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CAE Interior Cavity Model Validation using Acoustic Modal Analysis

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ABSTRACT

The ability to predict the interior acoustic sound field in a vehicle is important in order to avoid or to minimize unwanted noise conditions, such as boom or high pressure levels at cavity resonance frequencies. In this work an acoustic modal analysis is carried out for a minivan. The testing procedure is discussed and some results are shown. With the seats removed and for low frequencies the interior of the vehicle is similar to a rectangular box for which an analytical solution exists. At higher frequencies and with the seat, the interior acoustic field displays complex mode shapes.

INTRODUCTION

The interior acoustics is an important part in the customer perception of quality and overall vehicle value. At low frequencies it is important to manage modes so that cavity resonance frequencies are well separated from panel modes, engine operating frequencies, etc. Therefore CAE predictions in the early development stage and when it is still possible to make changes that affect the interior acoustic have high significance to a vehicle program. To ensure CAE quality, validations are carried out comparing predictions and measurements.

Before the full vehicle implementation, a validation exercise on a simple structure is made. A simple rectangular box shape, approximately the shape and size of a minivan is chosen. This has the advantage that an analytical model exists.

In the next step, an acoustical modal analysis was made on a minivan. Two cases were studied; standard vehicle conditions and with the seats removed. A roving array of microphones spanned the vehicle interior. The FRF response with respect to the source accelerometer was recorded. A woofer in the rear of the vehicle was used as the source and a small accelerometer was mounted on the center of the woofer. A swept sine signal from 10 to 250 Hz was used.

The results of this exercise have lead to improvement in CAE predictions and will guide improvements in future vehicle programs. Other findings include that the addition of seats into the vehicle interior cavity has two major effects; lowers the modal frequencies because of propagation path changes and added volume-absorption which dissipates acoustic energy and also as a result lowers the sound pressure amplitudes. The first two modes of the minivan are longitudinal; i.e. along the *x* axis. These modes are clearly identified in mode shape animations.

THEORETICAL MODEL OF A RECTANGULAR BOX

The interior of a minivan resembles a closed rectangular volume which has a simple analytical solution for its natural frequencies and acoustical modes (Blevins, 1984).

The natural frequencies can be calculated from

$$f_{ijk} = \frac{c}{2}\sqrt{\left(\frac{i}{L_x}\right)^2 + \left(\frac{j}{L_y}\right)^2 + \left(\frac{k}{L_z}\right)^2} \quad \text{(Hz)} \qquad (1)$$

and the mode shapes using

$$\Psi_{ijk} = \cos\frac{i\pi x}{L_x}\cos\frac{j\pi y}{L_y}\cos\frac{k\pi z}{L_z} \qquad (2)$$

with L_x = 3.8 m and is measured from the front bottom of windshield to lift-gate center, L_y = 1.7 m and is measured as the average distance between two sliding doors and L_z = 1.2 m and is the average height of the cavity measured from floor to roof. The speed of sound, calculated at room temperature, c = 343 m/s and the indexes for normal modes of vibration i = 0, 1, 2, … , j = 0, 1, 2, … and k = 0, 1, 2, …

A top view of the minivan, the rectangular box approximation and the associated dimension coordinates are shown in Figure 1. In Figure 2 is shown the calculated natural frequencies for the first 10 modes of the rectangular box with dimensions approximating the vehicle interior volume.

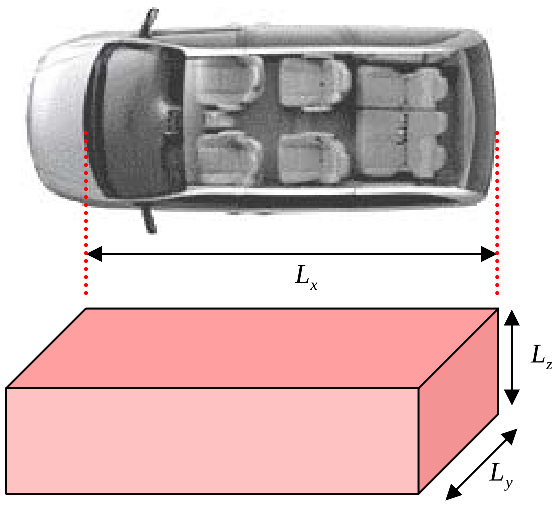


Figure 1. Top view of the vehicle and the 'equivalent' box dimensions.

