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Prediction of sound transmission through automotive door seal systems

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In automotive industry the door seal systems are an important contributor to vehicle interior noise in the middle and high frequency range. The aim of the study was to develop a numerical model in order to predict the sound transmission loss through elastomeric seals. At the early stage of the development process, this type of numerical tool is very interesting to investigate the influence of the design parameters of the seal.

Two steps were necessary: a static analysis to calculate the seal shape after compression (door closure event), an acoustic analysis based on dynamic parameters to determine the sound transmission. Finite element methods were used for both steps (commercial softwares). These two steps were validated experimentally for two types of geometry, different compression ratios and loading cases.

One original contribution of the paper concerns the determination of a valid non-linear model for the static part and of a good approximation of the dynamic behavior of the elastomer's Young modulus.
Finally, a sensitivity analysis was performed in order to evaluate the influence of the design parameters of the door seal system such as the compression ratio or the dynamic parameters.

1 Introduction

The necessity of a permanent increase of the acoustic quality of vehicles leads the car manufacturers to develop their knowledge on the acoustic behaviour of most of the parts. It is well known that the elastomeric sealing systems play a significant role on the acoustic performance of the vehicle, particularly in the medium and high frequency range considering interior noise.

Elastomeric sealing systems used in automotive industry are numerous. They include weatherstrip of window, glass run seals … The case of study will be the door seal system, and particularly two types of geometry will be studied: a simple one and a more complex one as you can see on figure 1:

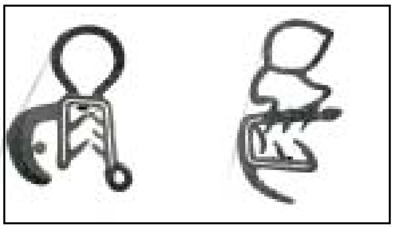

Figure 1: door seal system cross-sections

The aim of the study was to develop a complete numerical model in order to predict the sound transmission loss through the door seal system. This type of model is very interesting at the early stage of the development process. Using this type of predictive tool, you can find the best design to satisfy two requirements for example: door closure force and transmission loss.

The prediction of the transmission loss using classical finite element model has been developed by several studies [1] [2]. The main difficulty concerns the ability to represent correctly the two step of calculation: as a first step, we need to calculate the seal shape after compression using the appropriate visco elastic parameters (non linear behaviour of the material taken into account); as a second step, a fully coupled fluid / structure calculation has to be used in order to obtain the transmission loss of the door seal system.

The methodology that has been developed is summarised on figure 2:

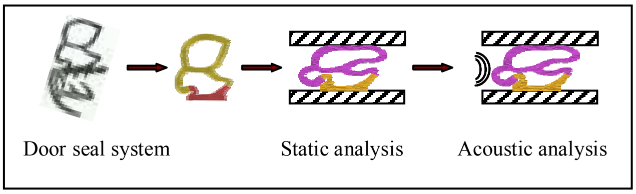

Figure 2: developed methodology

One original contribution of the paper concerns the determination of a valid non-linear model for the static part and of a good approximation of the dynamic behaviour of the elastomer's Young modulus.

In the first part, the static approach will be considered: non-linear model using the commercial software called Abaqus, experimental validation and sensitivity analysis for two types of geometry, different compression ratio and loading cases. In the second part, the acoustic approach will be described: fully coupled model using the commercial software called Actran, experimental validation in a coupled room and sensitivity analysis (geometry, material characteristics …).

2 Static analysis

The prediction of the static behaviour of the seal was determined by a classical finite element calculation (using the commercial software Abaqus).

2.1 Mechanical behaviour

First the mechanical behaviour of the elastomeric part was identified by uniaxial measurement conditions as we can see on figure 3. This type of data was then used to calculate the seal shape after compression. A specific non linear hyperelastic law was considered in the model.

Figure 3: uniaxial test

2.2 Seal shape

Then the calculation of the seal shape was compared to the measurements for two types of geometry, different compression ratios and loading cases.

