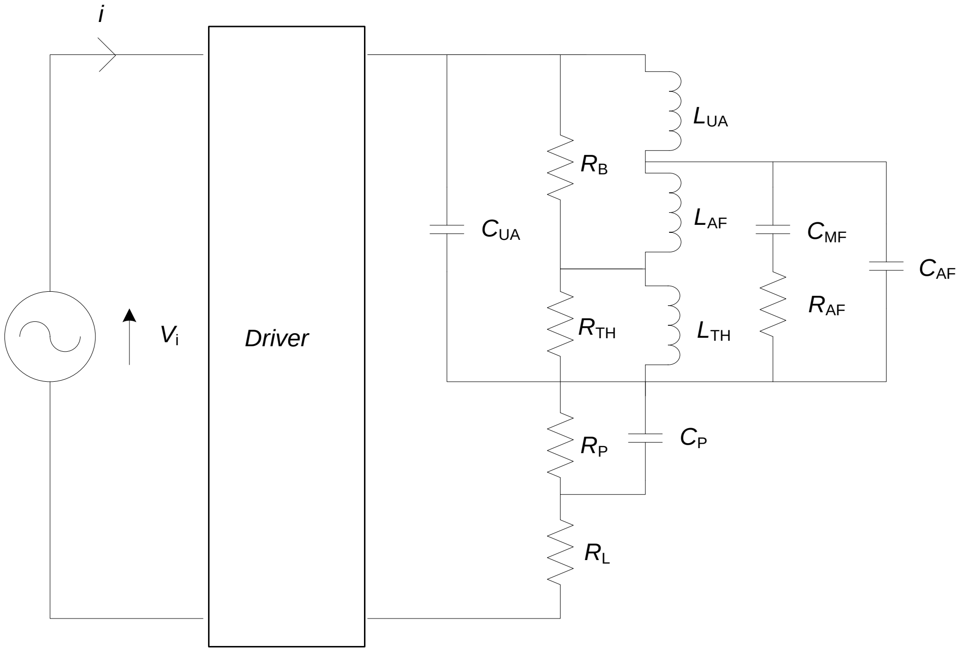

Fig. 5: Electrical equivalent circuit for the enclosure including damping material. The box titled "Driver" can contain, for example, the transducer model of Fig. 3. The above model contains C_P and R_P for the port mass and port loss of a reflex box–these are to be removed to simulate a closed box. If the simulation of a passive radiator (PR) system is desired, the model can be expanded with L_{CES} and Λ_{ES} for the PR (C_P becomes C_{MES} and R_P becomes R_{ES} for the PR, all in parallel).

The circuit appears to be relatively complicated, even though the model has been kept as simple as possible. The circuit describes several conditions, such as unfilled volume versus filled volume and the volume of the fibers, their mechanical properties and absorption and thermodynamic effects.

All components can be determined using basic information about the enclosure volume and the amount of damping material, as well as four available parameters for the damping material. The actual calculation of the response is handled by a computer.

The components in Fig. 5 are explained as follows:

L_{UA} and C_{UA} represent the compliance and mass-loading of the "Unfilled Air" volume, i.e., these components represent parts of the volume without fill material.

L_{AF} represents the compliance of the Air volume with Fill material.

L_{TH} and R_{TH} represent the Thermal volume expansion and loss.

R_{B} represents the absorption found in the box regardless of damping material, i.e., the absorption which stems from mounting the transducer in the box, related to Q_{B}.

C_{MF} represents the mass loading from the Mechanical Fiber onto the system. Other mechanical fiber properties have no influence, provided that the fibers can be considered stiff.

C_{AF} and R_{AF} represent the mass loading of the air and air-flow friction due to the fibers in the Air volume with Fill material.

The author's AES paper [25] contains all the necessary equations.

To illustrate the model's ability to simulate different quantities of damping material in an enclosure, a Microsoft® Excel spreadsheet [32] is available. Fig. 6 shows the same simulation as above but the speaker box is equipped with a bass-reflex port. Three situations have been calculated: A) without damping material, B) with 38 gram of damping material, and C) entire enclosure filled with damping material.

