Chapter 7. Car Acoustics

Background

As a listening environment, the passenger cabin of a car leaves a lot to be desired. Space is rather limited, so the loudspeakers can only be mounted where there some room left to do so, and where the sound would not be muffled or obstructed. Often, this would be in door panels, or in the rear deck under the window, so that the interior of the trunk can be utilized as a closed box. In order to model harmonic acoustic behavior of an enclosed space with a sound source, the following equation is sufficient:

$$(K[][] - k^2 M[][] + j\omega C[][])p[] = j\rho_0 \omega v[] \qquad k = \frac{\omega}{c} \qquad \omega = 2\pi f \qquad j = \sqrt{-1}$$

Where: K[][] is called acoustic stiffness matrix, M[][] is the acoustic mass matrix, C[][] is the damping of the system, p[] is the sound pressure vector, v[] is the excitation vector in qubic meters per second, f is the test frequency and c is the speed of sound.

For the source of the sound one can assume a “point source”, which is convenient, as it can be located in any of the mesh nodes. Larger sources would have to be accounted for as a part of the room boundary. Mathematically, the problem now reduces to assembly of the K[][], M[][] and C[][] matrixes and inverting the expression in the brackets. This way, vector p[] can be found for any frequency and location of the sound source represented by excitation vector v[]. Vibration of the cabin surfaces can alter the sound generated by the loudspeakers to various degree. The low quality, vibrating doors can actually resonate to the point, that the door panel behaves like a diaphragmatic absorber. This effect can alter the cabin’s resonant modes and the pressure distribution quite significantly. Taken to the extreme, the cabin would be able to maintain ¼ wave resonance. Proper mathematical treatment of this type of problem would involve “fluid-structure coupled systems”, where the air trapped in the interior of the cabin (fluid) influences panel resonances (structure) and vice-versa. The issue is quite complex and involves knowledge of physical properties of materials used for cabin construction. Fortunately, the quality of the interior finish in today’s cars tends to be better, so that at low-to-medium sound levels, the panel vibration is less influencing the sound than in cheaper, yesterday’s models with tinny doors.

In this release of the program, we would like to concentrate on somewhat simplified approach, where we assume, that vehicle interior is a well built, solid structure, with most panels exhibiting damping effect (soft but good quality finish). This damping is modeled as uniformly populated damping square matrix C[][] in the FEM analysis. In addition, you can assign absorption coefficient to each node of your mesh.

If you are professionally involved in this subject and would like to have your unique model analysed, please contact Bodzio Software with the model dimensions and geometry. We will implement it and send you updated software free of charge.

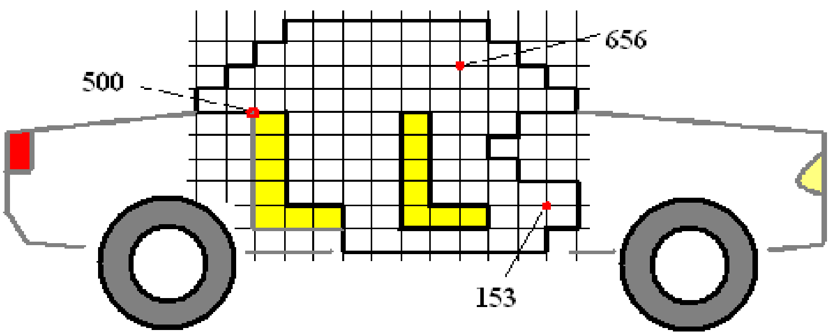

Figure 7.1 “Brick” element approximation of the passenger cabin

Furthermore, we will use simple “brick” elements to approximate the complex geometry of the passenger cabin – the white, or non-shaded area on Figure 7.1. With this in mind, we can proceed to estimate the modal influence of the cabin on the low end of the frequency response of the loudspeaker mounted in various places inside the cabin. Figure 7.1 shows the chosen locations for loudspeakers:

Speaker Location 1 – rear speaker, mounted behind the back seat (Node 500),
Speaker Location 2 – front door speaker, mounted low on the right front door (Node 153),
Microphone Location 1 – approximate position of the driver’s ears (Node 656).

Finally, we can enter the above model into the computer FEM program to obtain the 3D-view like the one shown on Figure 7.2.

