Nonlinear Numerical Analysis of Insertion Loss for Structures Having a Compressed Softened Porous Layer Sandwiched by Double Walls under Acoustic Excitation

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Abstract. We perform nonlinear numerical simulation of insertion loss for structures having compressed porous layers sandwiched by double walls under acoustic excitation. The double walls are consisted of a steel base plate and a plastic cover plate. We compare the insertion loss including Urethane foam with felt. These porous layers are compressed by a heavy frame around the cover plate. As increasing weight of the frame, Urethane foam has soft-hardening characteristics in its restoring force. If we set appropriate weight of the frame to get 33% compression, lower rigidity is obtained for Urethane foam. This leads to softer porous layer for the double walls. And vibration and sound decrease from the base plate to the cover plate. For felt, the hardening characteristics appear. This leads to the higher rigidity and the more transmission. In the numerical calculation using LS-Dyna as a nonlinear FEM code, geometric nonlinearity is considered. For compressed porous materials, Storaker model is adopted in consideration of material nonlinearity with hysteresis. Sound pressure on the surface of the base plate was computed in a reverberation box where there exists a sound source by our FORTRAN FEM code. Sound radiation powers from the cover plate were also calculated. We evaluated effects of compression on them.

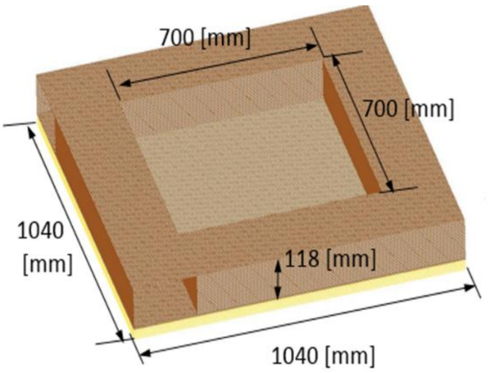
1. Introduction

To reduce undesirable noise and vibration in cabins of cars, sound insulation is usually used under the conditions of lighter weight. This leads to fuel economy for cars in consideration of the reduction of CO2. To decrease the interior noise, porous materials (e.g. fibrous materials or foams) are often sandwiched between car body panels and interior parts (e.g. insulators or carpets or resin parts) as the structure of double walls. These structures help us improve sound insulation performance around floor panels or dash panels and so on [1]-[4]. Many researchers have been studying numerically the structures including porous materials using linear analysis (e.g. [5], [6]). For these insulators, as the parts of double walls, the porous layers are sometimes compressed locally to avoid interference with other parts mounted on the cars. The compressed porous materials have often different materials properties after the compression. In this paper, we carry out nonlinear numerical simulation of insertion loss for structures having compressed porous layers sandwiched by double walls under acoustic excitation. The double walls are consisted of a steel base plate and a plastic cover plate. We compare the insertion loss between Urethane foam and felt. These porous layers are compressed by a heavy frame around the cover plate.

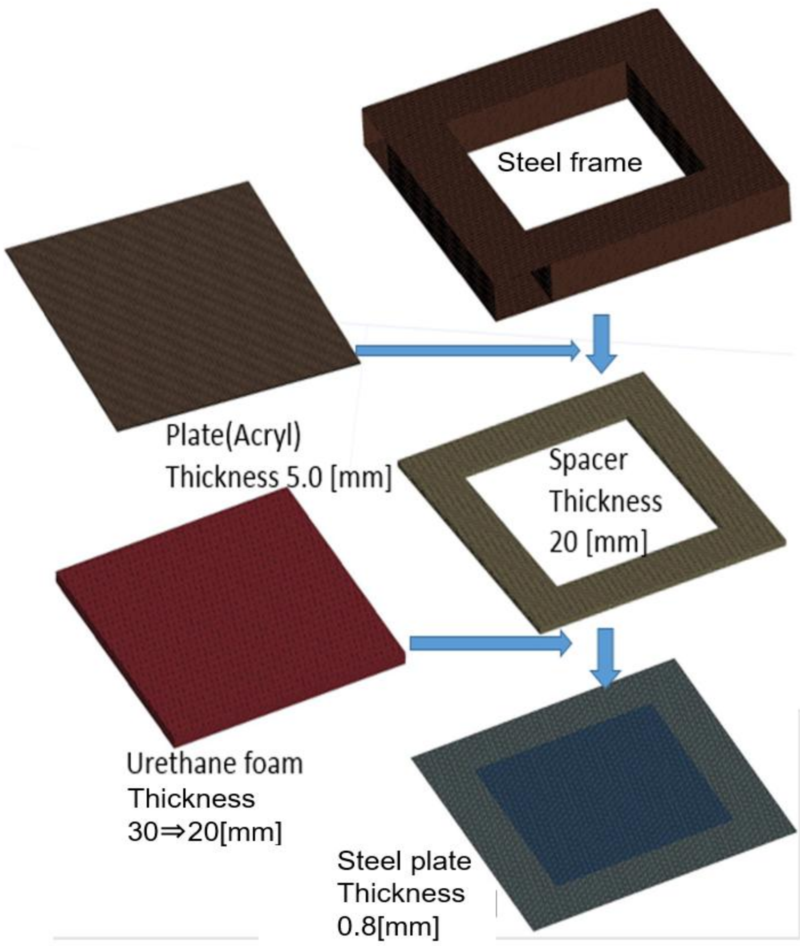
2. Calculation for Insertion Loss of Compressed Double Walls