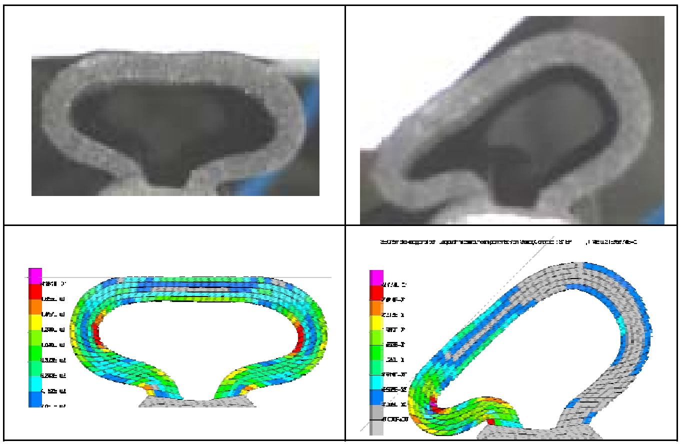

Figure 4: simple geometry, 2 loading cases
(Up: experimental; down: calculated)

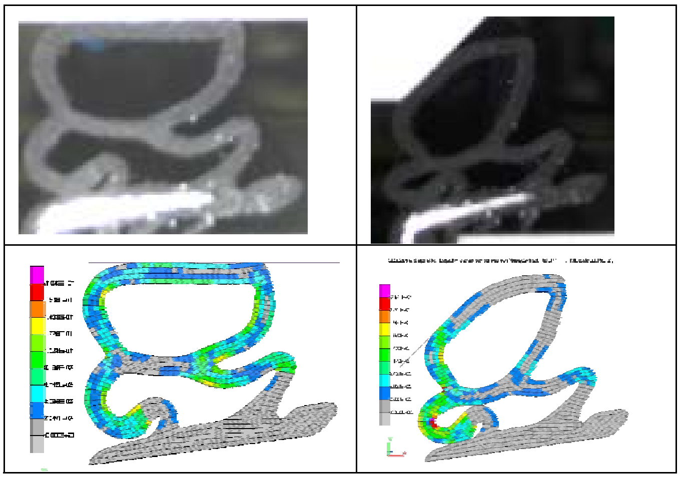

Figure 5: complex geometry, 2 loading cases
(Up: experimental; down: calculated)

As we can see on figure 4 and 5, the seal shape after compression for different geometry and different loading cases are very similar. The prediction of seal shape are very good.

But it's also interesting to have a look at the compression force to validate completely the hyper elastic law. That is the aim of the section 2.3.

2.3 Compression force

Then the calculation of the seal compression force was compared to the measurements for two types of geometry, different compression ratios and loading cases.