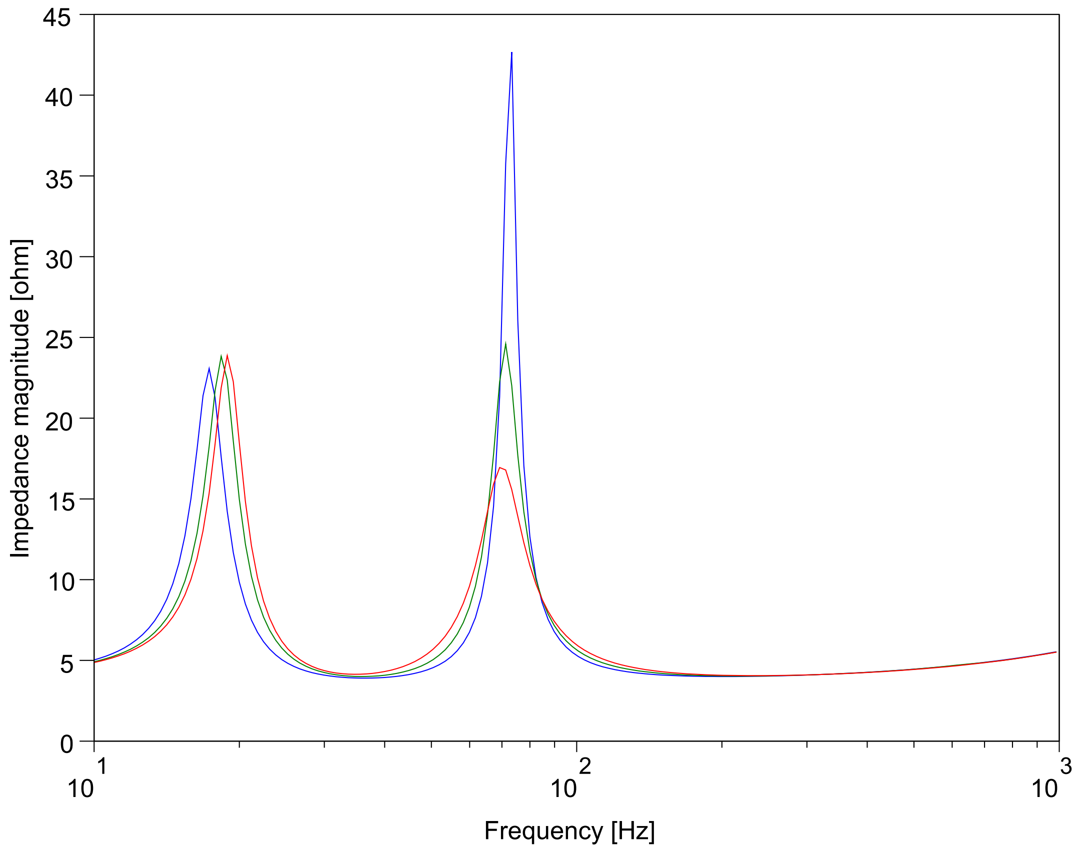

Fig. 6: Bass-reflex speaker without and with fibrous damping material. Blue curve is without damping material. Green curve is with 38 gram of damping material (box is 70% full). Red curve is with box 100% filled with damping material (54 gram). All curves are simulations based on the model in Fig. 5, with Q_L = 100 and Q_P = 100 (which means these losses are essentially ignored).

In the undamped box, the bass-reflex speaker's impedance has two peaks. The lower frequency peak has a lower magnitude than the upper peak. This trend is normal for bass-reflex speakers in practice and indicates the presence of frequency-dependent damping.

When damping material is added, the heights of the two peaks are more similar. This corresponds to a "normal" configuration of damping material in the speaker, and the system is now in the range where the conventional Thiele/Small model can be used. In practice, however, a typical bass-reflex box is not filled with such a large amount of damping material. The author estimates that conventional designs typically use less than 50% fill material in the box.

When a larger amount of damping material is added inside the box, filling the entire enclosure, the upper-peak is so heavily influenced that the function of the bass-reflex port diminishes. The system no longer operates as a bass-reflex system. In practice, the system will be more “choked” than the model indicates in Fig. 6 because the damping material will restrict the port opening; the effect for example on Q_P is not included in the simulation. Besides expecting lower peaks there should be a rise in impedance level at the port resonance frequency around 35 Hz.

