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Noise and vibration simulation using SEA:
Material Database construction technologies

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ABSTRACT

The rapid development and Time to market reduction of today design requires more and more simulation procedures including Acoustic and Vibration behaviour of the project. Advanced market fields in simulation are the aerospace and the automotive, but practically in any of the other machinery sectors, design and simulation should proceed in parallel at the early stage and before any prototyping.

SEA (Statistical Energy Analysis) is a simulation methodology widely used in automotive application but, as any other simulation method, requires correct input parameters in order to give out trustable results. Materials data for Acoustic and Vibration application are usually hard to find in bibliography and experiments shall follow different standards (ISO, ASTM, DIN, BS).

Each parameter is measured with an independent technologies and with different equipment, facing the problem of data format and consistency in the Data base construction. A single and unique system it has been developed in an integrated environment, capable of performing all necessary measurements and fill in the Database. Nowadays a good design simulation must be based on the knowledge of material characteristics.

INTRODUCTION

The SEA (Statistical Energy Analysis) approach to Noise and Vibration in automotive has gained more and more relevance in today design due to its simplified approach and fast calculation, compared to more traditional FEM-BEM techniques which, however, are still preferred for particular situation strictly related to body dynamics.

While correlation techniques have to be used with FEM-BEM methods, between vibrations and sounds, to identify sources and transmission paths, noise reduction techniques can be checked out with the possible results, with SEA approach it is used instead an energy transfer calculation to identify the contribution of each sources and propagation paths efficiency.

Both in FEM-BEM or SEA modeling one of the key point it is to introduce material properties in the model. These information can be achieved by experimental Methodologies to determine for instance: Acoustic Absorption, Impedance, Flow Resistivity, Tortuosity, Damping Loss Factor, Transmission Loss, etc. All relevant data have to be kept in a Data Base ready for exporting in the Model simulation software

Since all experimental procedure follows different standards and scientific approach, some data manipulation and reduction becomes necessary.

In this paper we will review some experimental methodologies for determination of the material properties and an application example to verification of an SEA model of a car.

MATERIAL PROPERTIES

A car is a complex system in which several type of materials play in turns several roles. From the noise and vibration point of view, we can consider the following table:

Isotropic solid	*(Structure Materials elastic same properties in all directions):* steel, glass, etc.
Orthotropic solid	*(Structure Materials elastic different properties vs. directions):* composite
properties:	*Density, Young's Modulus, Poisson's ratio, Damping Loss factor*
Damping Materials	*(un-elastic):* septum, rubbers, EPDM, etc.
Mass materials	*(un-elastic):* septum, rubbers, EPDM, etc.
properties:	*Density, Insertion Loss, Transmission Loss*
Spring like Materials:	
	Felts, Foam
properties:	*Acoustic Absorption and Impedance, Porosity, Flow Resistance, Tortuosity, Complex Modulus, Bulk Loss Factor*

Some of the above properties are well known while some others shall be defined within the Noise and Vibration context. The are the following:

Bulk Loss Factor
the complex part of the bulk elastic or Young's modulus of the porous material, when measured in a vacuum. It is a unit less quantity that describes the percent of total energy that is lost to heat in the process of propagation.
Defined as:

$$\eta = \frac{\Pi_{lost}}{\omega E}$$

P is the power lost to heat
E is the total energy of the material

Impedance of acoustic materials (plane waves)
Specific Acoustic Impedance (at x distance from a sample surface) is the ratio between pressure *p(x,w)* to the normal component of fluid velocity *u_n(x,w)*
Defined as:

$$Z = \frac{\hat{p}(x,\omega)}{\hat{u}_n(x,\omega)}$$

Porosity:
the density fraction of the porous material that is comprised of fluid, *r* is the mass density.
Defined as:

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$$h = 1 - \frac{\rho_{solid}}{\rho_{porous}}$$