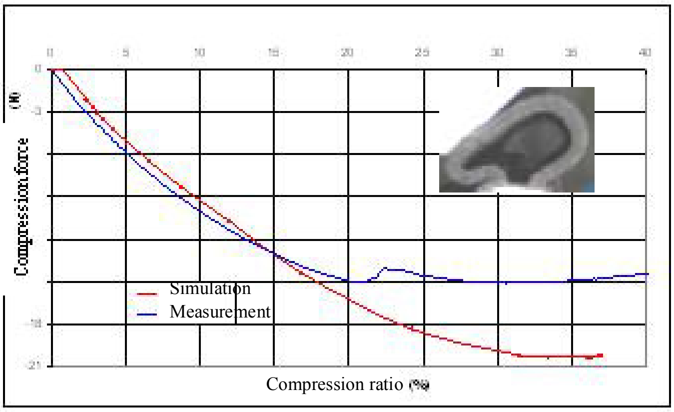

Figure 6: Compression force version compression ratio for the simple geometry

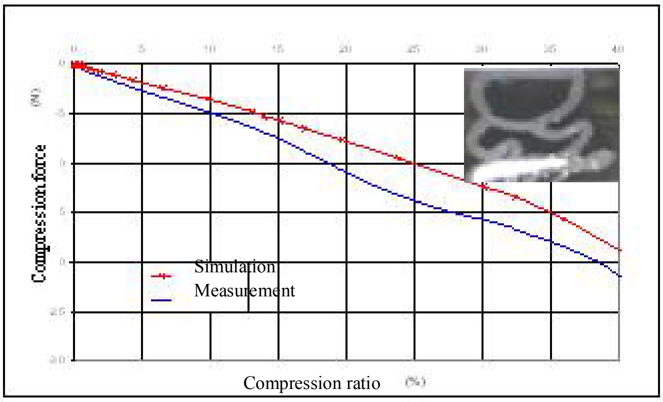

Figure 7: Compression force versus compression ratio for the complex geometry

As we can see on picture 6 and 7, the comparisons between the calculated and the measured compression force show that the hyper elastic model is accurate enough for our application. The differences that we can mention for the simple geometry for compression ratio upper to 20 % are due to experimental condition testing: slipping conditions that were not reproduced numerically.

Therefore, the static analysis was performed using classical finite element formulation with a non linear material model. It has been shown that numerical results are very promising. As the seal shape was computed after compression, the second step of the methodology was developed in order to predict the acoustic transmission of the door seal system.

3 Acoustic analysis

3.1 Experiments

ELASTOMER YOUNG MODULUS:

The storage and loss modulus of the seal were measured using a dynamic mechanical analyzer from 0 to 5 000 Hz as we can see on figure 8.

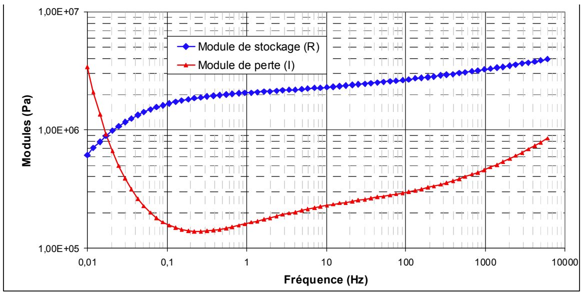

Figure 8: Storage and loss modulus versus frequency

TRANSMISSION LOSS:

The sound transmission loss was measured in a coupled room (hemi anechoic + reverberant room) in CTTM Le Mans. A specific experimental setup was designed to modify the seal compression ratio when the elastomer is located in the bay. We used acoustic intensity measurements to characterize the transmission loss. We can see the experimental setup and the coupled room on figure 9:

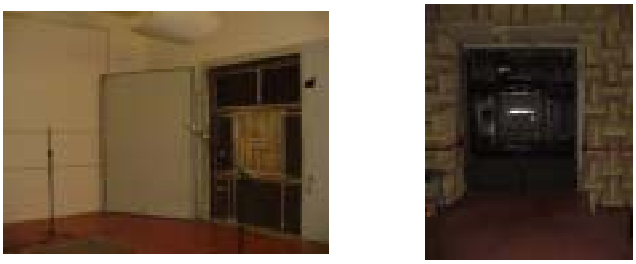

Figure 9: reverberant (left) and hemi-anechoic room (right)

The sound field in the reverberant room was considered to be diffuse. So, it was possible to use a simple formulation to calculate the incident acoustic power W_{inc} :

$$W_{inc} = \frac{S\,p^2_{moy}}{4\rho\,c} \quad (1)$$

where p_{moy} is the mean sound pressure (4 microphones), ρ the density of air (1.21 kg/m3), S the seal surface on the reverberant side and c the speed of sound in air (343 m/s).

Then the radiated power is simply calculated by the following formulation :

$$W_{rad} = SI_t \quad (2)$$

Where S is the seal radiating surface and I_t the measured intensity on the hemi-anechoic side.

The ratio of these values give us the transmission loss TL:

$$\tau = \frac{W_{rad}}{W_{inc}} \quad (3) \qquad\qquad TL = 10\log\frac{1}{\tau} \quad (4)$$

3.2 Numerical analysis

The commercial software ACTRAN was used to predict the fluid / structure interaction. This software is based on a finite / infinite element formulation.