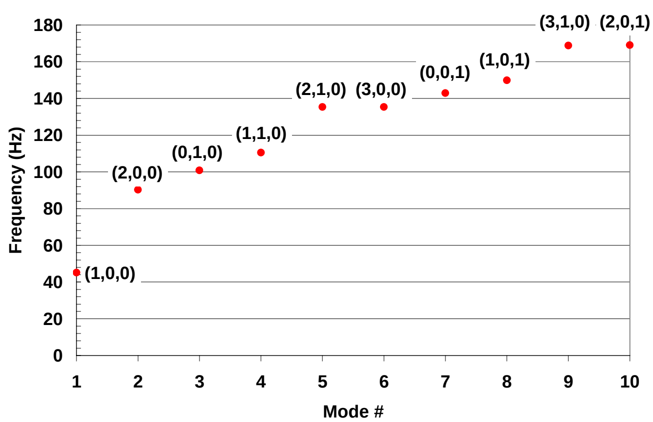
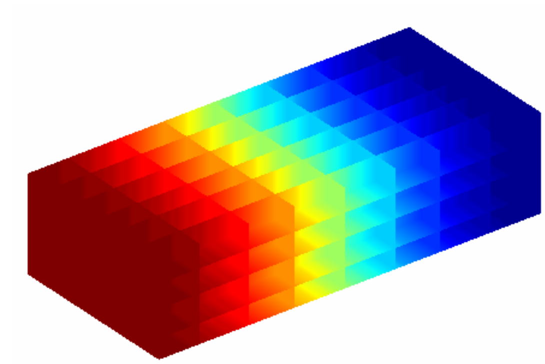


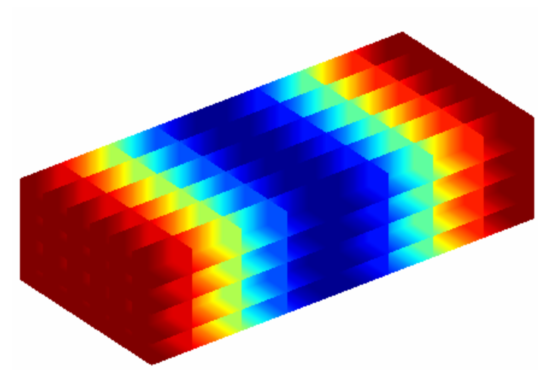
Figure 2. The first 10 natural frequencies of the 'equivalent' box, mode numbers and their index descriptions.

The differences between the vehicle and the 'equivalent' box include the seat and the non-uniform surfaces and the various boundary conditions such as the hard trim, carpets and headliner.

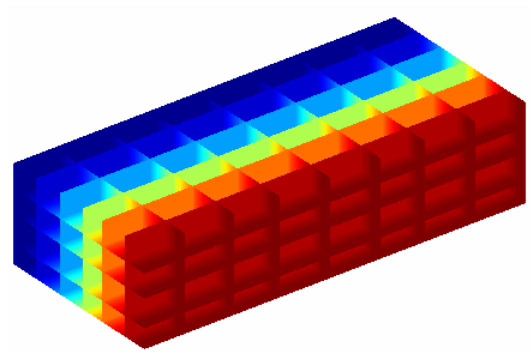
Figure 3 shows the idyllic mode 'shapes' at one instant in time. For acoustic modes the 'shape' can be represented by assigning a color scheme to the sound pressure level, for example, red assigned to a high pressure and blue to a low pressure. To animate a mode 'shape', the phase is included.



Mode 1, 44.2 Hz, (1, 0, 0)



Mode 1, 88.4 Hz (2, 0, 0)



Mode 1, 98.8 Hz (0, 1, 0)

Figure 3. The first 3 theoretical rectangular box mode shapes.

TEST SET-UP

A roving array of up to 7 microphones, Figures 4 and 5, spanned the interior and was moved along the length (front to back) and height (floor to roof) of the vehicle which defined a measurement grid.

A woofer in the rear of the vehicle was used as the source and a small light weight accelerometer was mounted at the center of the woofer to give a reference signal, Figures 6 and 7. An acoustic excitation as opposed to a structural-acoustic excitation is sufficient if one is only interested in the acoustic behavior of the cavity (Wyckaert, 1994).

The Frequency Response Functions (FRFs) between microphones responses at all the grid points with respect to the source accelerometer on the woofer were measured (P/a).

A swept sine signal from 10 to 250 Hz was used.