2.1 Numerical Model and Calculation Flow

Figure 1 shows a FEM model of the double walls having the compressed porous layer (e.g. Urethane foam). Figure1(a) is the overall view of the double walls with the heavy steel frame, while Fig.1(b) is the parts diagram.



(a) Overall view of double walls with frame



(b) Parts diagram

Fig. 1. FEM model for nonlinear vibration analysis for evaluation of sound insulation after compression of porous layer in double walls.

In the double walls, the bottom base plate is the steel plate and the upper cover plate is the Acrylic plate. The porous layer is sandwiched by these plates.

The size of the steel bottom plate and the upper cover plate are same as the outside size of the heavy steel frame. The outside size of the hard rubber spacer is also same as outside size of the frame. The inner size of the spacer is same as inner size of the frame. The size of the porous layer is also same as inner size of the frame. For the porous layers, we used a typical soft urethane foam and a felt around automotive body panels as sound insulators. The mass density of the Urethane foam before compression is 65[kg/m^3]. And the mass density of the felt before compression is 68[kg/m^3].

As shown in Fig.2, the heavy steel frame is set on the upper cover plate. Due to the heavy weight of this frame, the porous layer is compressed until the thickness of the porous layer is equal to the thickness of the hard rubber spacer.

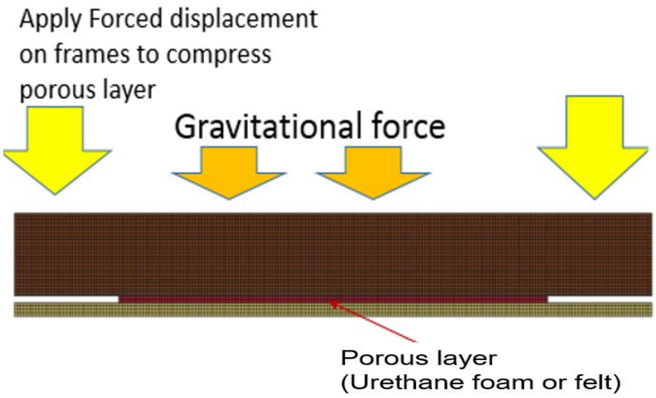


Fig. 2. FEM model for nonlinear vibration analysis for evaluation of sound insulation after compression of porous layer in double walls.