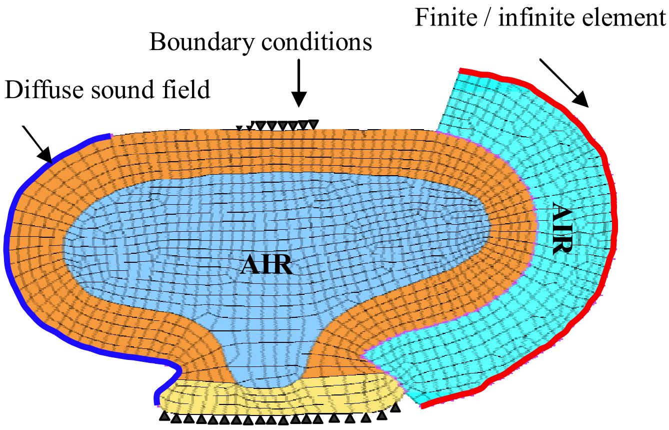

Figure 10: Actran model

A 2D model was built in order to predict the sound transmission loss as we can see on figure 10 for the simple geometry.

Infinite elements are located at the boundary of the air mesh to model an infinite environment (respecting Sommerfeld radiating conditions). These elements present a sufficient order to have a relevant modeling of the infinite environment. A diffuse field is created on the reverberant side. Appropriate boundary conditions are applied at the contact surfaces between the seal and the experimental setup.

Then this fully coupled model allows calculating the radiated acoustic power considering a diffuse sound field excitation on the reverberant side. We can mention an original contribution of the paper : the elastomer's Young modulus frequency dependence was precisely measured as mentioned in part 3.1 and it was taken into account for the simulation. Our study was limited to the following frequency range from 500 to 4 000 Hz. The results will be presented in third octave.

RESULTS:

First the results obtained by numerical simulations and the measurements were compared for two types of geometry and different compression ratio.

As an example, one can see the transmission loss comparisons for a simple and a more complex door seal with a compression ratio of 40 % on figure 11 and 12.

The sound transmission loss reaches minimum values between third octave 1 250 Hz and 2 000 Hz for the simple seal. These minimum values cover the third octave 1 250 and 1 600 Hz for the second geometry.

The general trend of the frequency dependence of the TL is very well represented. Some discrepancies can appear between numerical and experimental data where the sound transmission loss is underestimated by the numerical simulations. The discrepancies never exceed 5 dB, which is an acceptable value.

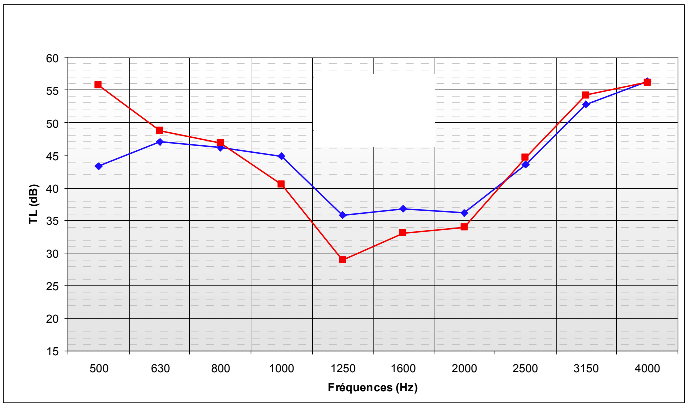

Figure 11: Transmission loss versus frequency – simple seal (: measurements, : calculation)