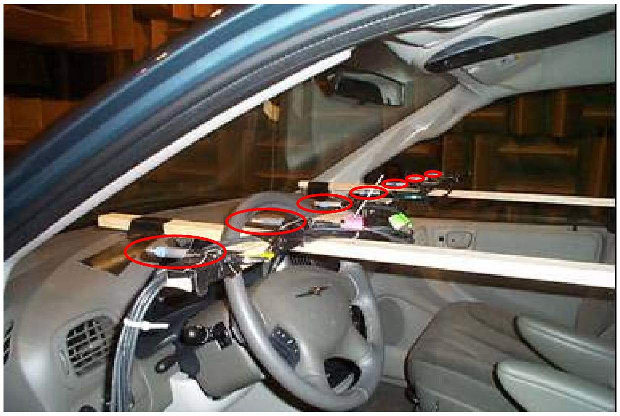


Figure 4. Red ovals show the location of the seven microphones in the movable array. In areas that were too narrow, five microphones were used. Two wooden strips extending from the front to the rear of the vehicle support the roving microphone array.

Figure 5. Wooden frames were used to raise and lower the movable array throughout the vehicle.

Figure 6. A woofer is located at the rear corner of the cargo floor in the vehicle. The woofer was isolated with soft foam from the vehicle body. An extra microphone was kept located close to the source as an added control.

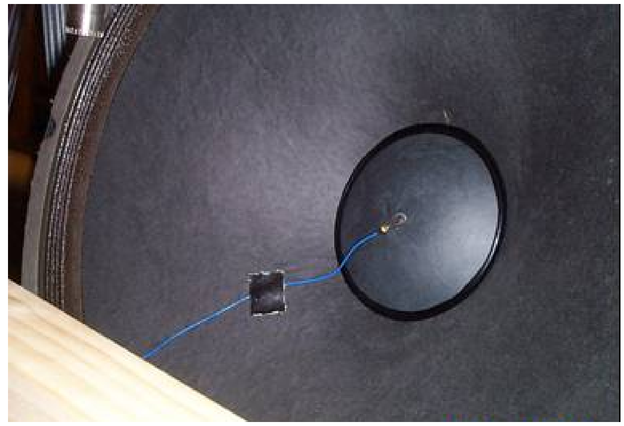


Figure 7. An accelerometer mounted on the dust cap in the center of the woofer was used to measure the acceleration and as the reference signal.

MEASUREMENTS

In total, 209 FRFs were measured, see Figures 8 and 9. The data was collected and analyzed using LMS Test.lab.

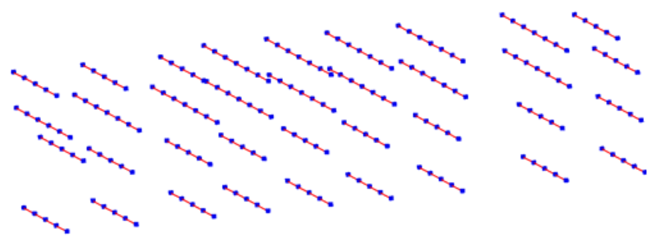


Figure 8. Measurement grid showing microphone locations.

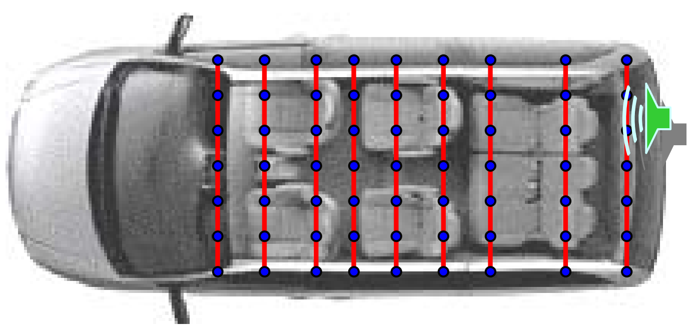


Figure 9. Speaker placed at the cargo floor looking upward and isolated from the floor with soft foam. Microphone rows seen from above, note that measurements were made near ear locations in all three rows.

Figures 10 to 12 show typical FRF results along the *x*-, *y*- and *z* directions. These were used during testing to ensure that the equipment was working properly and the testing was correct. If one or more results looked out of place, then actions were taken to determine why.

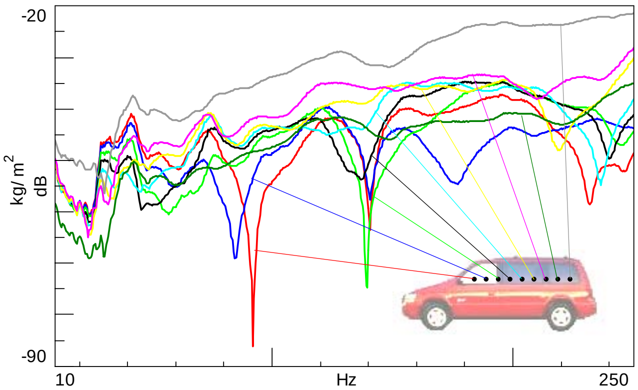


Figure 10. Example of measurements along a straight line in the *x*-axis at the center of the baseline vehicle.