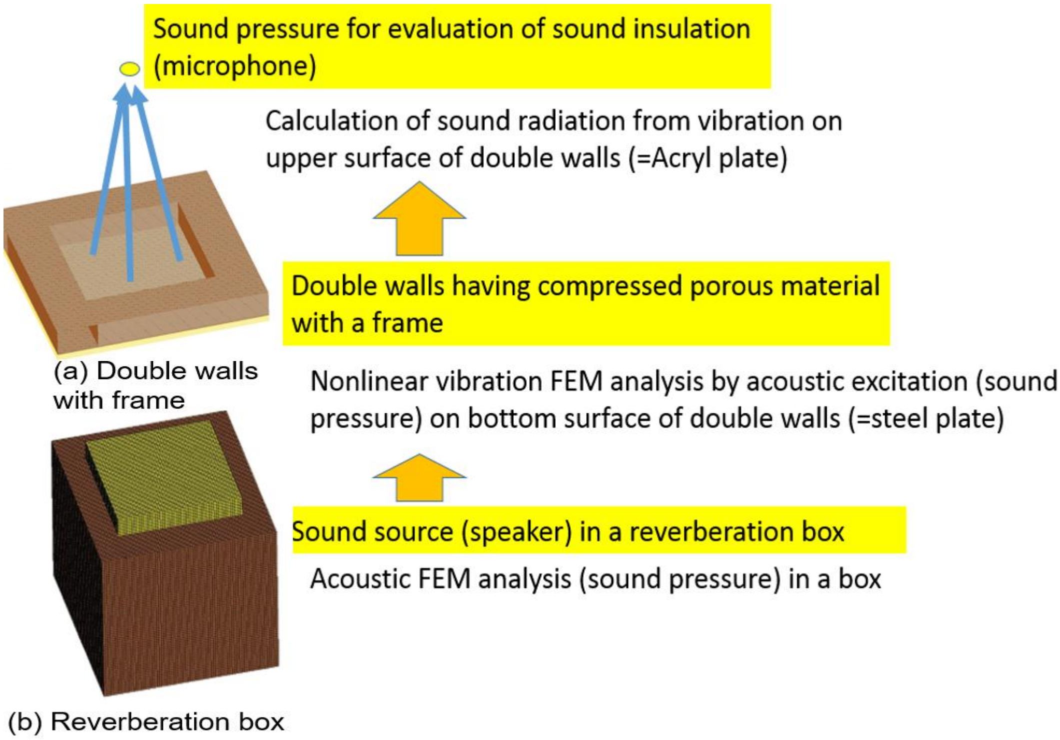


Fig. 3. Outline of calculation flow for evaluation of sound insulation of double walls after compression of porous layer.

In the numerical analysis, we apply the forced displacement on the frame to compress the porous layer. In that time, gravitational force is also loaded on the double walls (see section 2.3 in detail). After that we can prepare the post-compressed porous layer sandwiched by the double walls.

Figure 3 shows the outline of calculation flow. There exists a speaker in a reverberation box as shown in Fig.3(b). As illustrated in Fig.3(a), sound pressure is radiated from the top surface of the double walls having the compressed porous layer. The sound pressure is observed by a microphone. These parts and their setup correspond to an experiment.

From this calculation flow in Fig.3, firstly, acoustic FEM analysis are carried out using a numerical model of the reverberation box. We compute sound pressure on the back surface of the steel bottom plate from a speaker as a sound source in the box (see section 2.4 in detail). Next, we perform nonlinear vibration analysis of the double walls including the post-compressed porous layer. We calculate the vibration velocity on the top surface of the Acryl cover plate under the acoustic excitation (see section 2.5 in detail). Further, we compute sound pressure radiated from the top surface of the cover plate at the position of microphone away from the cover plate (see section 2.6 in detail).

2.2 Check Validity of Material Parameters of the Compressed Porous Layer

As preparation for the later simulation of insertion loss, we perform nonlinear FEM analysis of deformation for Urethane foam under compression using LS-DYNA. We calculate the deformation under compression to check the material nonlinearity in the stress-strain curve using Storakers model [9]. Figure 4 shows the calculation model. For this calculation, the size of the porous layer is 100 [mm]×100 [mm] and the thickness is 10 [mm] before the compression. The bottom of the porous layer is fixed. Distributed compression load is applied on the upper surface of the porous layer.