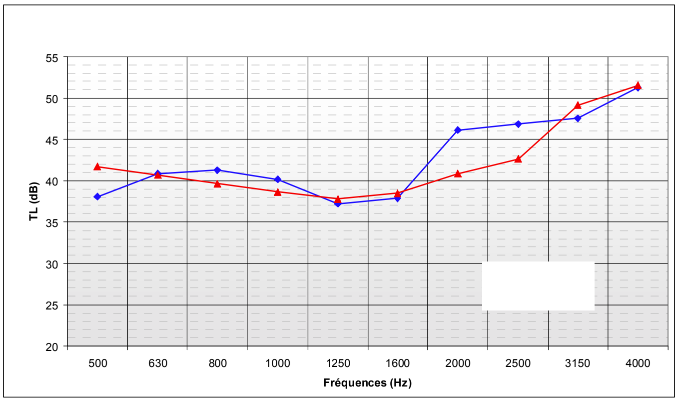

Figure 12: Transmission loss versus frequency – complex seal (: measurements, : calculation)

3.3 Sensitivity analysis

One main advantage of the numerical model is to be able to investigate the effects of several design parameters on the door seal sound transmission loss. Some of them will be described here:

- Compression ratio effect
- Material properties of the seal (damping and Young modulus)
- Seal geometry

COMPRESSION RATIO:

Figure 13 shows the effects of seal compression ratio (10 to 40%) on the transmission loss of the simple seal geometry.

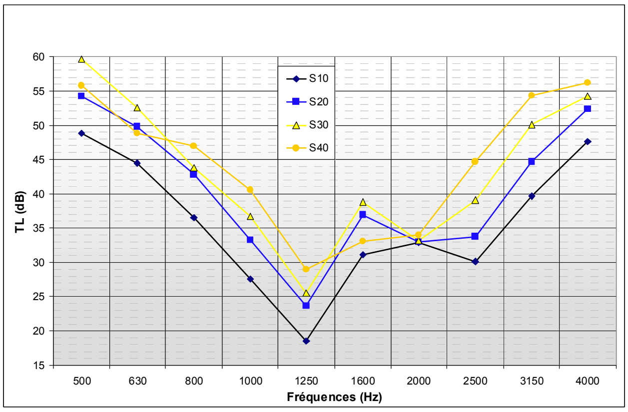

Figure 13: Transmission loss versus frequency – Compression ratio effect for a simple geometry

It can be seen that the increasing of the compression ratio will give higher transmission loss in the whole frequency range. The minimum TL value is reached at the same third octave frequency for the 4 configurations. This is due to the average effect of the third octave representation: the frequency of the resonance is actually increasing when the compression ratio becomes higher.

MATERIAL PROPERTIES:

Figure 14 and 15 show the effects of seal damping and Young modulus on the transmission loss of the simple seal geometry (40 % compression ratio):

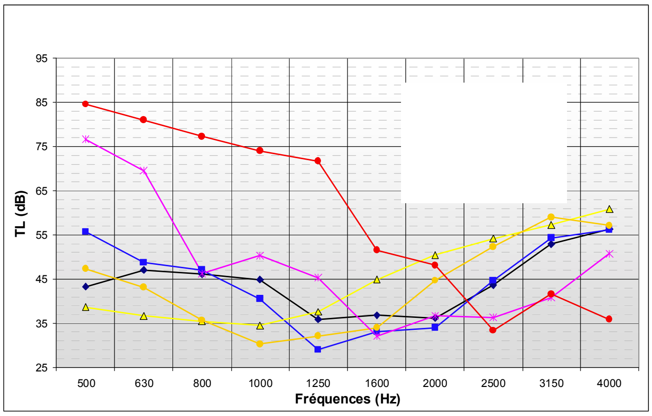

Figure 14: Transmission loss versus frequency – Young modulus effect (initial values, E*0.1, E*0.5, E*2, E*10)