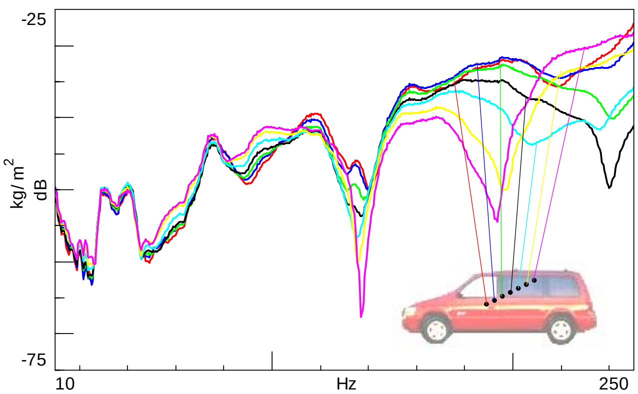


Figure 11. Example of measurements along a straight line in the *y*-axis at the center of the baseline vehicle.

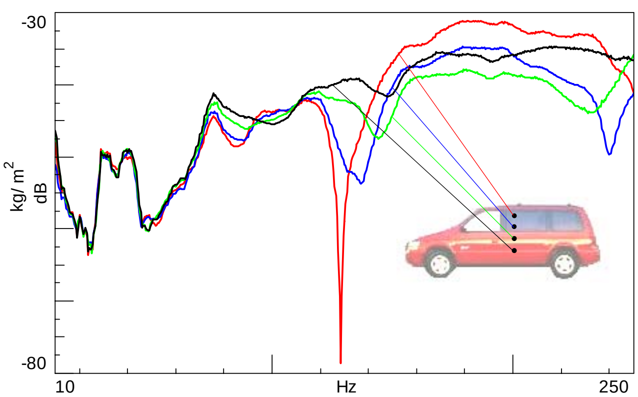


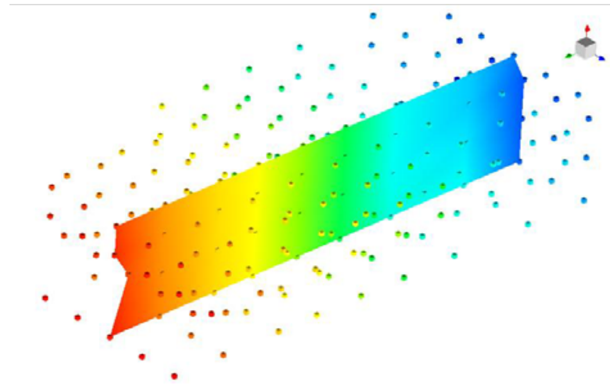
Figure 12. Example of measurements along a straight line in the *z*-axis at the center of the baseline vehicle.

RESULTS, SEATS REMOVED

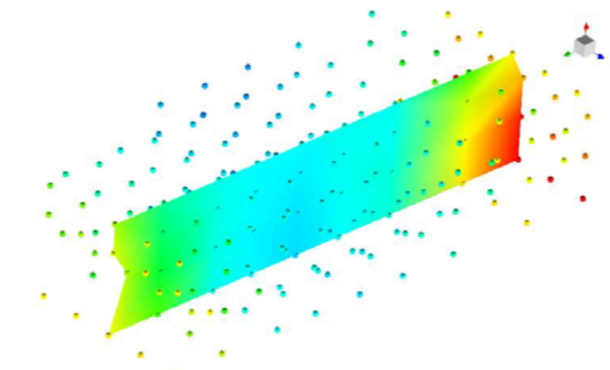
The initial testing was carried out with the three rows of seats removed from the vehicle. The first eight modes are shown in Figure 13. Each measurement point is color coded at one instant in time.

This condition was chosen because it results in a simpler CAE model, since the seats did not have to be modeled, and as it turned out, see Figure 14, the first ten measured natural frequencies agreed rather well with those calculated for a theoretical rectangular box.