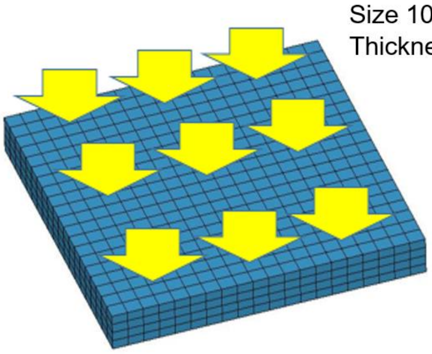


Fig. 4 Calculation model to check material parameters of compressed porous layer

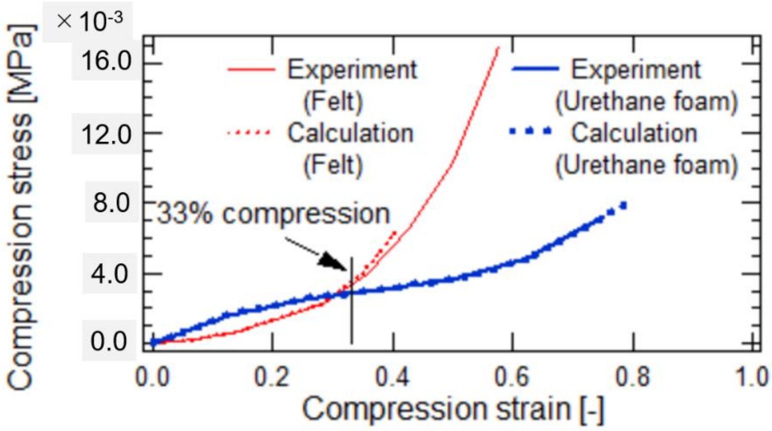
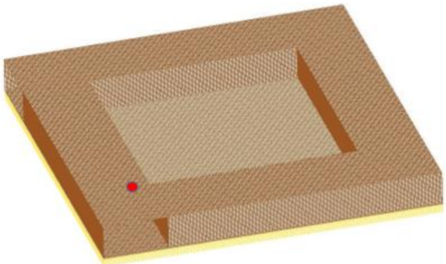


Fig. 5. Comparison between calculated results and experimental results for stress-strain curve of Urethane foam (blue) and felt (red) under compression.

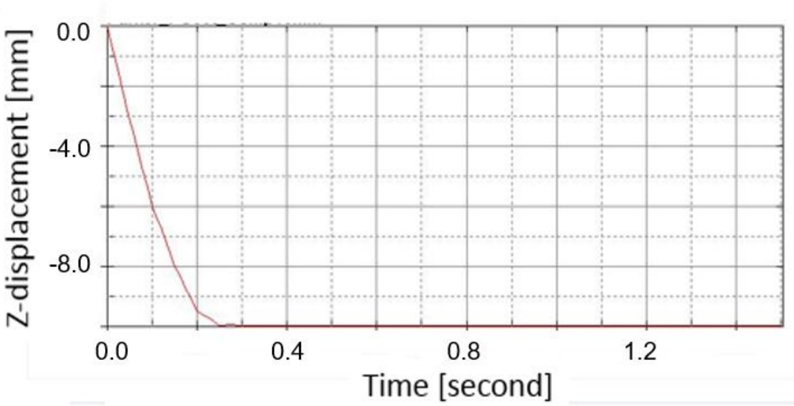
Figure 5 is the computed results of the relation between force and displacement. From this figure, this Urethane foam has soft–hardening characteristics in restoring force. And we can obtain the softer property after about 33% compression. This soft property of the Urethane foam is very useful for the later high insulation performance. Further, from this result, stress-strain curve for the Urethane foam is determined numerically as shown in Fig.5 as blue dotted line. In Fig.5, an experimental stress-strain curve is also written as blue solid line. The calculation result and the experimental result are agreed well. Therefore, we can confirm the material parameters for the Urethane foam are valid. Note that this model includes effects of the inside air and the microscopic resin frame (i.e. frame of cell or fiber) in the porous layer. In Fig. 5, stress-strain curves for a typical felt are also written. Calculation result (red, dotted line) and experimental result (red, solid line) are also agreed well. We can also say the material parameters for the felt are valid. In Fig. 5, after 33% compression, the slope of the Urethane foam is smaller than that of the felt. This means that the compressed Urethane foam is softer than the compressed felt.

2.3 Computation of Compression Property of Urethane Foam Using Nonlinear FEM Analysis

Before calculation for insertion loss of the double walls having the compressed porous layer, the porous layer is compressed numerically by a forced displacement applied on the stiff frame around the double walls. According to Fig. 5, if we set appropriate compression (i.e. 33% compression = thickness reduction from 30 [mm] to 20 [mm]), lower rigidity of the porous layer is obtained for this Urethane foam. This leads to the softer porous layer for the double walls. And we can expect transmission of vibration and sound decrease from the steel base plate to the cover Acrylic plate in the later calculation. As shown in Fig. 2, due to the forced displacement of the frame by moving the frame downward, the porous layer is compressed until the thickness of the porous layer is equal to the thickness (=20 [mm]) of a hard rubber spacer.



