6-2015

Structural Damping by the Use of Fibrous Materials

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# STRUCTURAL DAMPING BY THE USE OF FIBROUS MATERIALS

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Car door system
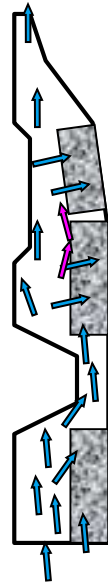
- If materials normally used for airborne noise control (i.e. fibrous material) can also be used to provide structural damping, then it is possible to reduce the weight of vehicle

- When a fibrous medium is placed close to the panel in the region where the oscillatory nearfield is significant, energy is dissipated by the viscous interaction of the flow and the fibers, and the panel vibration is damped
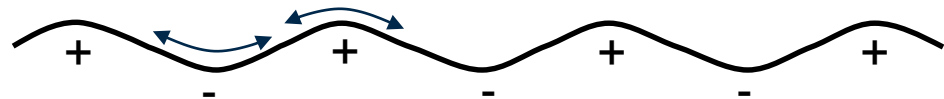
Experimental setup
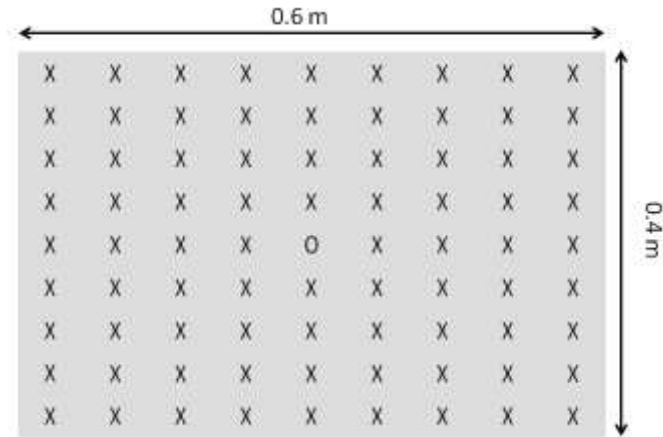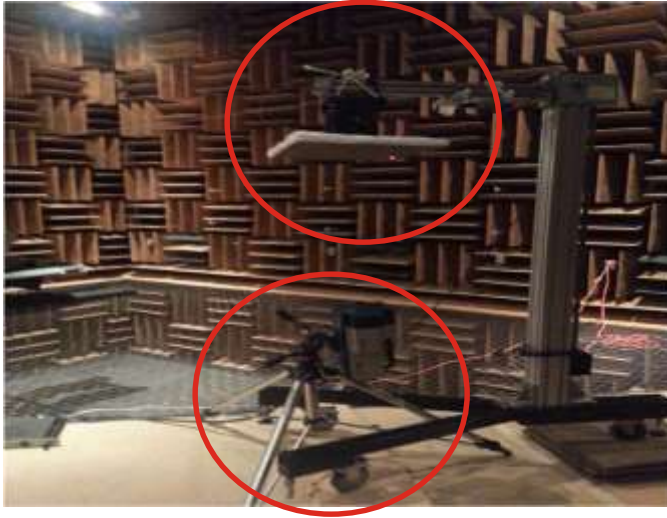
- A steel panel: 0.6 m by 0.4 m and thickness of 0.75 mm
- The panel was hung horizontally at its center by a bolt and stinger connected to a shaker
- Generated white noise (0 -10 kHz) from Polytec PSV-400
- Measured the panel surface normal acceleration and calculated the transfer accelerance, *a/F*
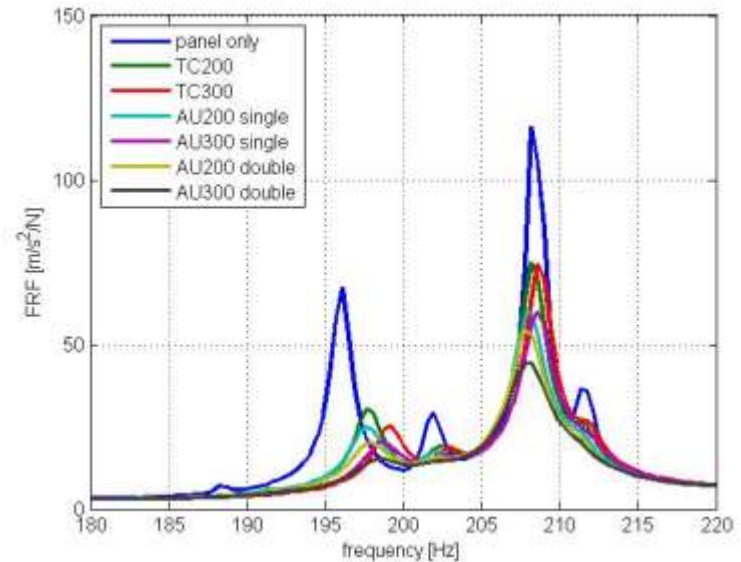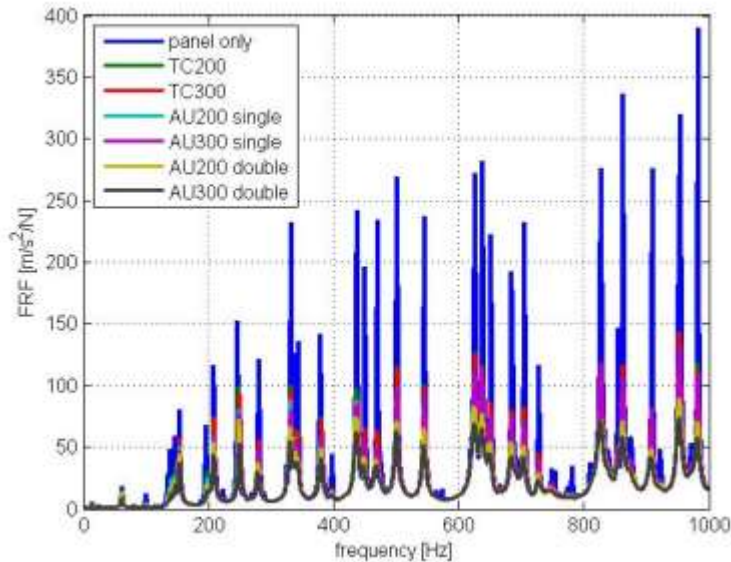
Acoustical material