Flow Resistivity: a measure of the resistance to fluid flowing through the porous material, Dp is the static pressure differential across a layer of thickness Dx, and n is the velocity of flow through the material
Defined as:
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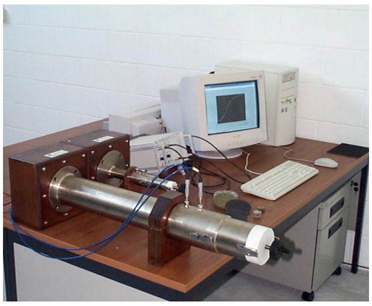
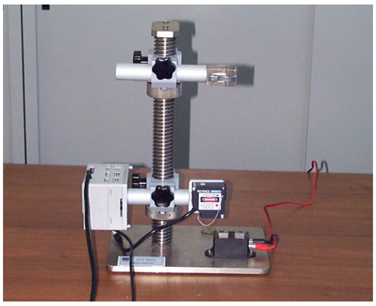
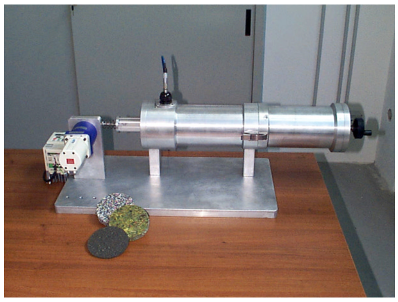
$$R = \frac{1}{v}\frac{\Delta_p}{\Delta_x}$$



Acoustic absorption and Impedance device SCS 902A / 9020



Oberst device SCS 902A / 9021
*** SAE device SCS 902A / 9022**



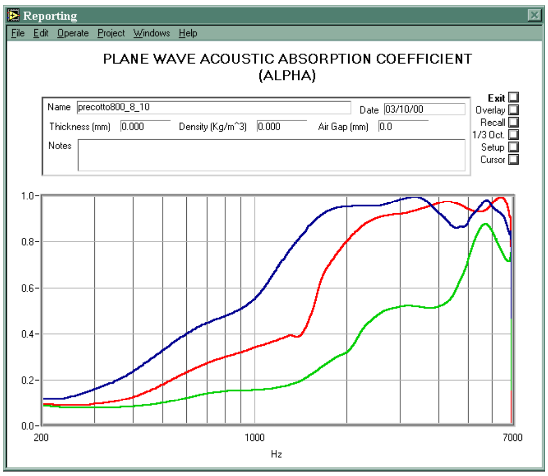
Flow resistivity device SCS 902A / 9021

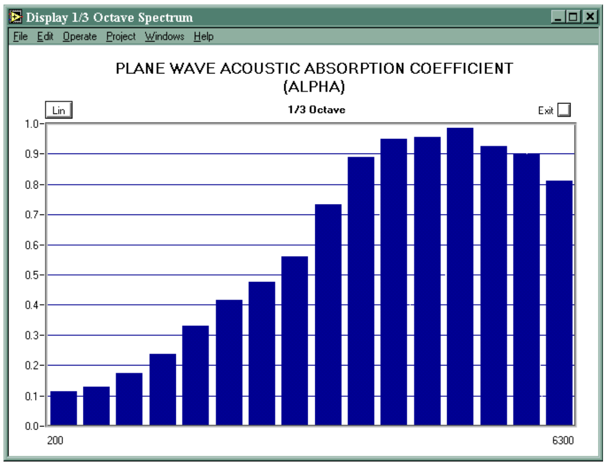
DATA NORMALIZATION

Material properties are experimentally determined using different technologies and standards. In order to make different data consistence among a common data base, some rules shall be followed. First consideration refer to Frequency resolution for data acquisition. For instance: Damping, Acoustic insulation and Absorption are all based on FFT measurements with different frequency resolution and in this case it is possible to use a data normalization through the PSD) Power Spectral Density).

$$[G_{aa}]_{psd} = \frac{G_{aa}}{\Delta_f}$$

PSD describes how Power of time series is distributed with frequency. Practically is the squared Fourier transform, i.e. auto-power spectrum *Gaa*, divided by frequency resolution in Hz.
It can be applied to spectra related to: *Insulation IL or TL spectra, Damping Loss factor, Acoustic Absorption, Impedance, Complex Modulus, Bulk Loss Factor*
Another point concerns the calculation within SEA models. The user can actually choose between constant and percentage bandwidths, i.e. FFT or 1/3 octaves, but 1/3 octaves is the preferred approach for noise prediction.





As a consequence FFT spectra must be converted in 1/3 octaves ones using some mathematical procedures.
Where Energy average is an issue, like Acoustic insulation spectra as IL or TL, FFT lines from the PSD spectrum are added together within a Frequency interval (F1 – F2) corresponding to the given 1/3 octave filter. If we define:

F = Frequency range
N = FFT lines
Df = Frequency resolution
Ns = minimum number of lines required for 1/3 octave filter
F*i* = Frequency of *i-th* data sample

The minimum 1/3 octave band is given by: $(N_s + 1) \cdot \frac{F}{N}$

Example: F = 20 kHz, N = 3720, Ns = 3 / 1/3 octave band width = 21.5 Hz is the "100 Hz" 1/3 octave filter.