The Analysis

It is often beneficial to determine the resonant frequencies of the cabin first. The actual modal frequencies and the corresponding distribution of the acoustic pressure are a great help in identifying the peaks of the frequency response of the cabin itself. The FEM analysis revealed several modal frequencies below 200Hz. As you may expect, the lowest mode (65Hz) follows the longest dimensions of the car, with peaks at rear window and front pedals – see Figure 7.3. Acoustic pressure distribution for the second lowest mode is shown of Figure 7.4. Secondly, the modal analysis allows me to determine the areas to avoid for possible locations of the speaker. The speaker placed in the exact modal spot for a given frequency will produce little acoustic output on this frequency. Obviously, it goes without saying, that choice of mounting locations for loudspeakers is quite limited, as the car was designed as a means of transportation first, and as a listening environment last.

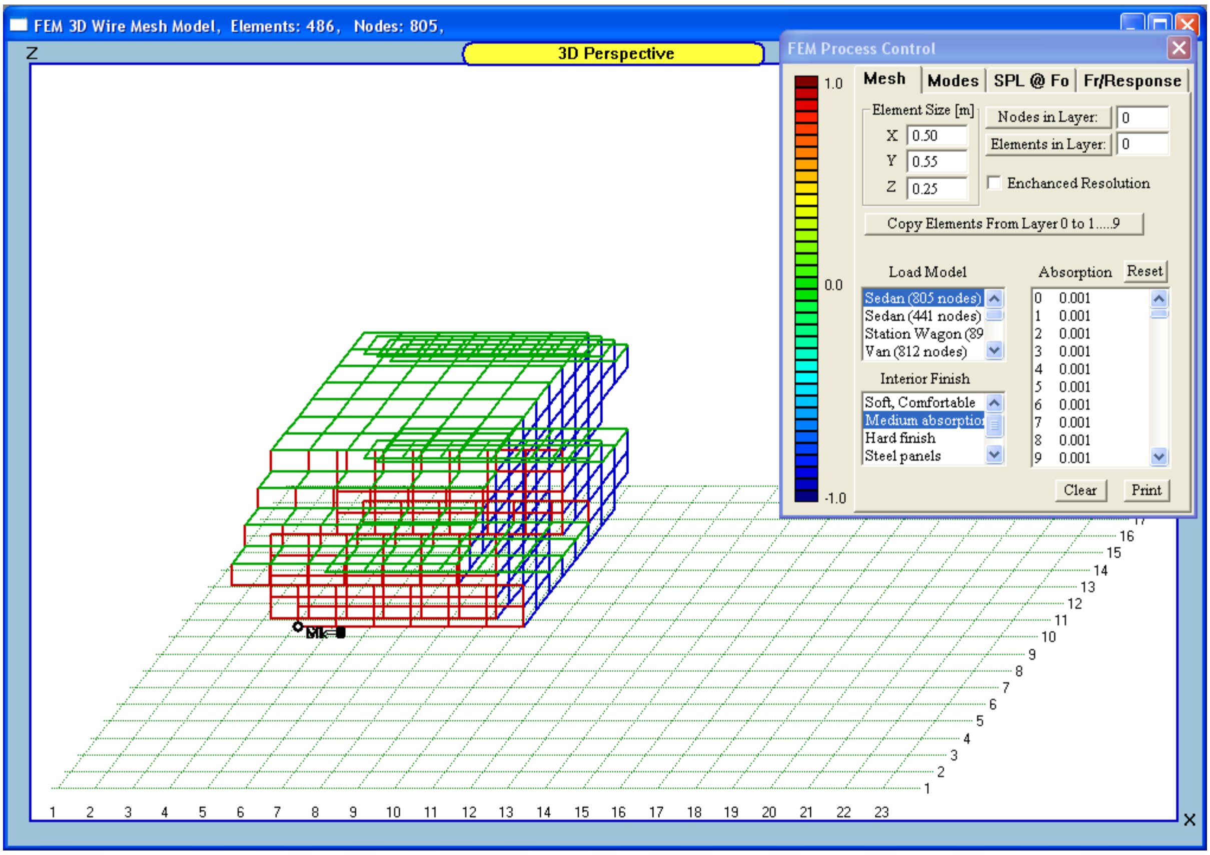

Figure 7.2. 3D-Passenger compartment model.