(a) Observation point (red marker)



(b) Time history while porous layer is compressed from 30mm to 20mm

Fig. 6. Relation between Z-displacement and time when the Urethane foam is compressed due to the forced displacement of the frame.

In the numerical analysis, we apply the forced displacement on the frame to compress the porous layer as nonlinear FEM compression analysis using LS-DYNA in consideration of material nonlinearity in the porous material and geometric nonlinearity. In that time, gravitational force is also applied on the double walls. Figure 6(b) shows the relation between time and the forced displacement at the specified point in the Fig,6 (a) on the flame. Figure 7 shows the time history of the velocity at this point. In Fig.7, a transient response occurs due to the compression, but the response is converged after 1.2 second. We used the compressed model under convergence for the later computation of the insertion loss.

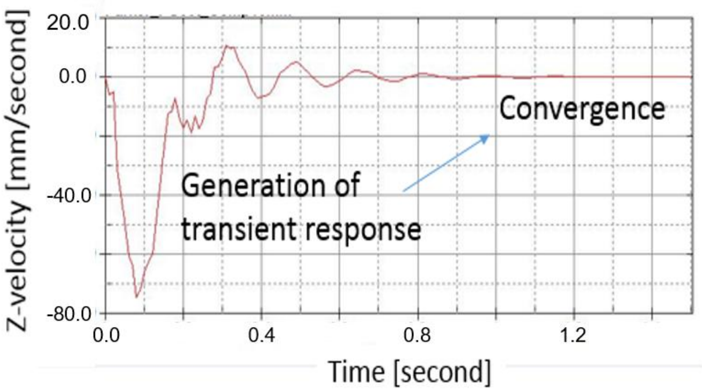


Fig. 7 Relation between Z-velocity and time when the Urethane foam is compressed due to the forced displacement of the frame.

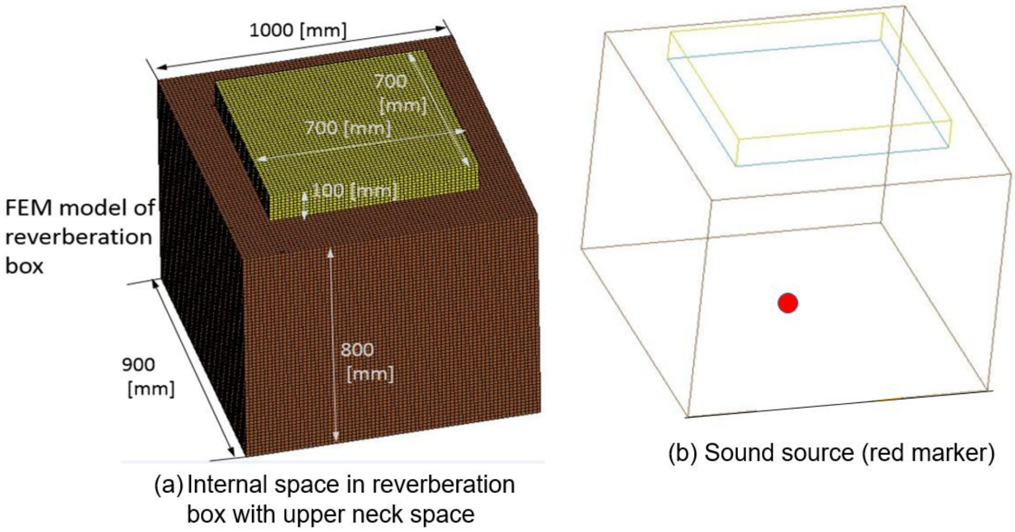
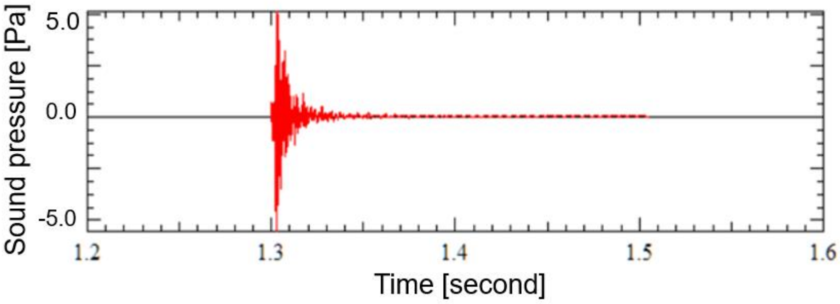