Transfer function derived FFT's are not necessarly concerned with Energy averaging, as it is the case of an Acoustic absorption coefficient measured according to ASTM standards. In this case only the energy values of FFT lines corresponding to 1/3 octave center bands are taken in account.

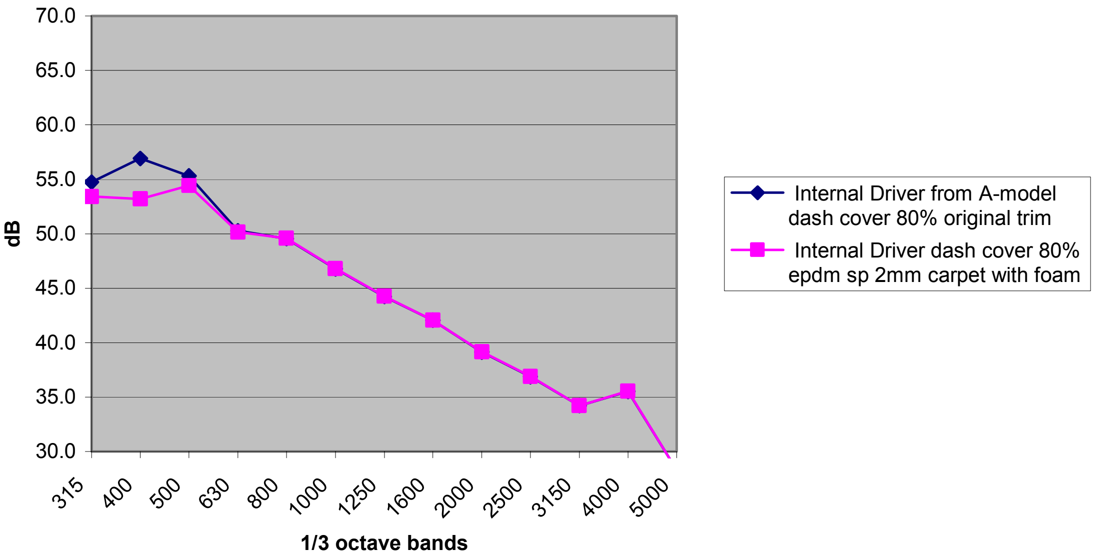
SEA MODEL VERIFICATION

The A-model was developed for airborne sound and the check up of model sensitivity has started from the verification of the carpet trim part, originally designed with a textile element between the carpet itself and the floor. This choice is due to a feeling taken from tests that the vehicle has some structure born problems not covered in the airborne acoustic model.

<u>The carpet trim</u>

In order to check the sensitivity of the model to the carpet trim, we have inserted a foam layer instead of the textile material, covering the bottom floor, actually we were able to obtain an interesting result as plotted in **Fig. 1**.

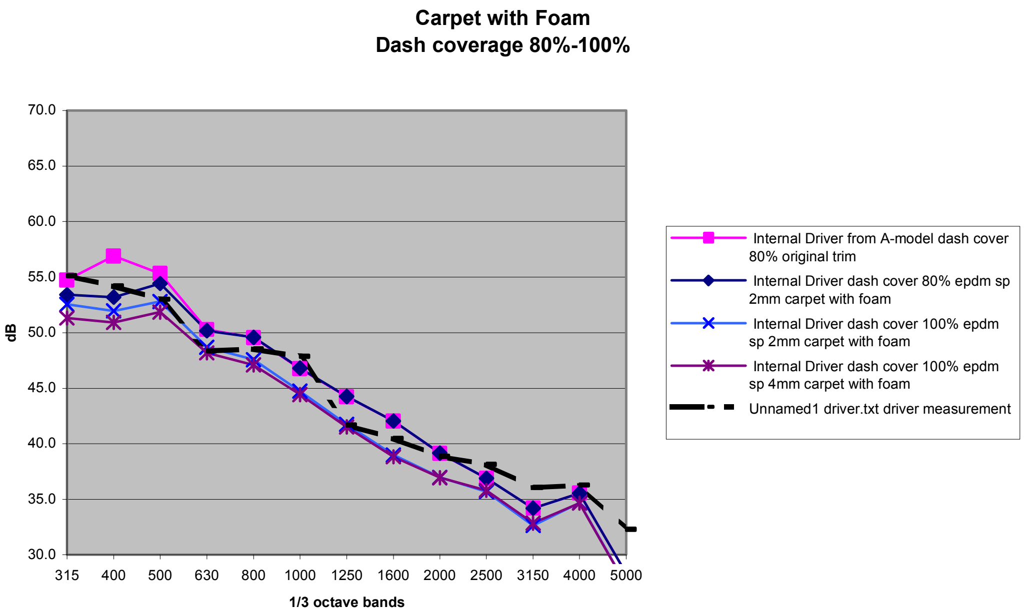
FFT results (left) and 1/3 octaves conversion (right) for the Acoustic absorption coefficient in the SCS 902A device



As you can see there is an interesting effect in the first 2 low frequency bands due to a sort of friction effect provided by the foam over the floor plate.