For the purpose of further analysis, we assume that my loudspeaker has a perfectly flat frequency response. Using such a source is quite legitimate and helpful in understanding what influence the cabin finish and geometry has on the frequency response of a typical loudspeaker. When the “perfect loudspeaker” is used, the only visible amplitude irregularities are the one’s that are introduced by the environment.

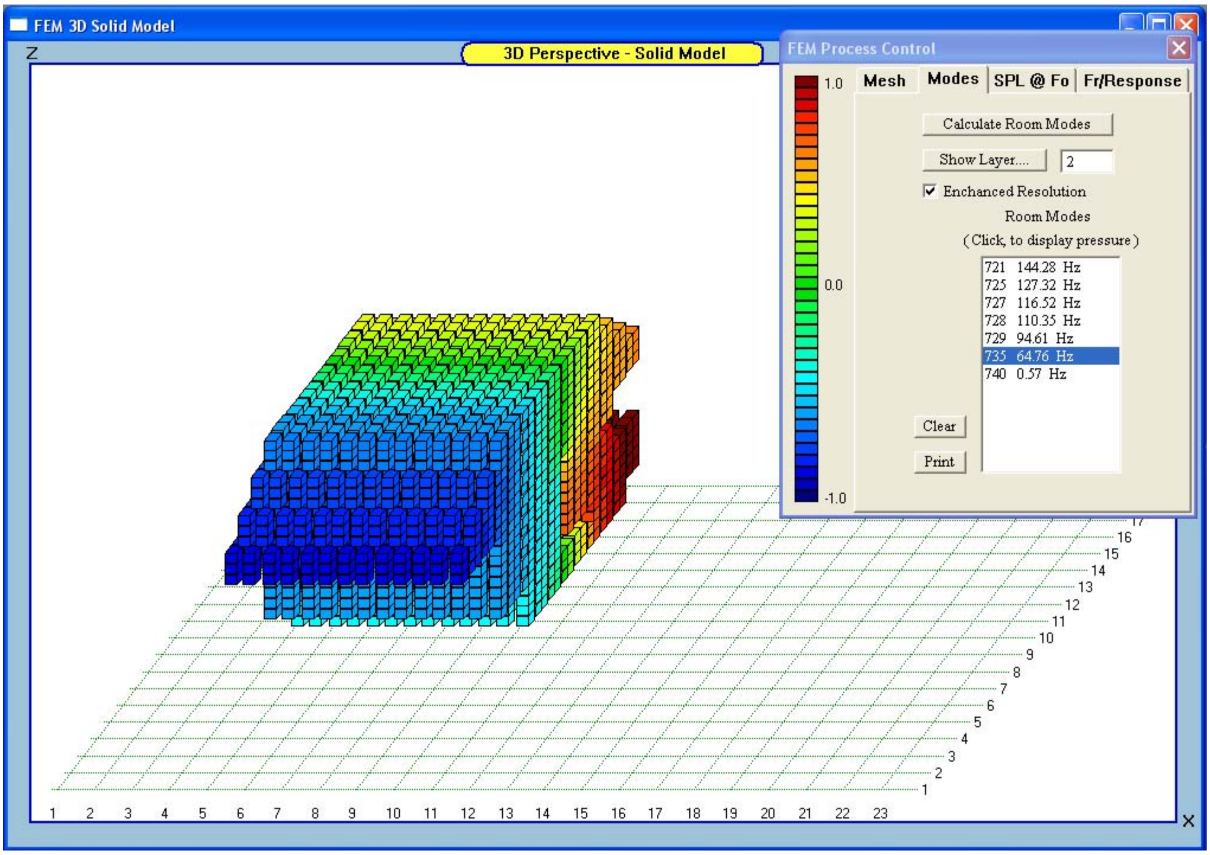

Figure 7.3. Acoustic pressure distribution for lowest mode. Red/blue color indicates high pressure and green color shows modal line.