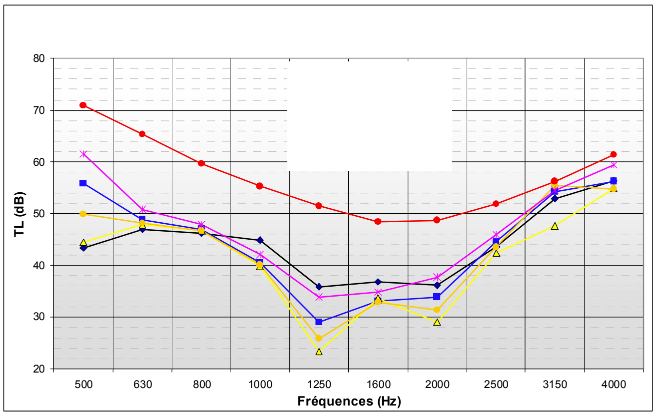

Figure 15: Transmission loss versus frequency – Damping effect (initial values, η*0.1, η *0.5, η *2, η *10)

The storage modulus has a strong influence on the transmission loss. We can observe important differences in its evolution depending on frequency. When the stiffness of the seal increases, then the resonance region will move to higher frequencies, the transmission loss will decrease in the low frequency range but increase in the high frequency range.

Damping also has a strong influence on the global evolution of the transmission loss of the seal and more specifically in the resonance region. A wrong evaluation of this parameter could lead to bad values of transmission loss in the resonance region.

Material properties measurements of the seal should be done as precisely as possible in order to avoid strong errors in the transmission loss prediction.

SEAL GEOMETRY:

Figure 16 shows the effects of seal geometry on the transmission loss (simple and complex seal geometry under 2 types of loading cases, 30% compression ratio):

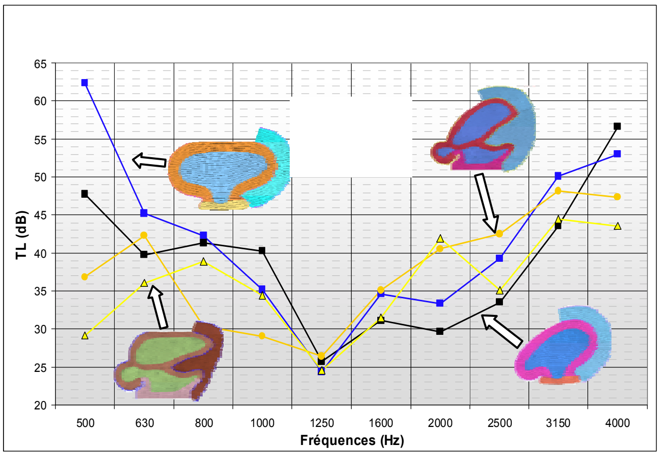

Figure 16: Transmission loss versus frequency – Compression ratio effect for a simple geometry

The material properties are the same for both geometry in order to determine the influence of the geometry. We can notice that the evolution of the transmission loss strongly depends on the geometry: differences reaching more than 5 dB in some frequency range.

Therefore, the shape of the cross section has a relatively important influence on the door seal transmission loss. Only the resonance region doesn't seem to be affected by the geometry because it stays constant in frequency and in TL value whatever the geometry type. But this is due to the average effect of the third octave representation.

4 Conclusion

As a first step, the static approach was considered: a non-linear model was used and results were compared to experimental data. A sensitivity analysis for two types of geometry, different compression ratio and loading cases was established.

As a second step, the acoustic approach was described: a fully coupled model using a commercial software was used. The comparisons between the experiments and the calculations showed very good results. Then a sensitivity analysis was performed to consider the effect of the seal geometry, the material properties and the compression ratio.

It was shown that this type of numerical methodology can be used in automotive industry in order to determine the most appropriate seal design considering sound transmission loss. The analysis of the closure force compared to the acoustic performance is another useful application at the early stage of the development process.

References

[1] Y. Gur, K. N. Morman, "Sound transmission analysis of vehicle door sealing system", *Proceedings of the 1999 SAE Noise & Vibration Conference*, *Traverse City, Michigan, Paper No. 1999-01-1804*, 1187-1196 (1999)

[2] J. Park, T. Siegmund and L. G. Mongeau, "Sound Transmission Through Elastomeric Sealing Systems", *Proceedings of the 2001 SAE Noise & Vibration Conference, Paper No. 2001-01-1411* (2001)