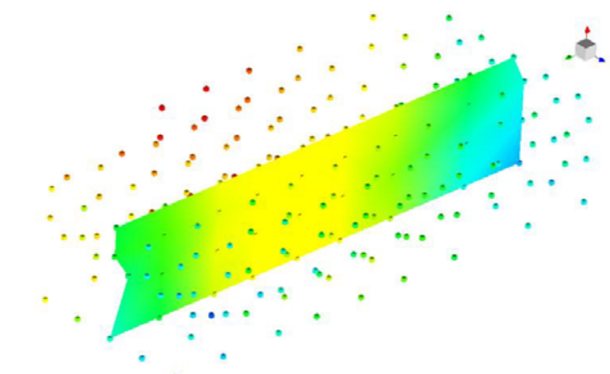
There is reasonable agreement between the fist few theoretical and measured mode shapes, especially the first one. However, the boundary conditions in the vehicle are not rigid and the shape of the cavity is both irregular and non-symmetric and the mode shapes quickly become difficult to discriminate.



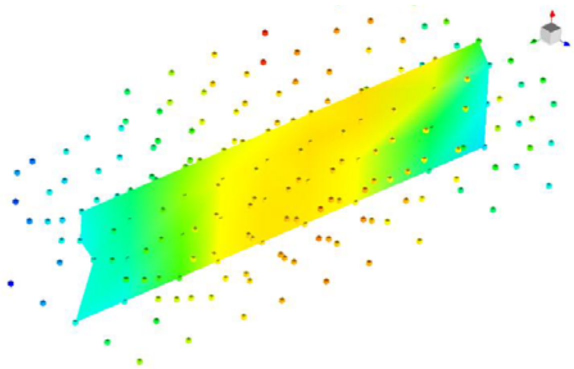
Mode 1, 42.9 Hz, seats removed



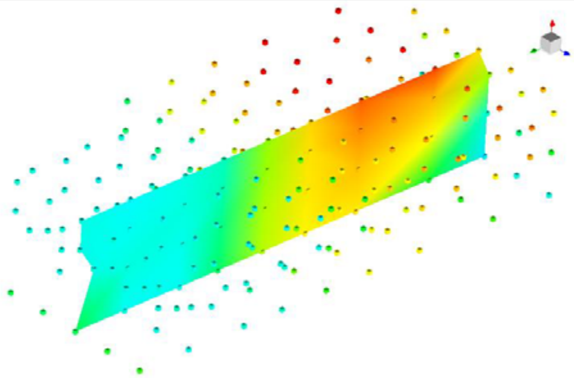
Mode 2, 85.8 Hz, seats removed



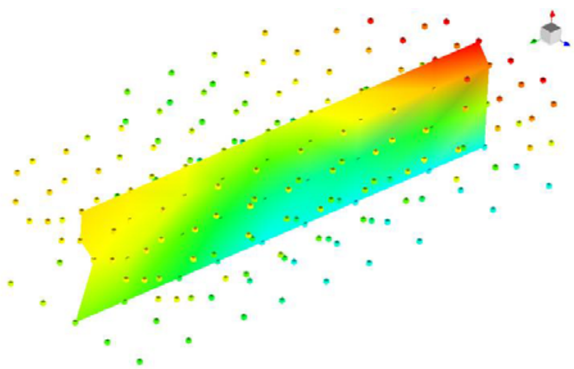
Mode 3, 101.4 Hz, seats removed



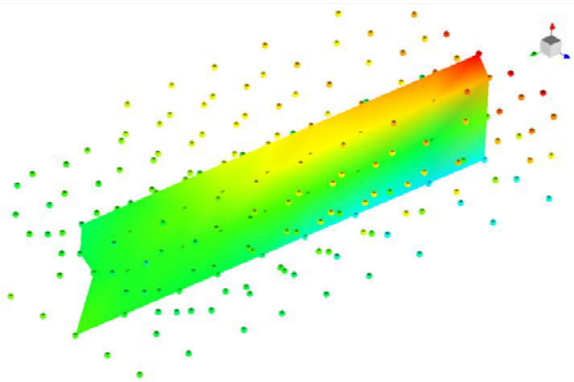
Mode 4, 103.0 Hz, seats removed



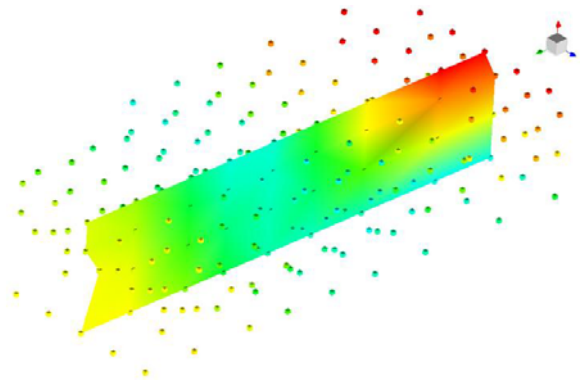
Mode 5, 126.9 Hz, seats removed



Mode 6, 138.5 Hz, seats removed



Mode 7, 144.3 Hz, seats removed



Mode 8, 159.5 Hz, seats removed

Figure 13. Extracted mode shapes 1 to 8 with the seats removed.