When a “perfect loudspeaker” is mounted at the back of the cabin, under the rear window, and the listening position is assumed to be where the driver’s head is, the modeled frequency response of such “perfect loudspeaker” is shown on Figure 7.5. Closer inspection of the plot would indicate: (1) there is overall gain of about 5-6dB averaged across 10-200Hz frequency range, (2) the plot exhibits a +12dB slope below 50Hz – this is due to the cavity effect, and (3) the frequency response is quite irregular between 70-95Hz and then improves as modal density increases towards 200Hz. The 65Hz mode determined from my earlier FEM analysis appears to be quite pronounced for the amount of dumping assumed earlier. Incidently, a closed box with corner frequency at around 50Hz, mounted in this car, theoretically can produce flat acoustic frequency response to DC. The 12dB/octave slope of the closed box, would ideally be compensated by the 12dB/octave rise from the cavity effect.

When the loudspeaker is mounted in the lower section of the front door, the modeled frequency response of my “perfect loudspeaker” is shown on Figure 7.6. Once again, closer inspection of the plot would indicate: (1) there is overall gain of about 7-10dB averaged across 10-200Hz frequency range, (2) the plot exhibits a +12dB slope below 40Hz – this is due to the cavity effect, and (3) the frequency response is quite irregular between 50-120Hz and then improves only slightly as modal density increases towards 200Hz. The 65Hz and 95Hz modes determined from my earlier FEM analysis appears to be quite pronounced for the amount of dumping assumed earlier. There is significant energy concentration between 55Hz to 115Hz. Also, the sharp dip at 52Hz is likely to audiable. Loudspeaker frequency response will be strongly influenced by the windows position. We have restricted my analysis to windows up position, which seals the vehicle interior and causes the cabin to exhibit resonances. Secondly, vehicle in these conditions exhibit the cavity loading which will introduce a 12 dB per octave rise in the bass at low end of the audio spectrum. The corner frequency can be estimated from the major dimension of the passenger cabin and varies between 45Hz to 90Hz with the higher frequency assigned to small cars.

If it was possible to designed vehicles with perfectly sealed passenger compartments, the loudspeaker cone moving forward would pressurize the cabin and the pressure would be maintained as long as the cone stays in the “forward” position. This is really equivalent to a DC (0 Hz) acoustic response.

The plots shown above for two different loudspeaker locations captured both effects: cabin resonances and +12dB/oct rise towards the low end of the audio spectrum. For the vehicle of the size chosen for my model, the corner frequency would be between 45Hz and 50Hz.