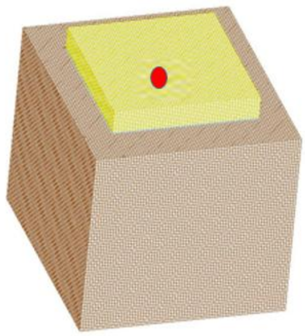
Fig. 8. Calculation model of the reverberation box and excitation points and input pulse for acoustic FEM analysis.

2.4 Acoustic FEM Analysis in Reverberation Box

By following the calculation flow in Fig.3 to evaluate the insertion loss, at first, we carry out acoustic FEM analysis for the internal space in the reverberation box as shown in Fig. 8 (a) using our original FORTRAN code in Gunma University. The internal space has an upper neck space as shown in Fig, 8 (a). Real eigenvalue analysis is carried out. From 0 to 4[kHz] band, more than 8000 eigenmodes exist. Using these modes, an input pulse is applied to the inner space from a speaker as a sound source. The speaker is set at the bottom of the box in Fig. 8 (b). Input amplitude is set to obtain the same sound pressure with the average value of the corresponding experiment at the observation point from 1kHz to 4kHz. Then, time histories of sound pressure are calculated on the back surface of the steel base plate by modal impact response analysis in consideration of modal damping using our MSKE method [7]-[8]. For an example in Fig.9 (a), we obtain a time history of sound pressure at a typical point in Fig. 9 (b). Figure 10 shows the sound pressure level at the point in frequency domain after FFT. The horizontal axis represents the analyzed frequency and the vertical axis is the Fourier spectrum of the sound pressure. From this figure, enough modes are generated in the region higher than 500Hz, and we can regard as diffused sound field. We use the time history of the sound pressure distribution as the acoustic excitation to the double walls having the compressed porous layer.



(a) Time history of sound pressure at the observation point



(b) Observation point (red marker)

Fig. 9. Time history of sound pressure at a node around the bottom surface of the steel basic plate in the double walls.

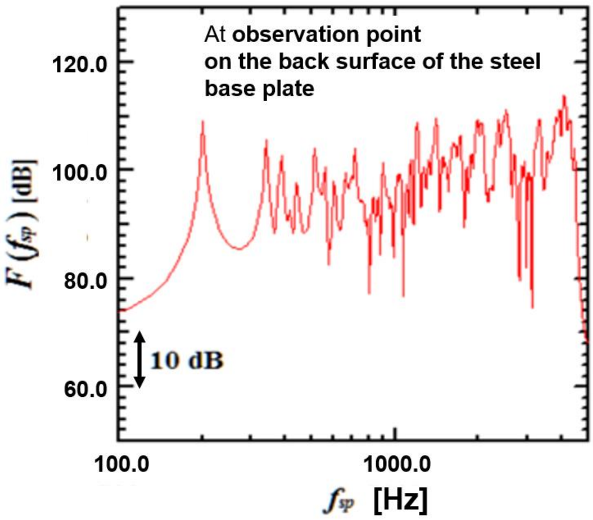


Fig. 10. Frequency response of sound pressure at a typical node around the bottom surface of the steel basic plate in the double walls.

2.5 Nonlinear FEM Vibration Analysis in Time Domain for Double Walls Having Compressed Porous Material

Next, according to the calculation flow in Fig. 3, we compute surface vibration on the upper cover plate (i.e. Acrylic plate) in the double walls having the compressed porous layer under acoustic excitation from the bottom surface of the steel base plate as described in the section 2.4. We use LS-DYNA as nonlinear FEM analysis in consideration of material nonlinearity of porous layer and geometric nonlinearity. The time history of the surface vibration on the upper cover acrylic plate is transformed in frequency domain using FFT to prepare the following computation of the sound radiation analysis in the next section 2.6.