A further run with a double mass on the carpet did not show any significant effects, and the analysis of the power inputs to the internal driver cavity has given information about the predominance of energy coming from the dash element.

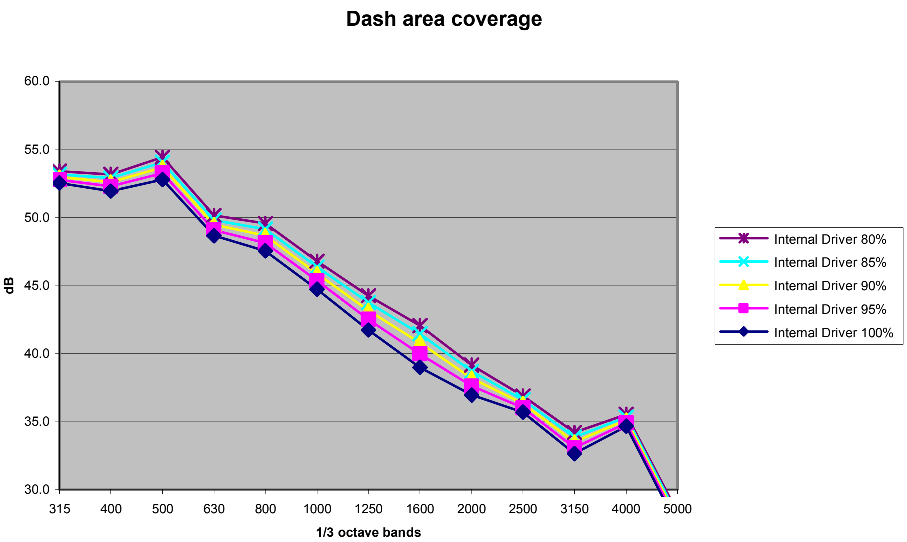
But also doubling the dash material mass did not improve the results, so the only explanation has to be reconnected to the % of area covered by the dash trim.



The Frontal area and Dash trim coverage

By extending the trim area coverage of the dash from 80% to 100% we can see a significant variations and also the effect of doubling the carpet mass it now gives a difference in the low frequency range.

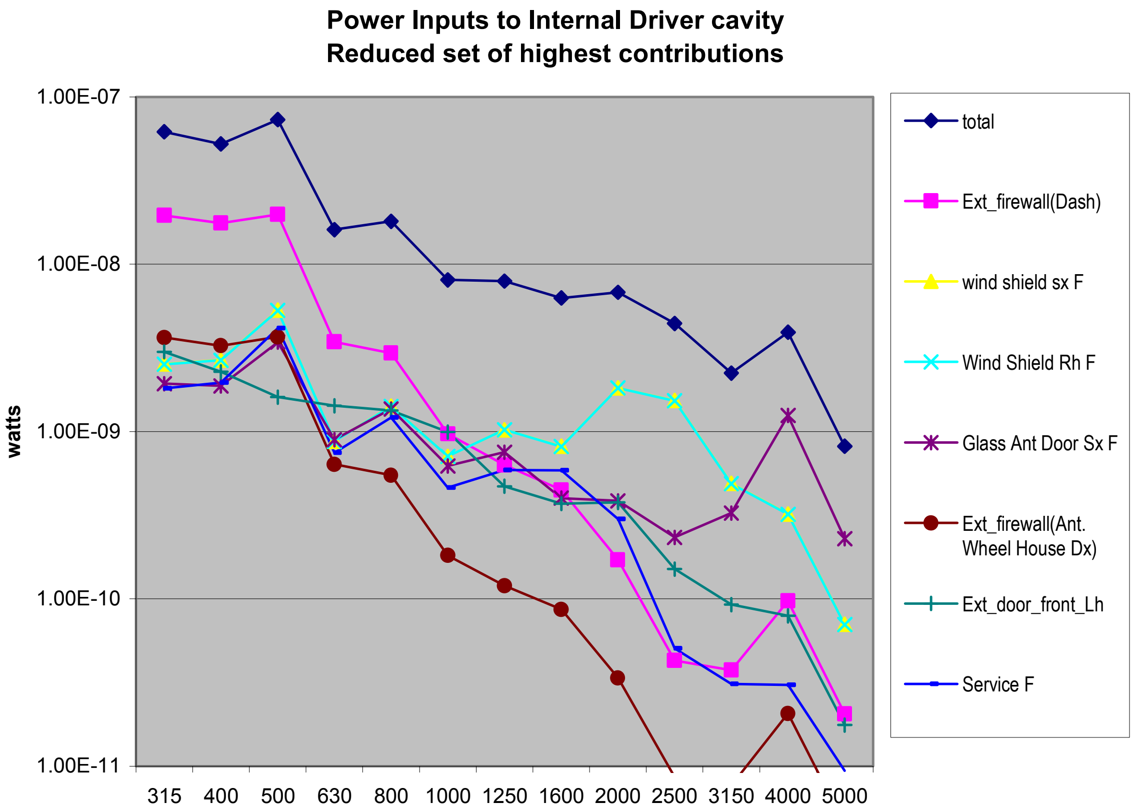
It is clear the effect of the % of area coverage by the dash trim. If we now make a short study on the influence of the area coverage by the dash trim part we will be able to plot the following results showing the big effects due to the accuracy (tolerances) which are necessary to produce the trim parts, around the cable passage, the steering column, etc.