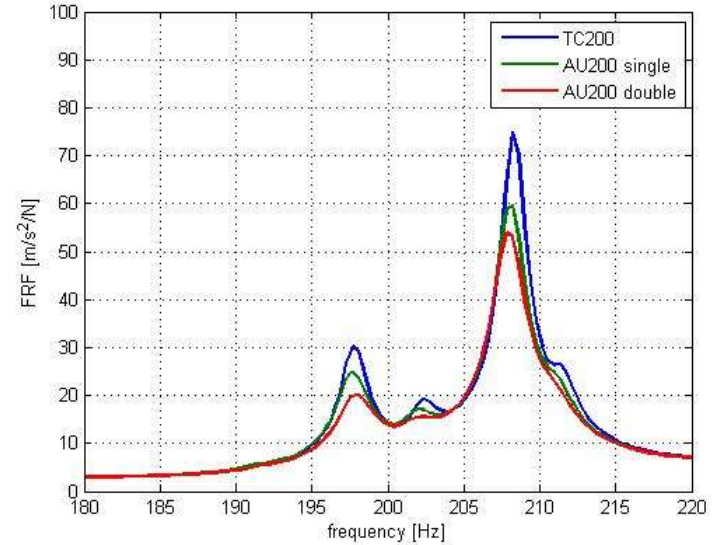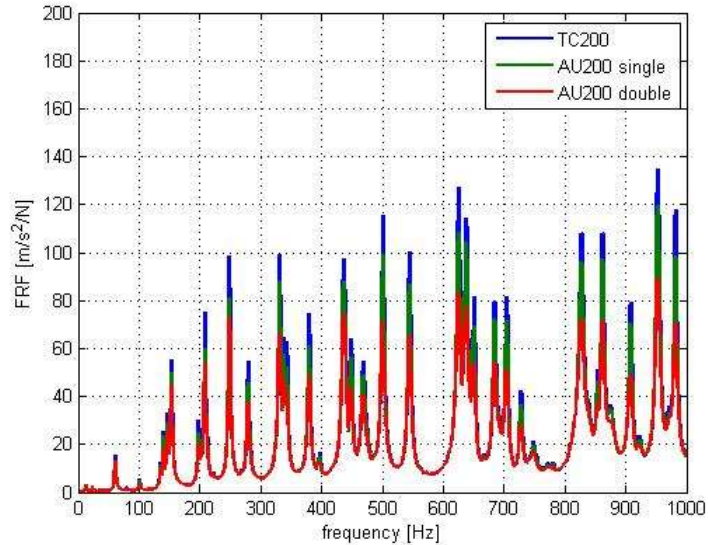| Model	Mass per unit area [kg/m^2]	Thickness [mm]
TC-200	0.2	29
TC-300	0.3	41
AU-200 single scrim	0.2	13
AU-300 single scrim	0.3	19
AU-200 double scrim	0.2	12
AU-300 double scrim	0.3	19

- 6 types of fibrous sound absorbing materials were used in the tests
- Materials were simply resting on the steel plate.