In practice, bass-reflex speakers typically have two impedance peaks of different heights, whereas simulations with the Thiele/Small model typically show two peaks of (nearly) equal height. An impedance error of 1 dB gives rise to an equally large error of 1 dB in the frequency response. For the bass-reflex, this means that the Thiele/Small model perhaps simulates the response level correctly at the box resonance frequency (f_B) but has non-negligible simulation errors at the two frequencies where the impedance peaks are located (and all frequencies in the vicinity), which means that the SPL level in practice is higher over the f_B and lower below the f_B. The frequency response curve has the wrong slope in the simulation and generally does not follow the desired target curve (alignment error).

Consequences of the extended model

When the model for the loudspeaker is extended as in the above model, it creates several consequences. The traditional model takes full advantage of the connection to filter theory, where the concepts are unambiguously defined. With an extended model this is no longer the case. The terms “resonance frequency” and “damping factor” can be defined in several ways.

The frequency response in an extended model has a more complicated nature. Although the resulting frequency response appears to be smooth (to the naked eye) the response cannot be characterized and simulated accurately in a way that matches a simple filter. There are too many variables for 2nd order (or 4th order) filters to cover all the possibilities at all frequencies.

The entire concept with alignments established with the traditional model (e.g., “Butterworth” alignments, etc.) can no longer be used. It is not possible to analytically determine the impulse response of the system based on the filter alignment. Factors determined from filter theory must be disregarded and the system response must be studied in other ways. To determine the impulse and step responses in the extended model, a numerical method can be used to calculate the inverse Laplace transform of the frequency response.

Technically speaking, one could specify a target curve based on a filter alignment and then trim the model parameters of the speaker system (box size, cut-off frequency, amount of damping, etc.) so that the practical system would be similar to the chosen filter alignment, but this would be an artificial constraint in the process of simulating and adjusting the system.

Whereas the traditional model has always led to a roll-off characteristic plus a 100% horizontally straight and flat frequency response at higher frequencies, the extended model allows the possibility of computing the entire frequency response curve as a result of the simulated impedance response up to relatively high frequencies while simulating the correct Sound Pressure Level (SPL). The power response will be calculated unless the model includes features about the transducer directivity (including the baffle effect) and loss factors at high frequencies. In practice, the model can provide valid results as long as the speaker unit is operated in the piston range, which for woofers is typically below a few hundred Hertz.

When calculating the system's impulse response in the time domain, the result is strongly dependent on the system's behavior at higher frequencies. Because the extended model includes a roll-off at higher frequencies due to the inductance part of the model, assessing the system's impulse response is difficult. The situation becomes worse when a low-pass filter is applied. Low-pass filters, especially passive filters, play a significant role in the total system response and their influence should be taken into account when designing the box. The study of simulated impulse-responses is not useful in determining if the bass alignment is of high quality.

These conclusions apply not only for the presented model but for any model with similar ambitions. For example, if the traditional model is expanded with a simple L_E induction, f_S and Q_{TS} will not be given by the filter theory and will only be what one chooses to define. One of the reasons is that when L_E is included, the peak impedance and the zero-phase crossing points no longer coincide. Thiele most likely knew what he was doing when he omitted inductance from the model.

Users of any of the extended models (e.g., the *LEAP* model [22]) may have a tendency to specify the enclosure as too large. The reason is that the user of the system can propose an alignment that pulls the system output as far down in frequency as possible while maintaining a flat frequency response. The enclosure proposal is in some cases very large, providing a "boomy" bass reproduction with poor impulse response.

The author has been searching for alternatives to assess the quality of bass alignments so that an appropriate balance between the lower cutoff frequency and impulse response can be estimated and chosen.

Group Delay Guideline

The proposal implemented in the author's Microsoft® Excel spreadsheet [32] is based on Group Delay. This phenomenon is exclusively calculated in the frequency domain for simplicity. Because the calculation is based on a given frequency and its neighboring frequencies, the result is independent of occurrences at high frequencies. Group Delay denotes the phase-change relative to the frequency (i.e., phase distortion) and thus only describes this particular type of impulse distortion.