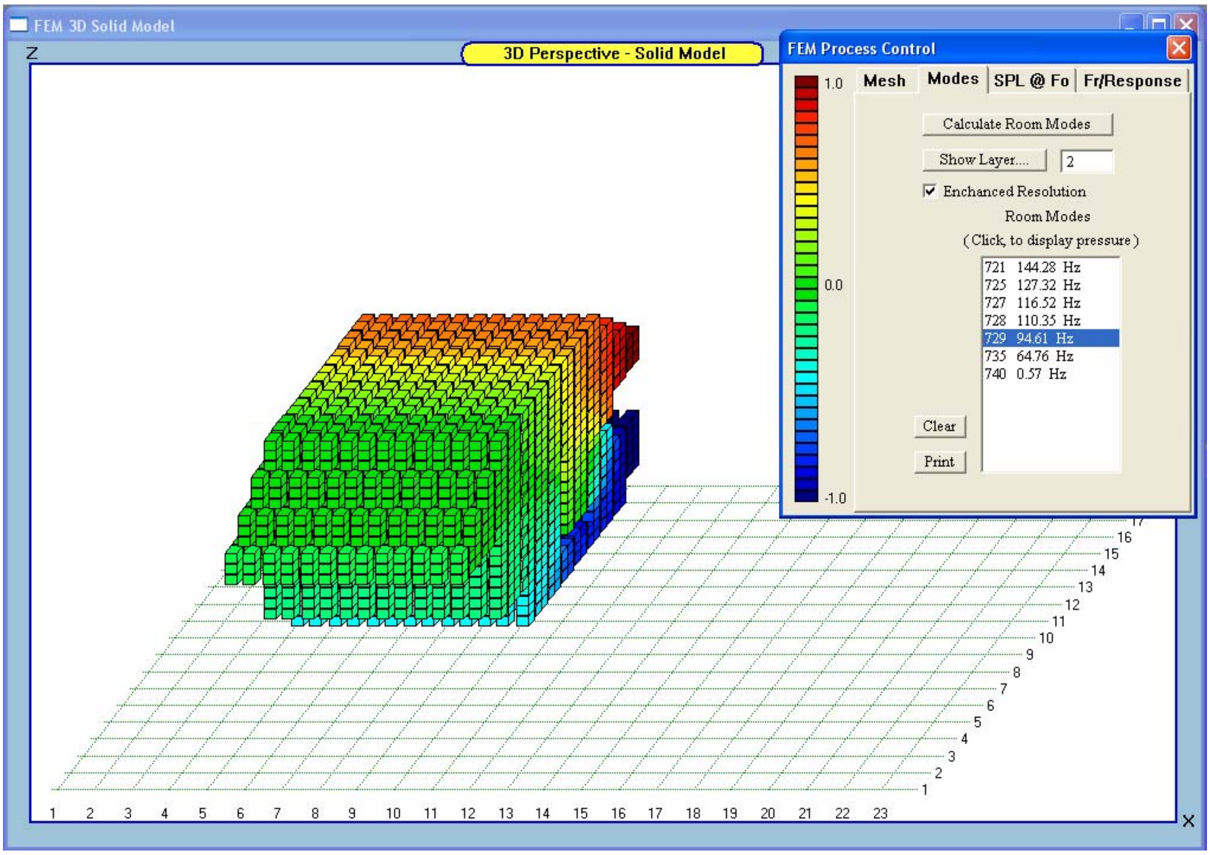

Figure 7.4. Acoustic pressure distribution for the second lowest mode.

Our earlier assumption, that the source of the sound is a "point source" implies, that it radiates the sound uniformly, in all directions. This assumption agrees well with restricting the analysis to about 200Hz, below which all sound sources appear to be omnidirectional. When both speakers are operating at the same time, the combined frequency response would be similar to the one shown on Figure 7.7. Despite its "bumpy" character, the combined frequency response would be preferred over each of the individual plots obtained for single driver operation. The peaks and valleys can not be avoided in the closed compartment, but their influence can be significantly reduced by making the interior "softer" (read: more absorbing). The remaining irregularities can be further reduced by electronic equalization, thus making the total frequency response more acceptable.

In the above short discussion, we have presented one possible approach to modeling car acoustics. The discussion was based on the steady state cabin response, where the compartment modes carry the acoustic energy. With the additional assumptions of: (1) "de-coupling" the volume of cabin air resonances from the car body resonances and (2) interior panel dumping was represented by uniformly populated dumping matrix C[][], it was possible to obtain frequency response plots for a couple of loudspeaker-microphone locations. The plots show clearly the modal (resonating) behavior of the passenger cabin together with the pressurization of the cabin at lowest audio frequencies (sometimes called the cavity effect).

The set of plots (Figure 7.7) present quite realistic situation of using contributions from two loudspeakers, with microphone placed in the driver's head horizontal plane. The accuracy of the analysis in the upper range of frequencies was limited by the size of the "brick" elements we have used. The dimensions were: Y=22cm, X=26cm and Z=14cm. Smaller elements would increase accuracy around 200Hz, but the penalty would be significantly increased computational time. The final frequency response shown on Figure 7.7 seems quite acceptable taking into account where we started (Figures 7.5 & 7.6).