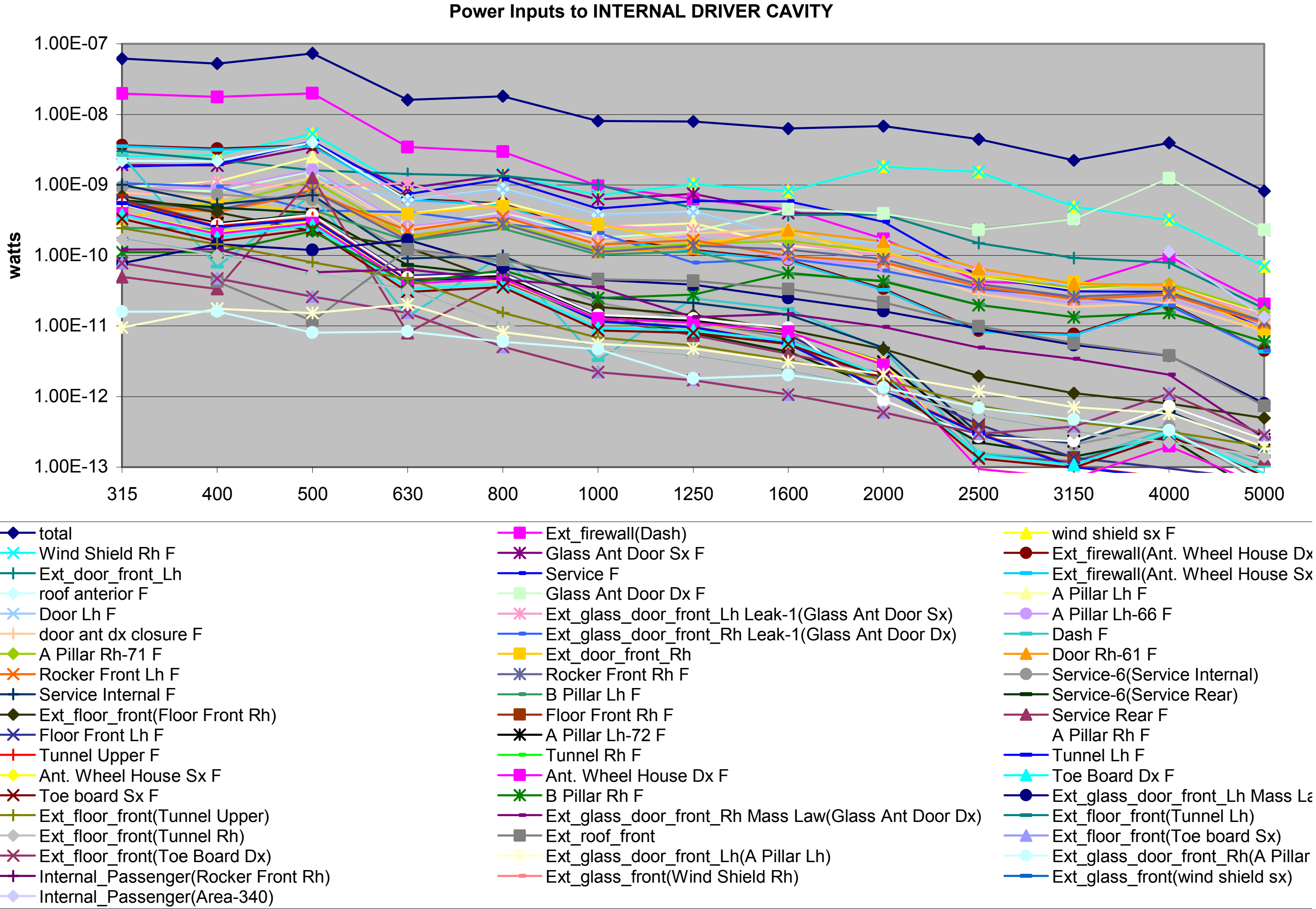
New trim configuration

Once we established now the new configuration with the foam-carpet solution and the dash area covered by 100%, we can check the power inputs in the internal driver cavity and see that now the energy contribution in different part of the spectrum are due to unexpected car elements, in order of magnitude, they are the windshield and the lateral windows, as