- Significant damping effect is seen in 0 to 1000 Hz.
- The peak locations in the panel-with-treatment case were slightly shifted.
- The peak magnitudes were reduced by half or more when the panel was treated with sound absorbing materials
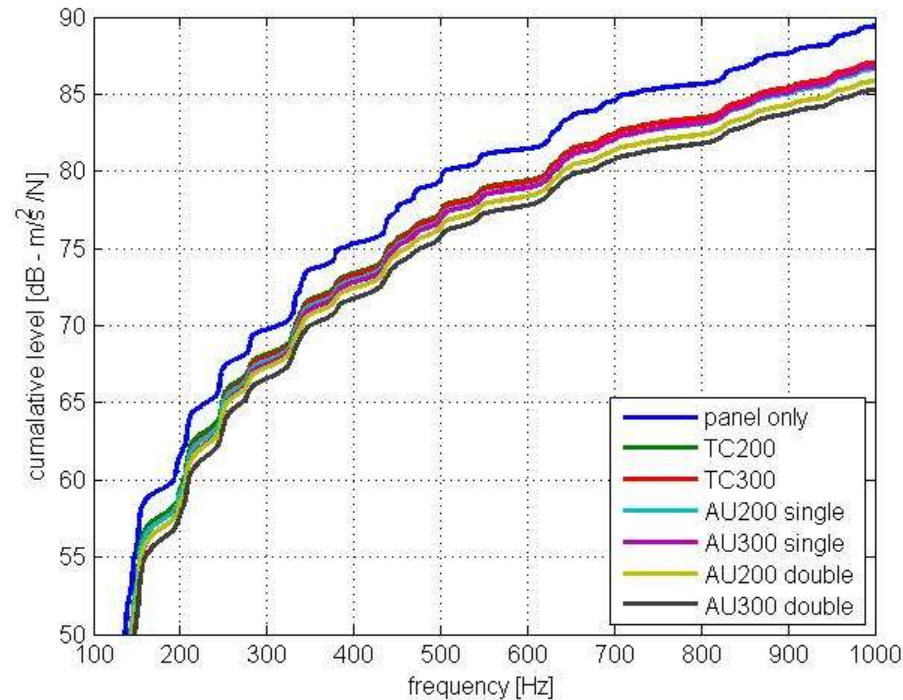
Measurement result

Model	Mass per unit area [kg/m^2]	Thickness [mm]
TC-200	0.2	29
AU-200 single scrim	0.2	13
AU-200 double scrim	0.2	12

- Comparing TC 200 with AU 200, mass is not only factor that controls damping effect
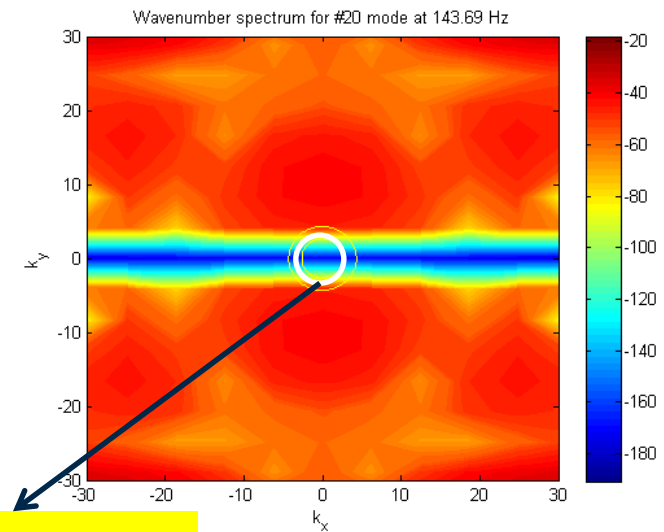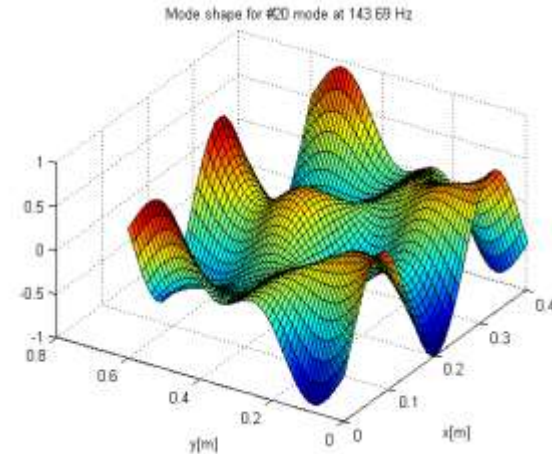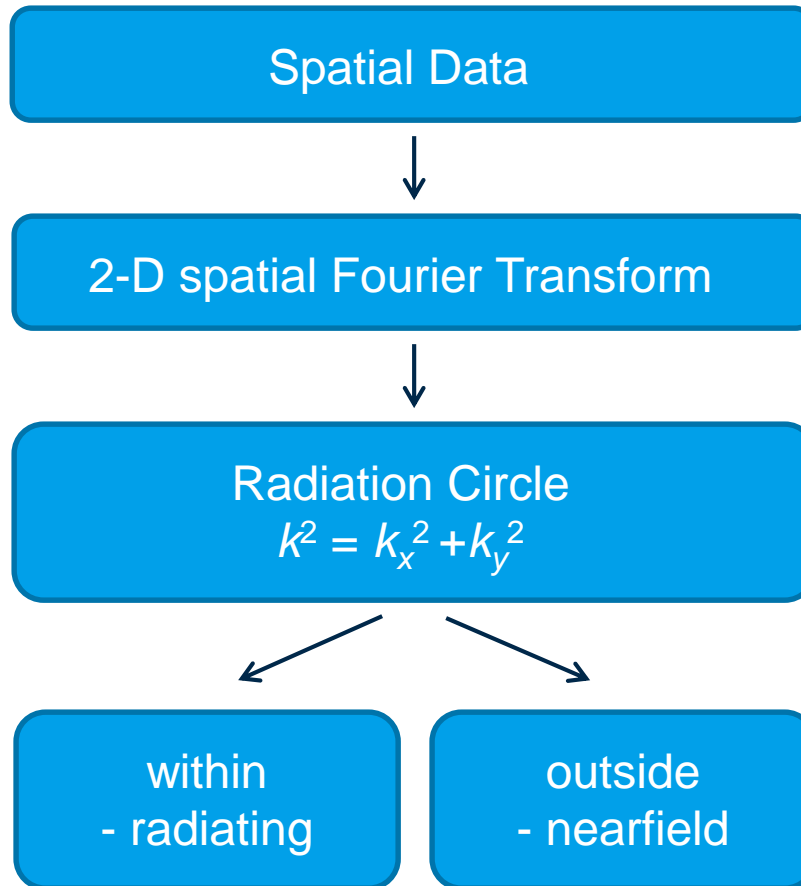
Measurement result