The audibility of Group Delay (GD) is studied in the general literature and in scientific papers [33, 34, 35, 36] without providing a clear picture. It shall be realized that a linear phase design will result in constant GD, which provided the frequency response is linear becomes a time delay and hence doesn't affect the shape of the signal. It can therefore be concluded that any given reference value for permissible GD is inherently flawed. As such, using for example the 30 ms limit as a reference, based on a study of Echo Detection by Helmut Hass [33] requires that the reference value is used under specific circumstances. Blauert and Laws [34] have studied the audibility of GD at higher frequencies (500-8000 Hz). Again it must be stated that the found GD threshold values must imply the use of a certain signal and other conditions, etc. (see the original paper [34] for details).

Instead it must be concluded that the phase distortion associated with GD can be audible, if it changes quickly with frequency.

When studying in particular the bandwidth limited response of a loudspeaker, looking at the high-pass response, applying the above found GD limits cannot be supported scientifically and must be considered rather controversial.

Anyway, the author has extrapolated the measurements of Blauert and Laws and found a usable curve that can serve as a guideline for Group Delay at lower frequencies, as shown in Fig. 7. The curve reaches 30 ms at approximately 17 Hz, thus showing an apparent link between GD, echo detection and lower audible limit frequency. The author is not an expert in psychoacoustics and cannot assess whether this is a coincidence or whether such a relationship exists.

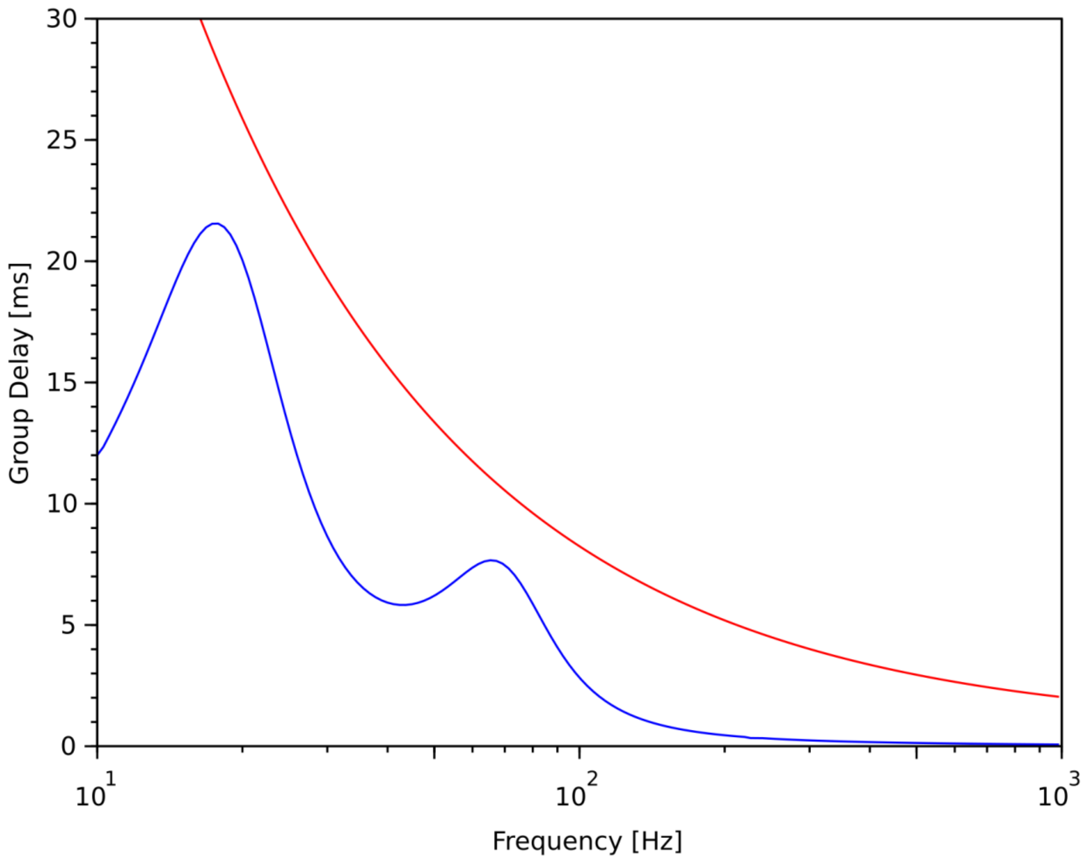

Fig. 7: Group Delay plot of the bass-reflex system from the previous example (blue curve) and the guideline (red curve).