you can see on the following figure of a reduced ensemble of power inputs to the internal driver cabin.

The most important contribution are pointed out in the following figure, resulting from the model run, showing “all” contribution to the SPL inside the cabin.



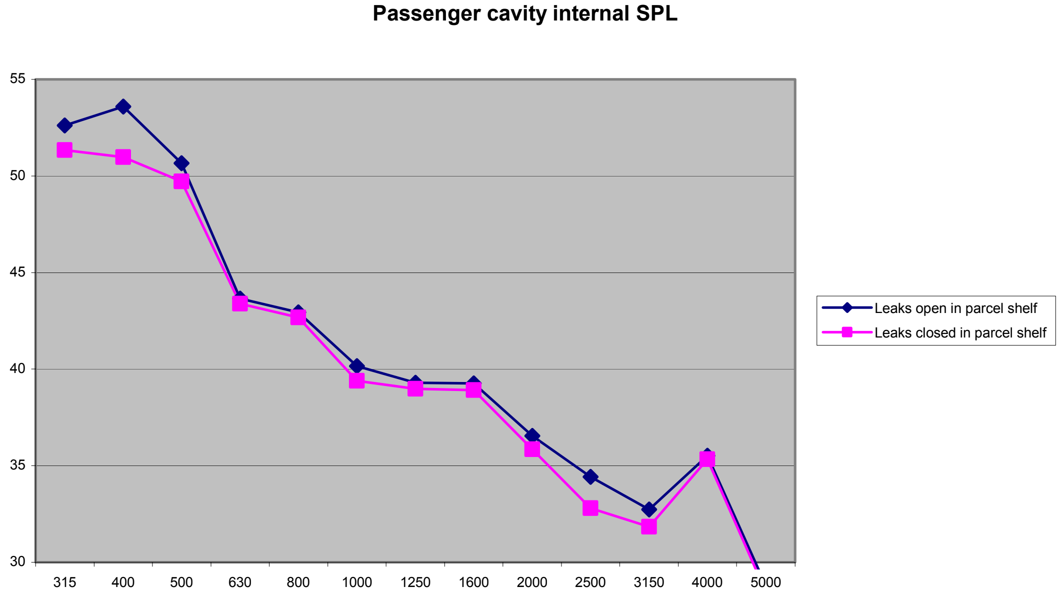
The general ensemble of Power Inputs to the internal driver cavity are plotted in the figure on next page.