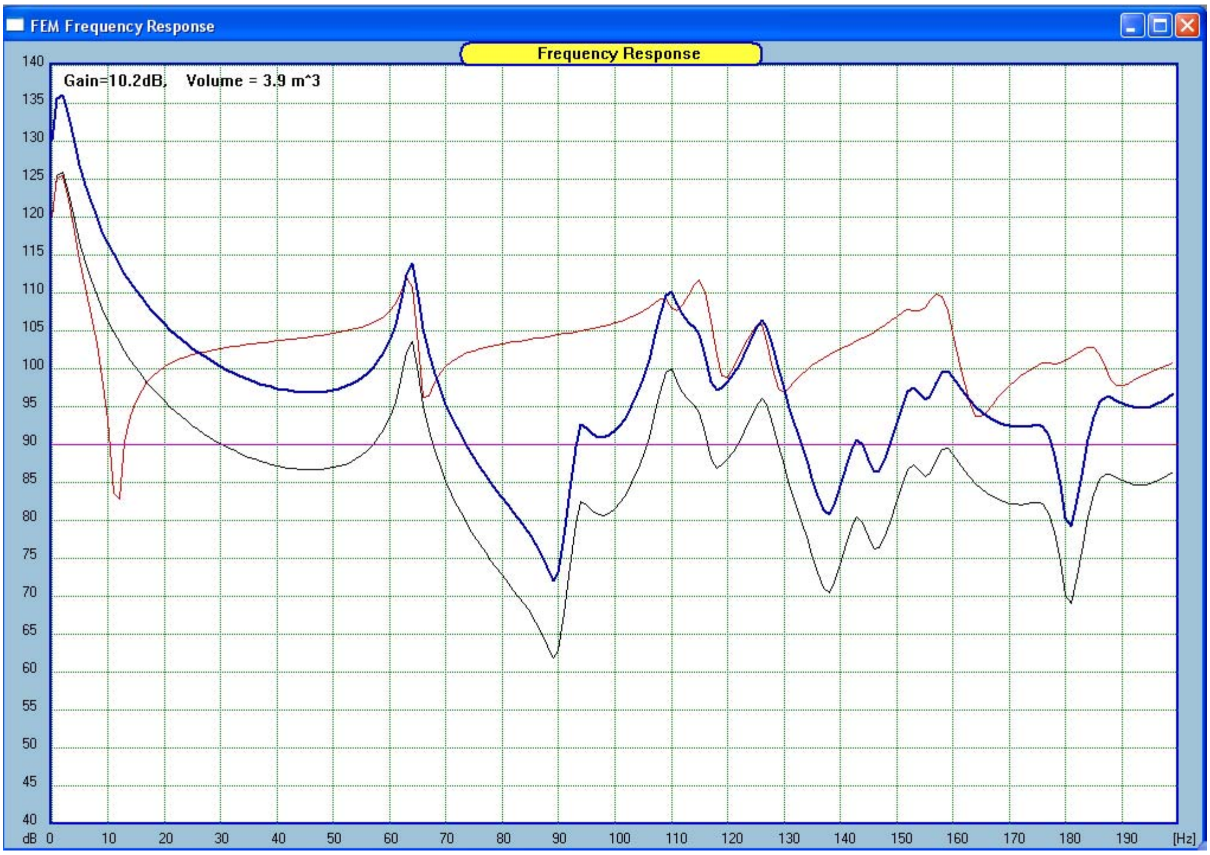

Figure 7.5. Loudspeaker at the rear of the passenger cabin (blue).