- Cumulative, space-averaged accelerance level was reduced by 3 to 5 dB when the sound absorbing materials were added compared to the untreated plate.
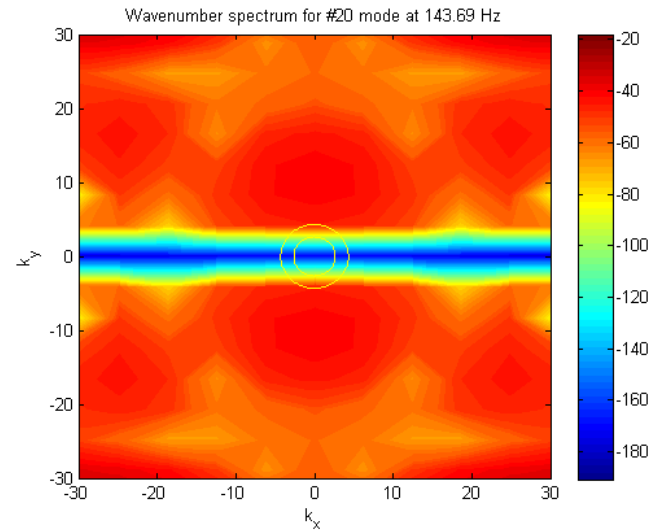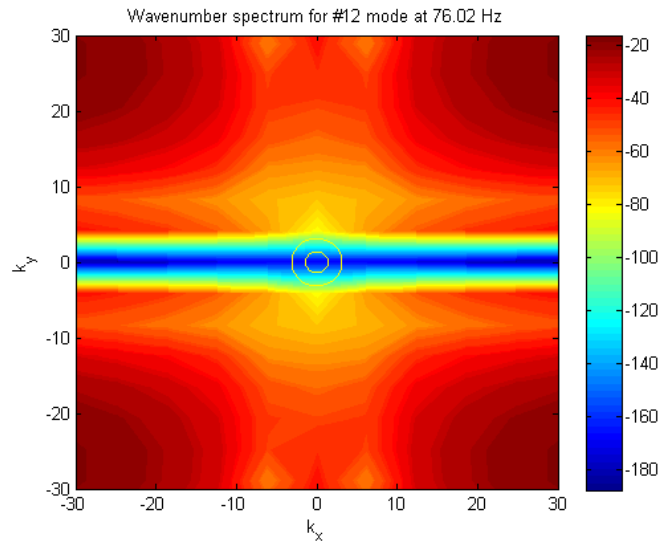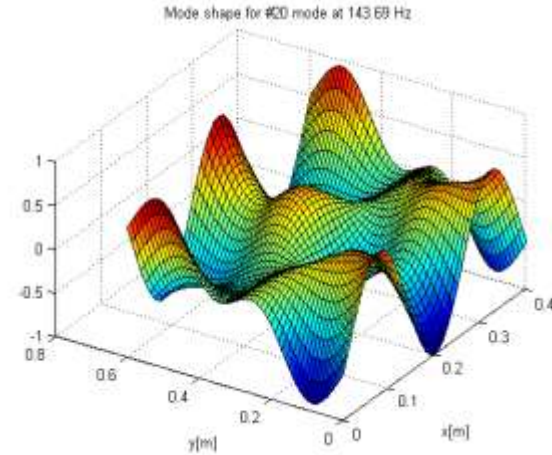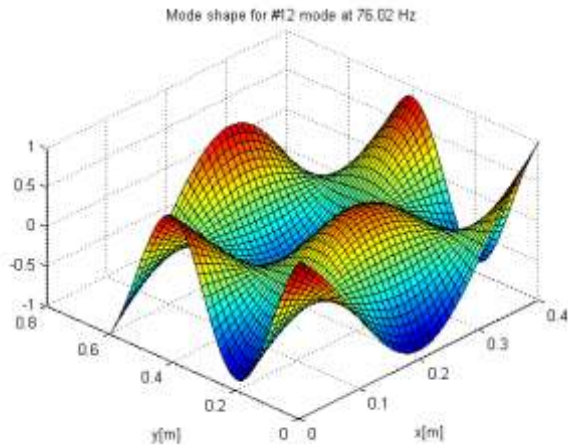
FEA model
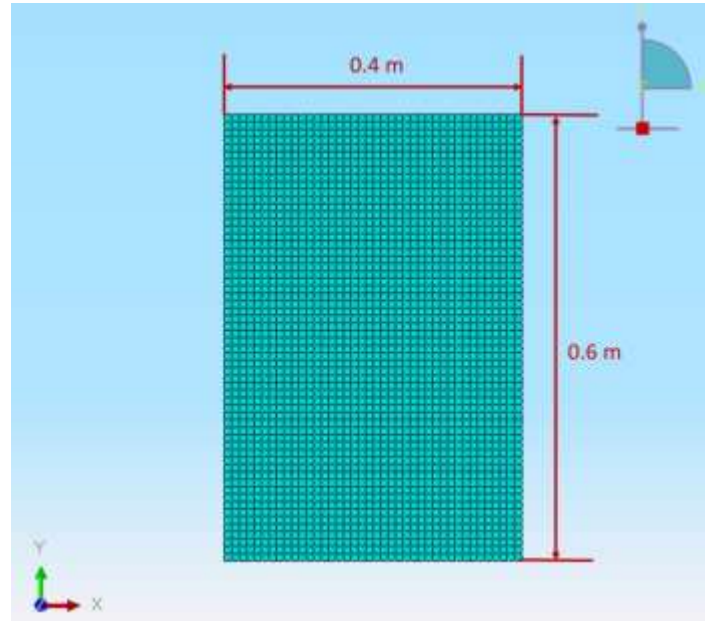
- ABAQUS^{TM} was used to perform the modal analysis
- 2D shell
- Free-free boundary condition
- The size of element: 0.01 m X 0.01 m