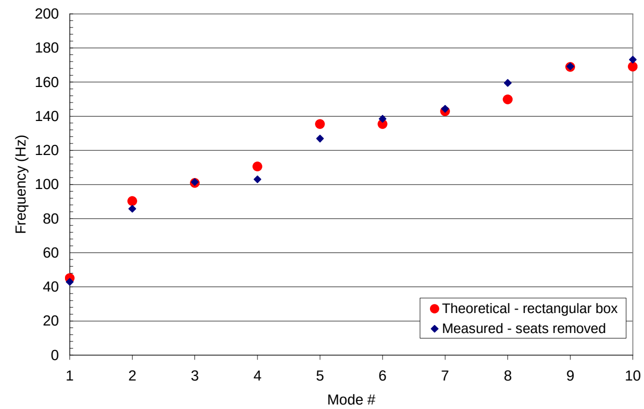


Figure 14. The first 10 natural frequencies of the ‘equivalent’ box compared with measured results with the seats removed.

A summary of the measured and theoretical results are given in Table 1.

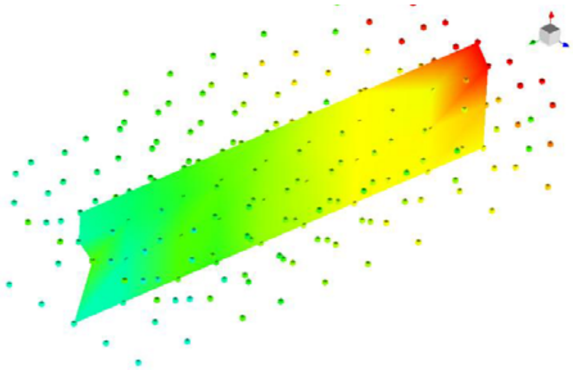
Mode	Index *i*	Index *j*	Index *k*	Theory (Hz)	Test (Hz)	(%)
1	**1**	**0**	**0**	44.2	42.9	6.4
2	**2**	**0**	**0**	88.4	85.8	5.7
3	**0**	**1**	**0**	98.8	101.4	2.4
4	**1**	**1**	**0**	108.3	103.0	2.5
5	**2**	**1**	**0**	132.6	126.9	5.6
6	**3**	**0**	**0**	132.6	138.5	4.8
7	**0**	**0**	**1**	140.0	144.3	1.5
8	**1**	**0**	**1**	146.8	159.5	4.5

Table 1. Comparison of measured and theoretical natural frequencies, measured damping for the case with the seats removed.

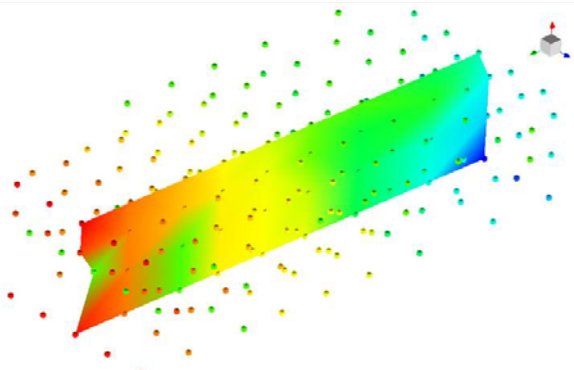
RESULTS, WITH SEATS

The next test was carried out with the three rows of seats in the vehicle or baseline conditions. The first eight modes are shown in Figure 15.

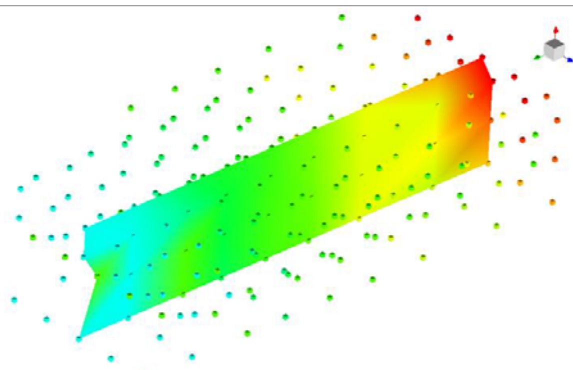
The first three calculated modes are somewhat close to each other in frequency and are grouped together in Table 2. Note also that now there is rather poor agreement between the natural frequencies with those of a simple rectangular box.