Power Inputs to INTERNAL DRIVER CAVITY
watts
1.00E-07
1.00E-08
1.00E-09
1.00E-10
1.00E-11
1.00E-12
1.00E-13
315
400
500
630
800
1000
1250
1600
2000
2500
3150
4000
5000
total
Ext_firewall(Dash)
wind shield sx F
Wind Shield Rh F
Glass Ant Door Sx F
Ext_firewall(Ant. Wheel House Dx
Ext_door_front_Lh
Service F
Ext_firewall(Ant. Wheel House Sx
roof anterior F
Glass Ant Door Dx F
A Pillar Lh F
Door Lh F
Ext_glass_door_front_Lh Leak-1(Glass Ant Door Sx)
A Pillar Lh-66 F
door ant dx closure F
Ext_glass_door_front_Rh Leak-1(Glass Ant Door Dx)
Dash F
A Pillar Rh-71 F
Ext_door_front_Rh
Door Rh-61 F
Rocker Front Lh F
Rocker Front Rh F
Service-6(Service Internal)
Service Internal F
B Pillar Lh F
Service-6(Service Rear)
Ext_floor_front(Floor Front Rh)
Floor Front Rh F
Service Rear F
Floor Front Lh F
A Pillar Lh-72 F
A Pillar Rh F
Tunnel Upper F
Tunnel Rh F
Tunnel Lh F
Ant. Wheel House Sx F
Ant. Wheel House Dx F
Toe Board Dx F
Toe board Sx F
B Pillar Rh F
Ext_glass_door_front_Lh Mass La
Ext_floor_front(Tunnel Upper)
Ext_glass_door_front_Rh Mass Law(Glass Ant Door Dx)
Ext_floor_front(Tunnel Lh)
Ext_floor_front(Tunnel Rh)
Ext_roof_front
Ext_floor_front(Toe board Sx)
Ext_floor_front(Toe Board Dx)
Ext_glass_door_front_Lh(A Pillar Lh)
Ext_glass_door_front_Rh(A Pillar
Internal_Passenger(Rocker Front Rh)
Ext_glass_front(Wind Shield Rh)
Ext_glass_front(wind shield sx)
Internal_Passenger(Area-340)

The Rear area and the Parcel shelf

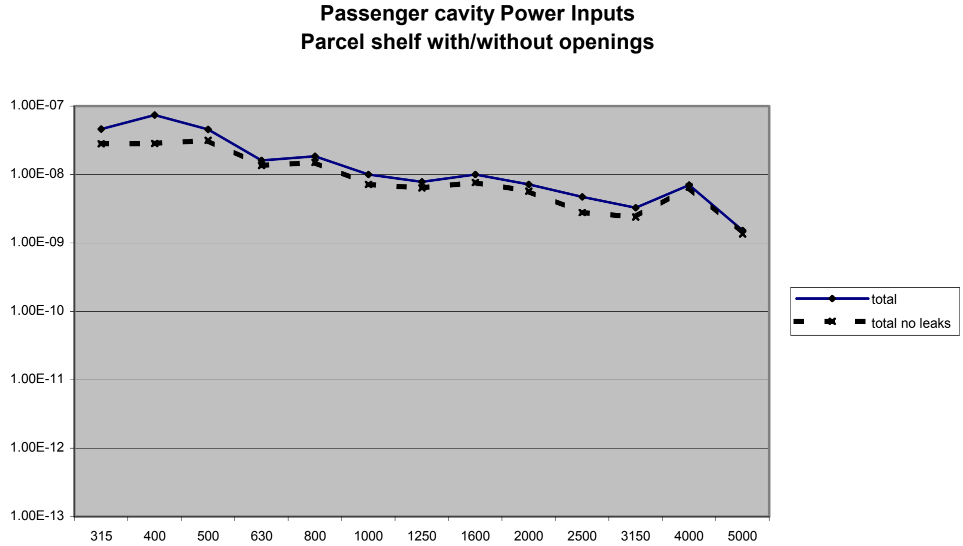
In order to evaluate the noise in the internal cavity due to the contributions coming from the car rear, we have divided the internal cavity in 2 acoustic cavities subsystem: the driver and the passenger.

The A-.model runs shows the influence of the opening in the Parcel Shelf as well as contributions from the Under seat and the ext windows. Actually we made different runs of the A-model and the results are showed in the graphs below.