2.6 Radiation Analysis from Surface Vibration of Acrylic Plate in Double Walls Having Compressed Porous Layer

Finally, by following the calculation flow in Fig. 3, we compute sound pressure at the observation point in Fig. 3 from the surface vibration on the upper cover plate (i.e. Acrylic plate) in the double walls having the compressed porous layer as described in section 2.5. The observation point is 0.50 [m] in Z-direction away from the center of the surface in the cover plate. In this calculation, we set an infinite baffle around the Acryl cover plate.

2.7 Evaluation of Insertion Loss of Double Walls Having Compressed Porous Layer

Insertion losses are computed using SPL (i.e. sound pressure level) at the observation point in Fig. 3 as follows.

Insertion Loss ＝SPL of the steel base plate without the porous layer and the acrylic cover plate
－ SPL of the double walls having the compressed porous layer.

Figure 11 represents insertion losses for the Urethane foam and the felt with / without the 33% compression. The blue lines in Fig. 11 show the calculated results of the insertion loss using the Urethane foam. In this figure, the thick blue line shows the insertion loss with the 33% compression of the Urethane foam, and thin blue line shows that without the compression. As you can see in this figure, the insertion loss increases because of the 33% compression for the Urethane foam. After the 33% compression of this Urethane foam, the Urethane foam become softer because of the soft-hardening characteristics (see in Fig.5) in the restoring force due to the material nonlinearity. This leads that the vibration propagation from the steel base plate to the acrylic cover plate decreases for the compressed Urethane foam. Therefore, we obtain less sound radiation from the double walls having the compressed Urethane foam. In other words, we can get the higher sound insulation due to the compression for this Urethane foam.

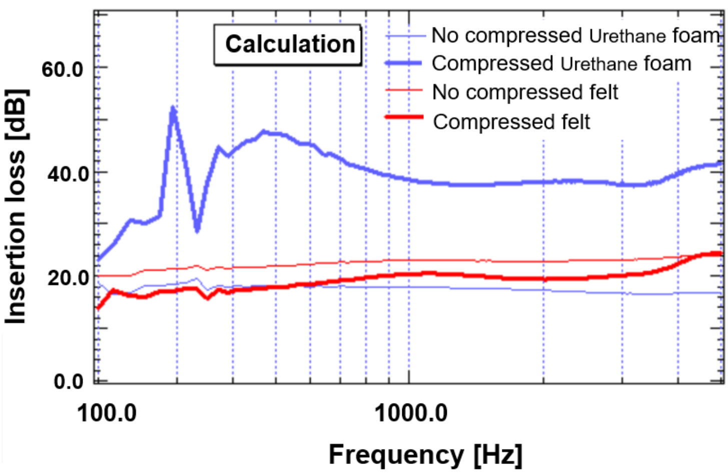


Fig. 11. Calculated insertion losses for the Urethane foam and the felt with / without 33% compression in the double walls.

On the contrary, the red lines in Fig.11 show the calculated results of the insertion losses for a typical fibrous porous layer (i.e. felt). In Fig.11, red thick line shows the insertion loss with the 33% compression of the felt, and the thin red line shows that without the compression. As can be seen in the figure, the insertion loss decrease because of the 33% compression for the felt. Inversely for this felt, the porous layer become harder because of a hardening characteristics in the restoring force due to the material nonlinearity as shown in Fig. 5. This causes that the vibration propagation from the steel base plate to the acrylic cover plate increases. Then, the sound radiation from the double walls having the compressed felt increases. We have lower sound insulation due to the compression for this felt.

We can conclude that after compression properly, the Urethane foam has the higher sound insulation performance than that of the felt.

Figure 12 shows experimental insertion losses for the Urethane foam and the felt with / without the 33% compression in the double walls. Qualitatively, these experimental results have the same tendency with the calculated results in Fig.11.