Wavenumber Transform Analysis

Spatial Data

$\downarrow$

2-D spatial Fourier Transform

$\downarrow$

Radiation Circle
$$k^2 = k_x^2 + k_y^2$$

within
- radiating

outside
- nearfield

Mode shape for #20 mode at 143.69 Hz
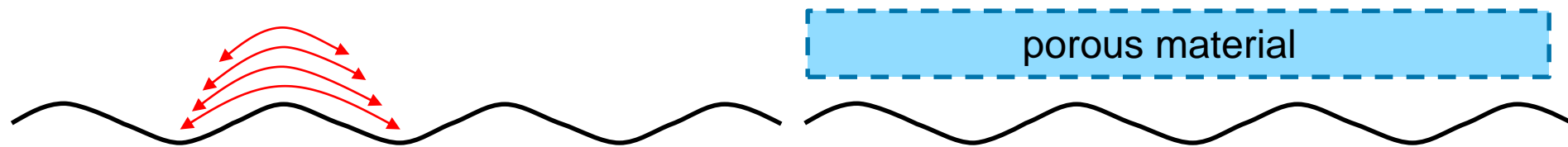
Wavenumber spectrum for #20 mode at 143.69 Hz

Radiation circle

Sound radiation simulation

Nearfield Damping

- Non-radiating components

  - Incompressible oscillatory velocity
  - Magnitude decreases exponentially with distance from panel

porous material

  - Place fibrous material in the nearfield
  - Energy dissipated by viscous interaction of fluid and fibers