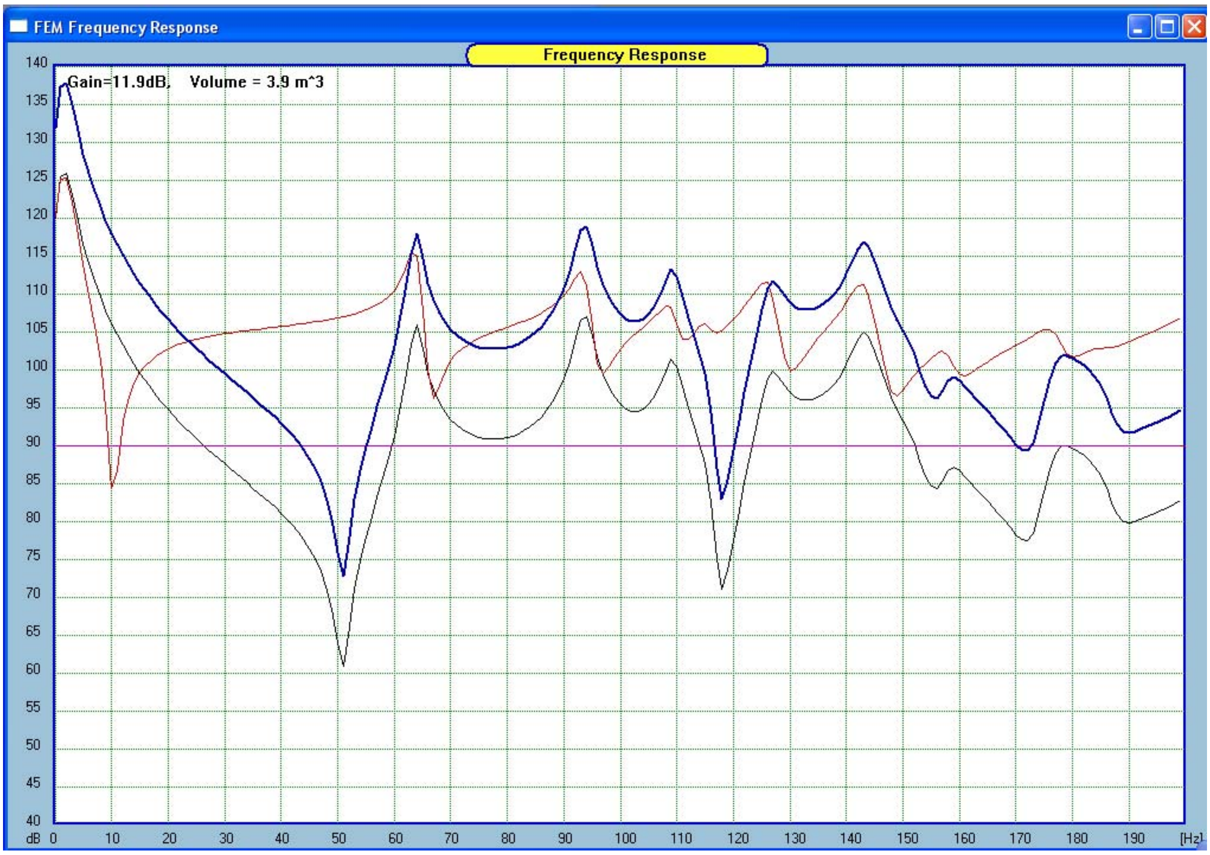

Figure 7.6. Loudspeaker mounted in the front door (blue).

Operation of the 3D FEM Car Audio Functions is explained in details in Chapter 15.