The mathematical expression for the Group Delay guideline is:

$$GD_{limit} = \frac{1000 \cdot 1.1606}{5.6413 \cdot f^{0.81511} - f}$$

Eq. 1: The suggested threshold limit of audible Group Delay (in milliseconds) in the audio range, where f is the frequency [Hz].

The number of significant digits in equation 1 does not reflect the precision, but serves as a reference for others to be able to reproduce the same graph with visually exactly the same result.

This guideline, which the author proposes for the appropriate dimensioning of bass enclosures, cannot be seen as a sharp boundary between the audible and non-audible. In reality, the audibility of GD is not well understood and still subject to debate. Zwicker and Fastl [35] described that it is dependent on the input signal: they provide an example of two test signals with different GD where the phase distortion is equally at the audible threshold according to listening tests. The goal in speaker system design must, however, be an inaudible GD, even with the most impulsive input signals.

When designing a speaker, first and foremost evaluate the frequency response, the linearity as well as the roll-off characteristic in the bass. Secondly consider taking a look at the impulse response. When taking a look at the GD, remember that first and foremost the GD shall be without sharp changes. It can also be questioned whether larger changes can be considered inaudible if the frequency response has dropped off, say, more than 20 or 30 dB.

The author has found this guideline to be helpful in sizing the final enclosure volume such that the best compromise between the low end roll-off (high-pass frequency response) and proper impulse response is chosen. In practice it has proven in a couple of cases to correlate with his own and others' perceptions of alignments with good bass reproduction. Admittedly the author doesn't know how or why it seems to work in practice and the guideline is presented without scientific backing.

The traditional model typically produced bass-reflex speaker designs with an optimal transducer of $Q_{TS} = 0.32$, placed in a box where the closed box system has a $Q_{TB} = 0.71$, resulting in good bass response by introducing a suitably tuned port. With the extended model, one can no longer maintain a single point as the optimum; a number of optima exist that apply to different transducers with different model parameters.

Therefore it is recommended to test the suggested reference curve on your own designs, either on existing designs that you can reverse engineer (and possibly redesign with different cabinet volume and tuning for further assessment) or try it out on your next design.

The author believes that this guideline can be used for both closed boxes, bass-reflex boxes and other enclosures, including active equalized systems such as sixth-order bass-reflex systems.

In practice, the system impulse response and the GD response will be significantly dependent on external factors such as the listening room's acoustics, which can significantly contribute to the overall perception of the quality of the system, especially at lower frequencies (and can to some extent be handled with active EQ, possibly with a Digital Signal Processor). L. R. Fincham [36] even mentions the recording studio as another source of Group Delay.

The proposed Group Delay guideline is not dependent on any particular model and could also be used with the traditional Thiele/Small (T/S) model, but system GD is calculated with higher precision when a better model is applied.

Limitations for the T/S approach versus limitations for the extended model

The traditional Thiele/Small model has a relatively large limitation in relation to the simulation of modern speaker systems. An incorrect transducer model is combined with an incorrect enclosure model; then the total system is simulated. This works well in the situations that were known when the model was introduced, namely systems that use damping material in the box and for situations where $\alpha = V_{AS} / V_B$ is fairly small, i.e., the enclosure is relatively large.

When speaker system designers moved away from the beaten path, they found that the simulations could not predict what happens in practice. This was particularly true with modern (small) boxes and systems with no or very little damping material.

Small boxes are a challenge. When people design relatively large enclosures and the change in resonance frequency is relatively small, the change in transducer damping with frequency is also relatively small. Increasingly smaller enclosures cause a significant change in resonance frequency (sometimes combined with active EQ), meaning that further extrapolations from the known origin (model parameters determined from measurements in free air) and differences in impedance peak height become more pronounced.

The model presented here extends the traditional enclosure and transducer model and simultaneously extends the possibilities for the model to include variable amounts of damping material. The model provides a better understanding of what actually happens in the speaker system and separates the effects from the transducer from the effects related to the enclosure.

The traditional model can be made to fit if one adjusts practices to the model, but this is not the purpose of a model. The model must provide the means of simulating the desired speaker before building the first prototype, preferably without inherent restrictions on design choices. Any speaker that is designed and simulated with the traditional model should be verified with measurements to determine how closely it meets the expected outcome and possibly make necessary adjustments.

Verification by measurement of the system is still recommended, but the extended model should give rise to adjustments that are smaller than if one had used the traditional model.