Effective nearfield thickness

- How thick should the fibrous layer be to be effective?
- Calculate nearfield depth
- Wave equation

$$k_z = \sqrt{k^2 - k_x^2 - k_y^2} \quad \text{where, } k = \frac{\omega}{c}$$

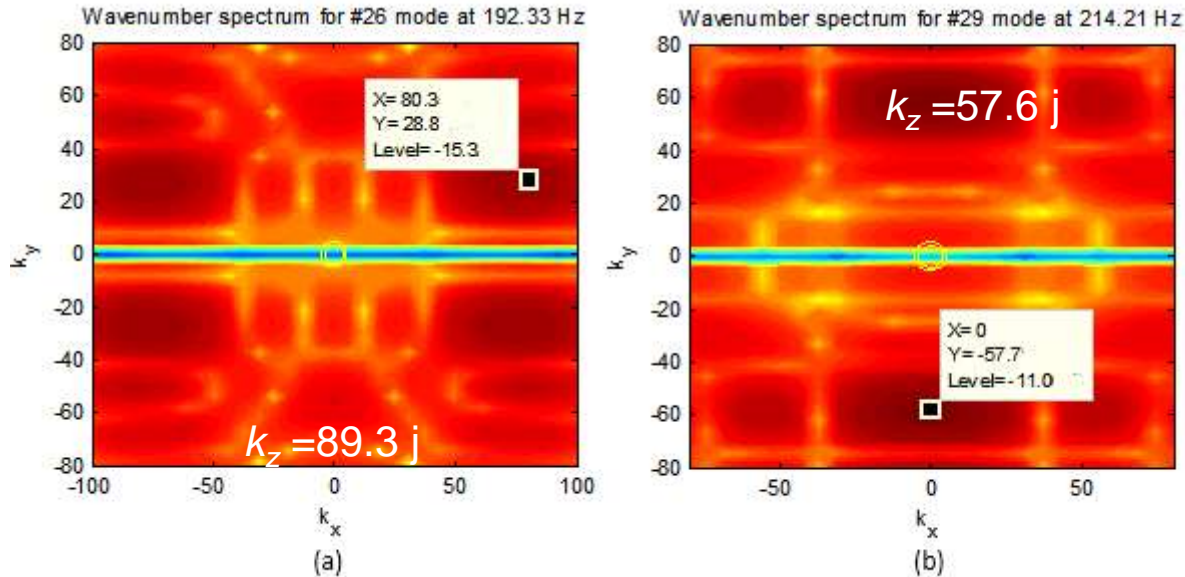- The velocity magnitude in the nearfield region decays exponentially with the result that we can define

$$\xi = \exp(-\gamma \Delta_n) \quad \text{where, } k_z = j\gamma$$

where

ξ: proportional to the strength of the nearfield motion

Δ_n: the thickness of the fibrous material at the n th natural frequency that creates significant damping

Comparison with measurement

- Find location of peak in nearfield region of the wavenumber spectrum of each mode
- Calculate nearfield depth based on location of peak in nearfield wavenumber spectrum

Effective nearfield thickness


Fibrous material thickness prediction
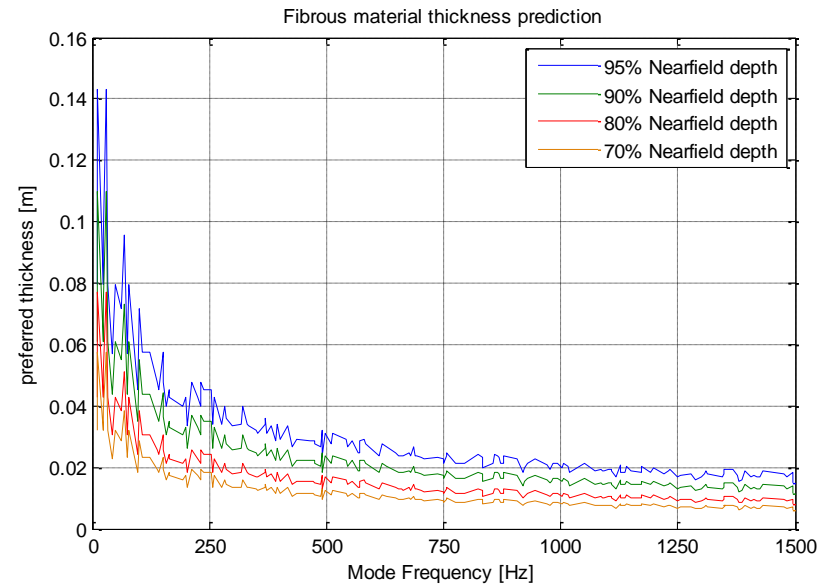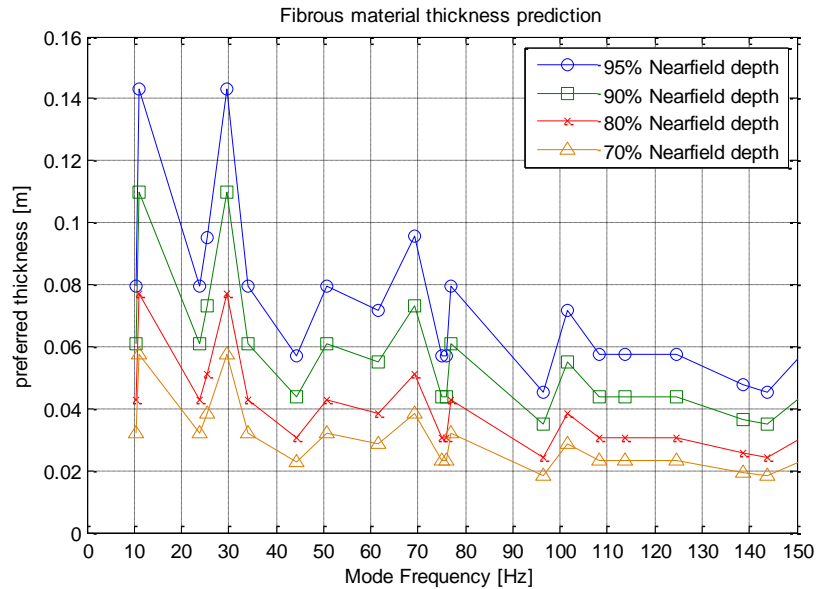
- 0.142 m thick fibrous layer occupies 95% of the nearfield region at the 2nd (10.98 Hz) and 5th (29.70 Hz) natural frequencies of the steel panel