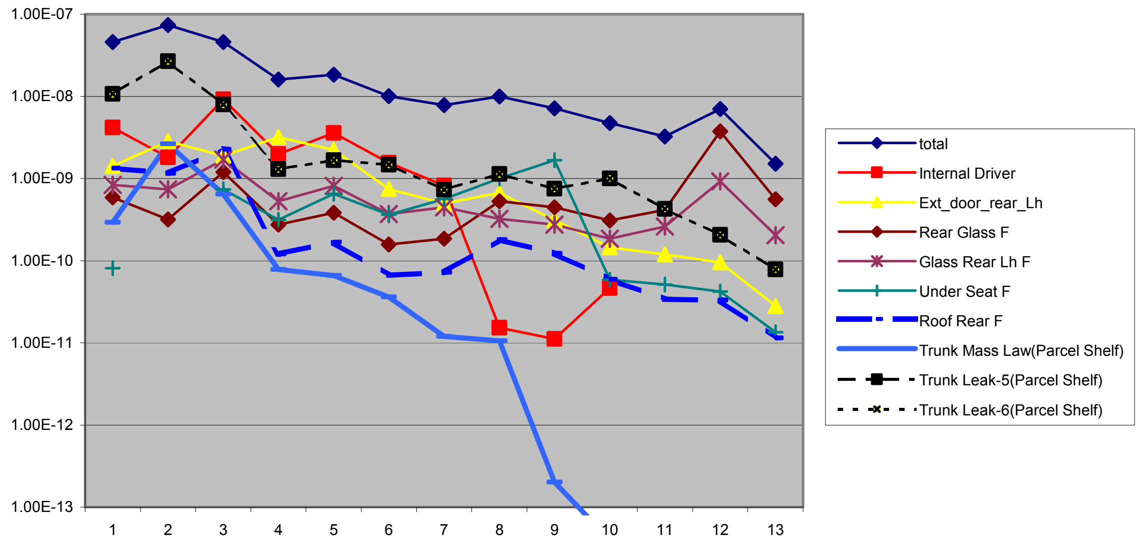


The present simulation is concerning the airborne effect, in reality the effect of the opening in the Parcel shelf with the car in running conditions, gives a major effect. The power inputs to the passenger cavity with and without openings in the Parcel shelf are reported in the following graphs.

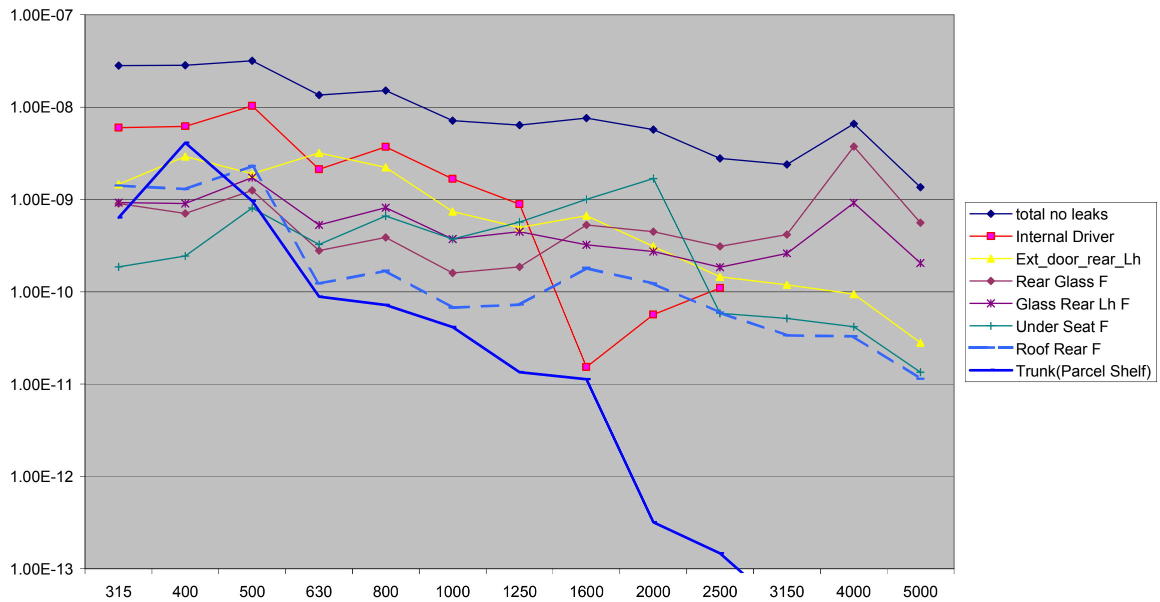
The difference between the Total acoustic power input in the passenger cavity is quite similar to the difference reported above between the corresponding Sound Pressure Levels resulting.



Power inputs to Passenger cavity
Parcel shelf with Loudspeakers openings



Power inputs to Passenger cavity
Parcel shelf with no openings



By taking some actions to avoid openings in the Parcel shelf, as one can see from the graphs above, the noise level in the passenger compartment is mainly due to the acoustic energy coming from the driver cavity. However the critical openings in the Parcel shelf are the loudspeakers, due to the size of them, but even in case you closed them, than you will have the rising of the effect of the ventilation openings.

That is why we consider that both type of openings must be taken in consideration in the design of some trim parts between the Parcel shelf and the trunk.

Other power inputs are coming from the rear glass and from the Under seat. The first one cannot be changed, while for the Under seat we have tried to add some damping material on it and than we run another time the model to see what happen.

Results are on the following graph were one can see that the model is also sensitive to this new improvement.

Power inputs to Passenger cavity
Parcel shelf with no openings