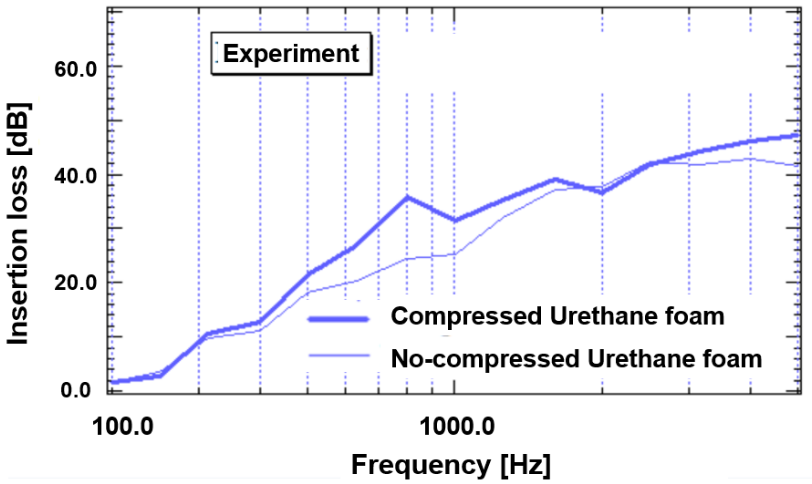


Fig. 12 Experimental insertion losses for the Urethane foam with / without 33% compression in the double walls.

5. Conclusion

We carry out nonlinear numerical simulation of insertion loss for structures having compressed porous layers sandwiched by double walls under acoustic excitation. For the porous layers, we used a typical soft urethane foam and a felt around automotive body panels as sound insulators. As increasing compression load, Urethane foam has soft-hardening characteristics in its restoring force. If we set 33% compression, lower rigidity is obtained for Urethane foam. This leads to softer porous layer for the double walls. And transmission of vibration and sound decrease from the base plate to the cover plate. For felt, the hardening characteristics appear in its restoring force. This causes the higher rigidity and the more transmission after compression. These results show that by appropriately compressing urethane foam and using it in a sound-absorbing double wall, it is possible to reduce sound transmitted into the interior of cars.

References

[1] T. Yamaguchi, Y. Kurosawa and S. Matsumura, “FEA for damping of structures having elastic bodies, viscoelastic bodies, porous media and gas”, ***Mechanical Systems and Signal Processing***, Vol.21, No.1, pp.535-552, 2007.

[2] T. Yamaguchi, Y. Kurosawa and H. Enomoto, “Damped vibration analysis using finite element method with approximated modal damping for automotive double walls with a porous material”, ***Journal of Sound and Vibration***, Vol.325, pp.436-450, 2009.

[3] T. Yamaguchi, H. Hozumi, Y. Hirano and Y. Kurosawa, “Nonlinear transient response analysis for double walls with a porous material supported by nonlinear springs using FEM and MSKE method”, ***Mechanical Systems and Signal Processing***, Vol.42, pp.115-128, 2014.

[4] Y. J. Kang and S. Bolton, “Finite element modeling of isotropic elastic porous materials coupled with acoustical finite elements’’, ***Journal of the Acoustical Society of America***, Vol.98, No.1, pp.635-643, 1995.

[5] N. Attala, R. Panneton and P. Debergue, “A mixed pressure-displacement formulation for poroelastic materials’’, ***Journal of the Acoustical Society of America***, Vol.103, No.3, pp.1444-1452, 1998.

[6] T. Yamaguch. J. Tsugawa, H. Enomoto and Y. Kurosawa, “Layout of Sound absorbing materials in 3D rooms using damping contributions with eigenvectors as weight coefficients’’, ***Journal of System Design and Dynamics***, Vol.4, No.1, pp.166-176, 2010.

[7] K. Takebayashi, A. Tanaka, K. Andow and T. Yamaguchi, “Modal loss factor approximation for u-p formulation FEM using modal strain and kinetic energy method”, ***Journal of Sound and Vibration***, Vol.505, pp.1-14, 2021.

[8] B. Storakers, “On material representation and constitutive branching in finite compressible elasticity”, ***Journal of the Mechanics and Physics of Solids***, Vol.34, pp. 125-145, 1986.

[9] M. Laroussi, K. Sab and A. Alaoui, “Foam mechanics non-linear response of an elastic 3D-periodic microstructure”, ***Journal of Solids and Structures***, Vol.39, pp. 3599-3623, 2002.

[10] T. Yamaguchi and K. Nagai, “Chaotic vibrations of a cylindrical shell-panel with an in-plane elastic- support at boundary”, ***Nonlinear Dynamics***, Vol.13, No,3, pp. 259-277, 1997.