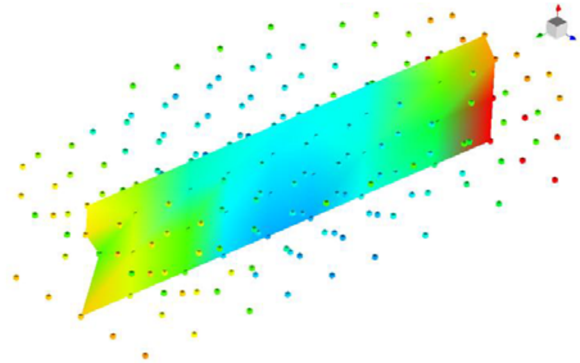
Mode 1, 38.6 Hz, baseline



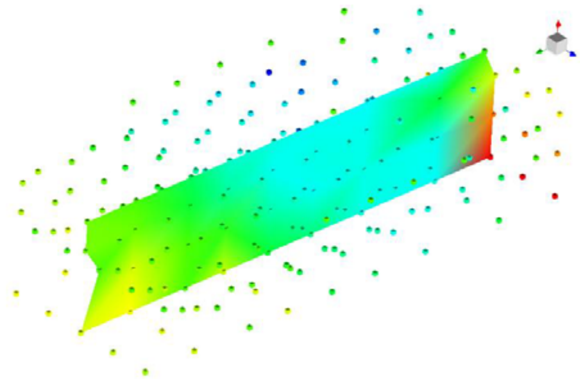
Mode 2, 40.7 Hz, baseline



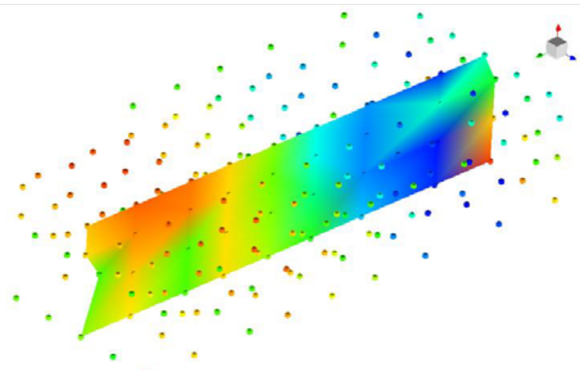
Mode 3, 49.4 Hz, baseline



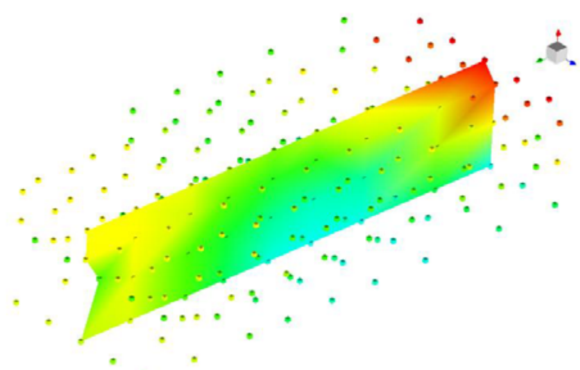
Mode 4, 74.9 Hz, baseline



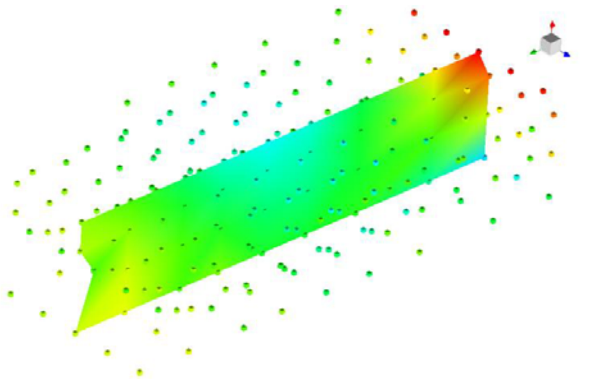
Mode 5, 94.2 Hz, baseline



Mode 6, 118.5 Hz, baseline



Mode 7, 128.2 Hz, baseline



Mode 8, 146.1 Hz, baseline

Figure 15. Extracted mode shapes 1 to 8 with the seats intact (baseline).