- 6 cm fibrous layer could occupy at least 70% of the nearfield region of the steel plate even at the lowest modal frequency

- As the natural frequency increases, the necessary thickness of the fibrous material decreases

Comparison with measurement

Fibrous material thickness prediction
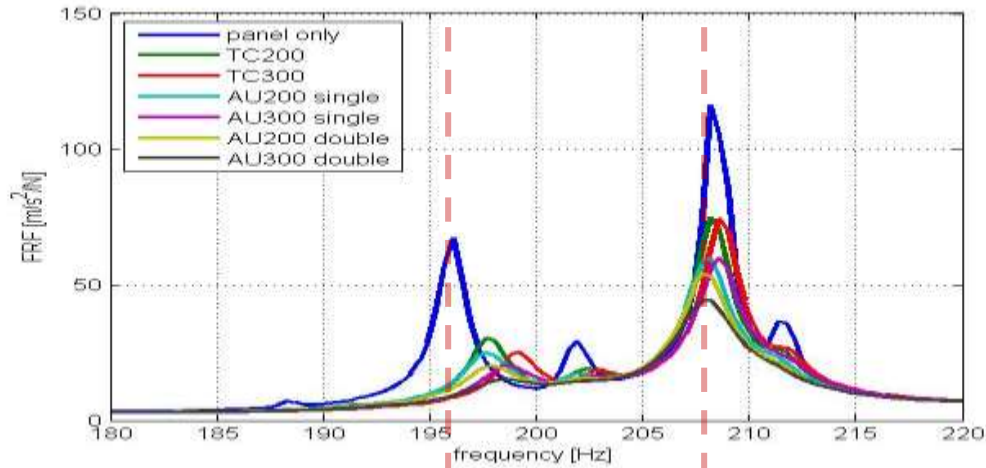
- **Fibrous layer treatments range**
  - thickness from 12 mm to 41 mm

- **Consistent with required thickness to be effective**

Simulation study
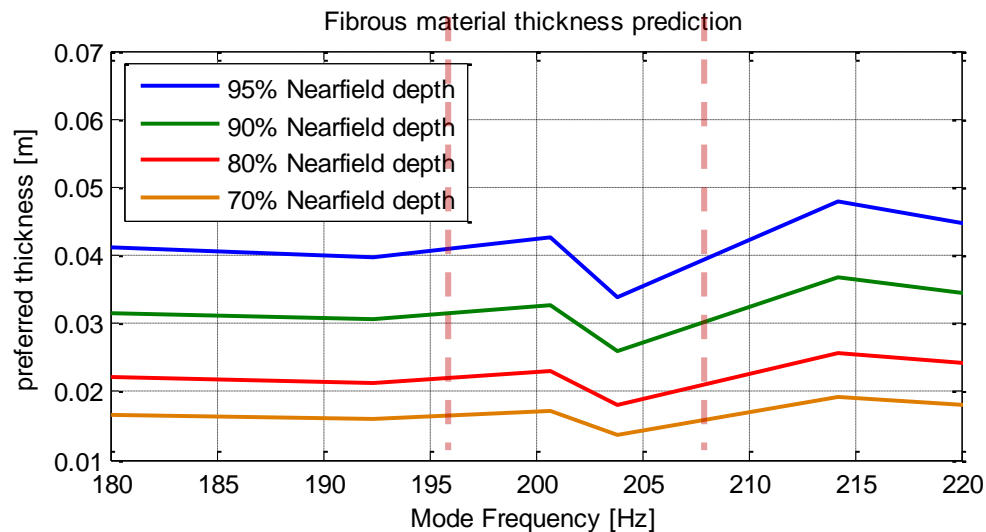
- Since the fibrous materials considered as limp, Johnson-Champoux-Allard model was used - modeled as fluid with complex density and bulk modulus