The extended model, shown here in all its simplicity, shows the limitations of bass-reflex simulations when large amounts of damping material is desired or where damping material is placed in non-conventional ways, e.g., exceptionally close to the transducer. A further improvement would require geometric aspects to be included into the model, for example the use of a Finite Element Analysis (FEA) or, as illustrated by Putland [28], a Finite Difference approach.

In addition, the extended model will be limited by its design as a linear "small-signal" model, which means that all the components in the circuit are assigned a fixed value. The nonlinearities can make the simulations more complicated to interpret, and there is a risk that after a thorough study of the nonlinear effects, the designer may be more confused. Sometimes more information is not better.

The extended model presented here is only capable of simulating damping in a simplified way. In practice, such phenomena are generally not linear. The importance of the extended model lies in its ability to separate transducer and enclosure effects.

Furthermore, the FDD model provides a superior simulation of the inductance in the speaker unit. A better simulation of the electrical side provides for a better separation of electrical versus mechanical (motional) impedance; this is important for box simulations because the box manipulates the mechanical side. The entire goal of defining transducer model parameters is to separate electrical and mechanical parameters so that the mechanical side can be correctly manipulated (with a box) and extrapolated.

There is an additional limitation in the presented FDD model that we have not previously considered. The model considers the suspension stiffness as a fixed value, independent of frequency. In practice, the stiffness changes with frequency, meaning that the calculated enclosure volume will typically be too small. Other factors come into play, such as the diaphragm area, S_D, which is normally not known with great accuracy. In practice, one should therefore build the box to be approximately 10% larger than calculated to obtain the expected resonance frequency of the system.

Conclusions and Summary

Presented in this chapter is a minimal extension of the Thiele/Small model to correct the damping in the audio range. The model is insufficient for true dynamic simulations but well-suited for frequency-response simulations and the sizing of enclosures.

The FDD model combined with the extended box model aims to improve the level of understanding and hopefully provide opportunities for further improvement efforts, not only within the small-signal domain but also outside these limitations. The author's goal for the presented model is to allow for simulations that are equally as good as or better than the traditional model.

The simulations are expected to be at least as good if the designed speaker system lies within the framework of what can be simulated with the traditional method; in the case of more specialized situations, the extended model will be significantly better.

The presented transducer model requires relatively few parameters and can therefore be placed in an ordinary product data sheet for the speaker unit. The extended model is publicly available and can be implemented by anyone who may wish to do so.

The presented model shows that there is a physical disconnection covered by a confusion of concepts in the traditional model. There may have been attempts to fix this, either on the transducer side or the box simulation side, but verification of such one-sided attempts would have shown a lack of improvement in simulations of the overall system response. The reason is that Richard Small skillfully combined the transducer and the enclosure in his model. To improve overall simulations of the system, one must make improvements on both sides; both the transducer and enclosure model must be improved. It shows the strength of the "system" mindset that has characterized the work of Richard Small and Neville Thiele.

Small's model may be an oversimplification of reality, but this can be defended. The models used in this study for the extended model were developed in a period from 1989 to 2002–more than 25 years after Richard Small presented his results. Using contemporary models of absorption (see Beranek [3]) may not have resulted in any improvement whatsoever. Implementing improvements of the models would have entailed a great deal of work and may not have resulted in an improved understanding of the model at the time, and therefore may have reduced the success of the method. Simulations of normal situations with the standard Thiele/Small method were within approximately 1 dB–a level of precision previously unheard of.

Small's work forced the manufacturers to document their transducers with parameters. The overall model provides a better prediction than the simplifications of the transducer and enclosure models imply because of the combination of the two. The fact that the Thiele/Small model has been the *de facto* standard for so many years is a testimony to the qualities of the model and should not be ignored.

The question is whether it is time to set aside the traditional model and update it to a more modern standard, not only as a result of the developments that have been made since 1973 but also as a result of the computational possibilities inherent in the use of modern computers.

An extension to the traditional model will require reconsideration of conventional engineering methods and practices such as defining the resonance frequency, f_S and the damping, Q_{TS}, of the driver (if the information is still desired), as well as how to approach losses in enclosures and how to evaluate the quality of a simulated bass alignment. The Group Delay approach is suggested, not as a finalized method but as a suggestion for further discussion. The author has used this guideline and found good agreement with listener preferences.

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