Mode	Index *i*	Index *j*	Index *k*	Theory (Hz)	Test (Hz)	(%)
1	1	0	0	44.2	38.6 40.7 49.4	8.7 2.9 2.4
2	2	0	0	88.4	74.9	6.2
3	0	1	0	98.8	94.2	6.2
4	1	1	0	108.3	118.5	6.1
5	2	1	0	132.6	128.2	6.3
6	3	0	0	132.6	146.1	3.0
7	0	0	1	140.0	156.1	3.1
8	1	0	1	146.8	168.0	3.4

Table 2. Comparison of measured and theoretical natural frequencies, measured damping for the case with the seats in the vehicle.

EFFECT OF SEATS

It was found that the seats significantly changed the acoustic frequencies and mode shapes. This was also seen by (Kim, 1999) who also noted that the various impedance boundary conditions had little effect. Their advice is to calculate the acoustic modes with rigid boundaries and then reflect the absorbent effect on the pressure level by using measured damping factors.

Furthermore, when the interior cavity happens to be strongly coupled to a panel, measured results can be quite different from those predicted with rigid boundaries.

In Figure 16 is shown a comparison of two measurements with and without the seats. The results are somewhat counter intuitive since some of the natural frequencies actually decrease when the interior volume of the cavity also decreased with the addition of the seats. The cause of this can be attributed to the absorption or damping of the seats which can be quite significant compared to the other absorptive surfaces (Sanderson, 2002) and to changes of the sound field which affects the propagation path.

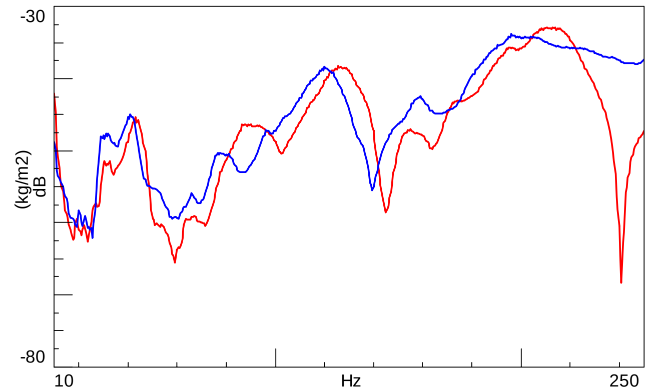


Figure 16. Measurements of P/a at approximately the driver's head position, (—) without seats and (—) with seats (baseline).

CONCLUSION

It was found that the addition of seats into the vehicle interior cavity has two major effects:

Lowers the modal frequencies because the acoustic cabin becomes more elaborate increasing the effective length and changing the mode shapes

Adds volume-absorption which dissipates acoustic energy and as a result lowers the sound pressure amplitudes.

The acoustic modes of the minivan interior cavity are heavily damped 4-8%. This damping is mainly due to the seats, then the carpet, headliner and to a lesser degree the instrument panel and hard trim. This damping needs to be included in CAE models in order to predict the correct natural frequencies and mode shapes.

The flexibility of the panels surrounding the cavity (roof, floor, door, trim, etc…) and their coupling to the acoustic modes play a role in softening the cavity-walls hence making it seem acoustically longer or larger than physical dimensions.

The first four modes of the baseline minivan are longitudinal; i.e. along the *x* axis. These modes are identified in mode shape animations. When the seats are removed, only the first two modes are longitudinal, the third mode is now in the *y* axis, between the floor and roof. It is therefore important to model the effects of the seats correctly.

In future testing, more sources (MIMO) will be used in order to resolve repeated modes or closely spaced modes.

REFERENCES

1. R. D. Blevins, “Fluid Systems,” in *Formulas for Natural Frequency and Mode Shape*, Van Nostrand Reinhold, New York, 1984, Chapter 13.
2. K. Wyckaert, H. Van der Auweraer and W. Hendricx, “Correlation of Acoustical Modal Analysis with Operational data for Road Noise Problems”, *Third International Congress on Air- and Structure-Borne Sound and Vibration*, Montreal, June 13-15, 931-940, 1994.
3. S. H. Kim, J. M. Lee and M. H. Sung, “Structural-Acoustic Modal Coupling Analysis and Application to Noise Reduction in a Vehicle Passenger Compartment”, *Journal of Sound and Vibration*, **225**(5), 989-999, 1999.
4. M. A. Sanderson, G. Ebbitt, P. Deacon, S. Wang and R. Woodcock, “Performance of Perforated Leather Seats: SEA Predictions and Validation”, *The 2002 International Congress and Exhibition on Noise Control Engineering*, Dearborn, August 19-21.