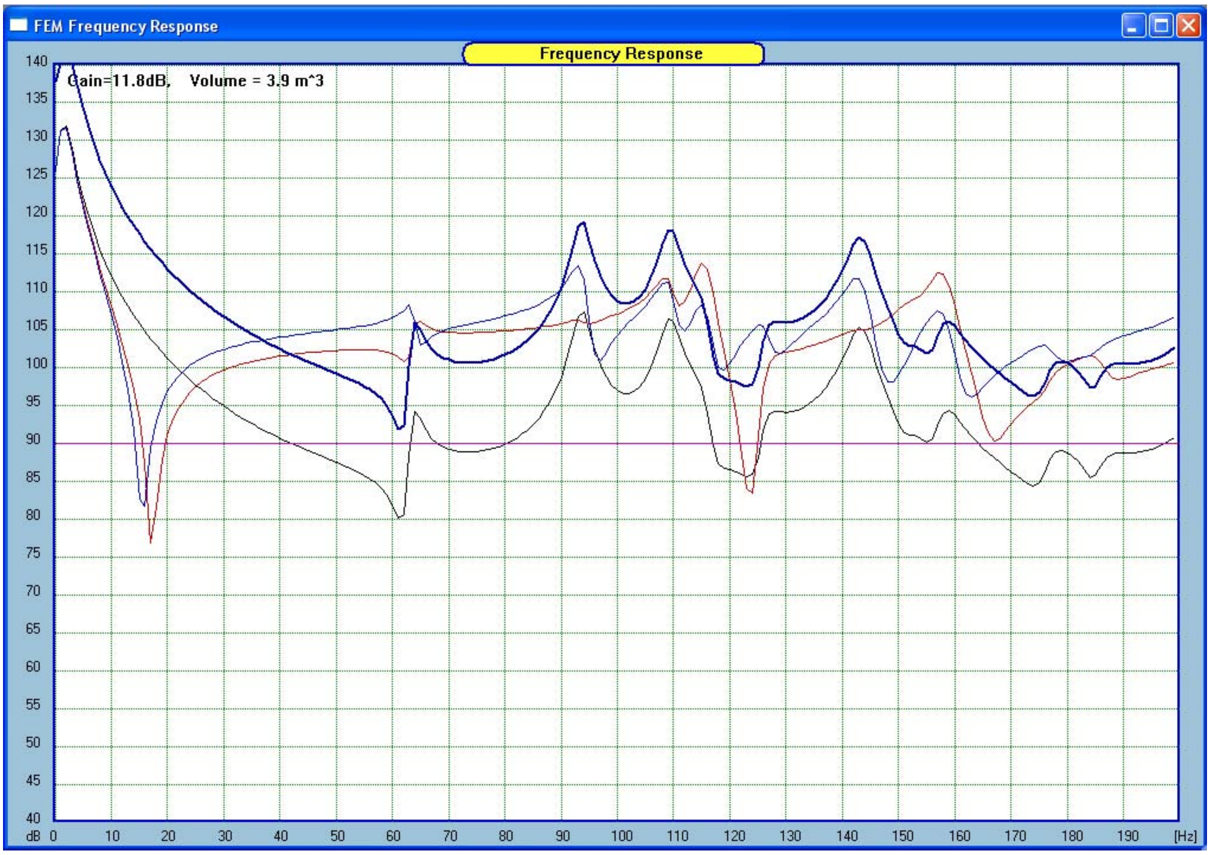

Figure 7.7. Frequency response due to SPL of two speakers combined (blue).

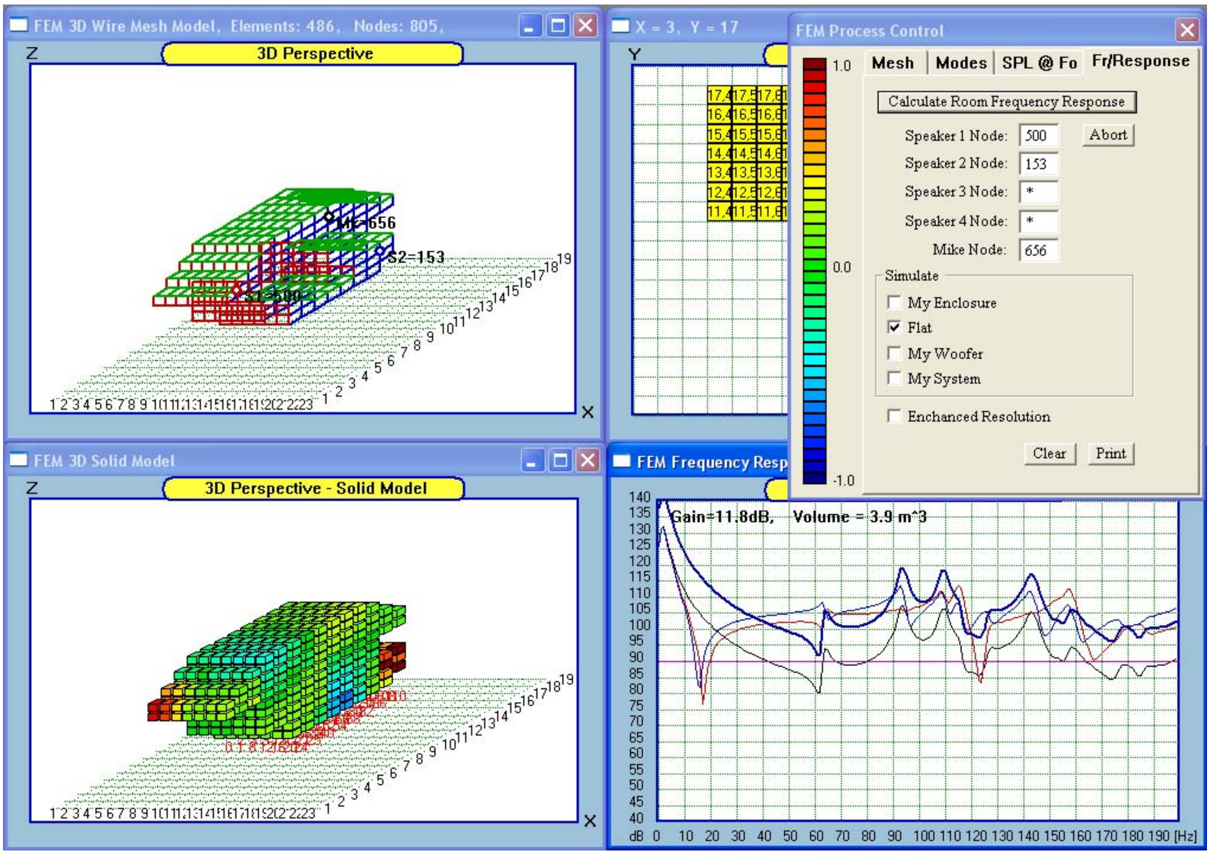

Fig 7.8 Selecting nodes for our car model.