- Complex density

$$\tilde{\rho}(\omega) = \frac{\alpha_\infty \rho_0}{\phi}\left[1 - j\frac{\sigma\phi}{\omega\rho_0\alpha_\infty}\sqrt{1 + j\frac{4\alpha_\infty^2\eta\rho_0\omega}{\sigma^2\Lambda^2\phi^2}}\right]$$

where, ρ is porosity, α_∞ is the dynamic tortuosity, σ is the flow resistivity, η is the dynamic viscosity of air, and Λ is the viscous characteristic length.
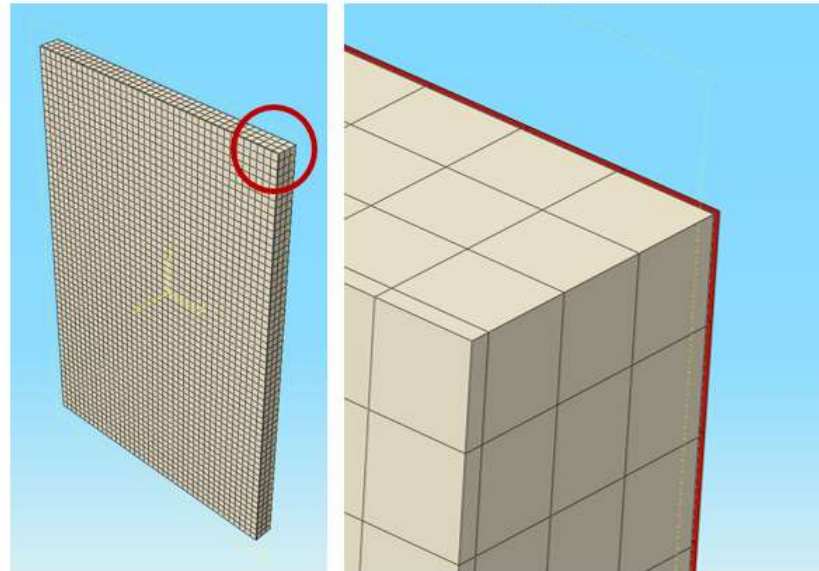
- The dynamic bulk modulus

$$\tilde{K}(\omega) = \frac{\gamma P_0 / \phi}{\gamma - (\gamma - 1)\left[1 - j\frac{8\kappa}{\Lambda'^2 C_p\rho_0\omega}\sqrt{1 + j\frac{\Lambda'^2 C_p\rho_0\omega}{16\kappa}}\right]^{-1}}$$
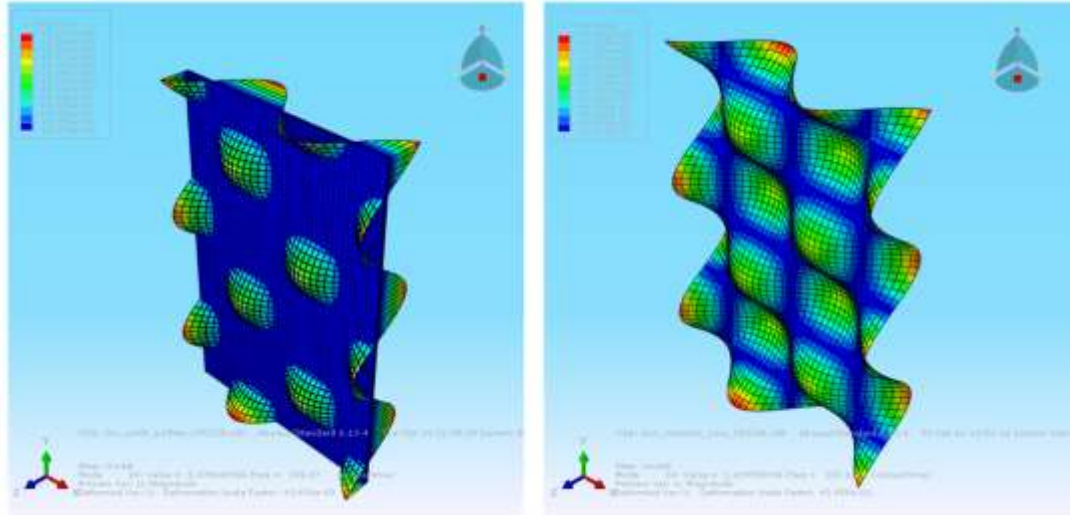
- To apply limp material case:

$$\tilde{\rho}'_{eq} \approx \frac{\tilde{\rho}_{eq}M - \rho_0^2}{M + \tilde{\rho}_{eq} - 2\rho_0}$$ where $M = \rho_{mat} + \phi\rho_0$

Future simulation study

- Detailed properties of TC 200

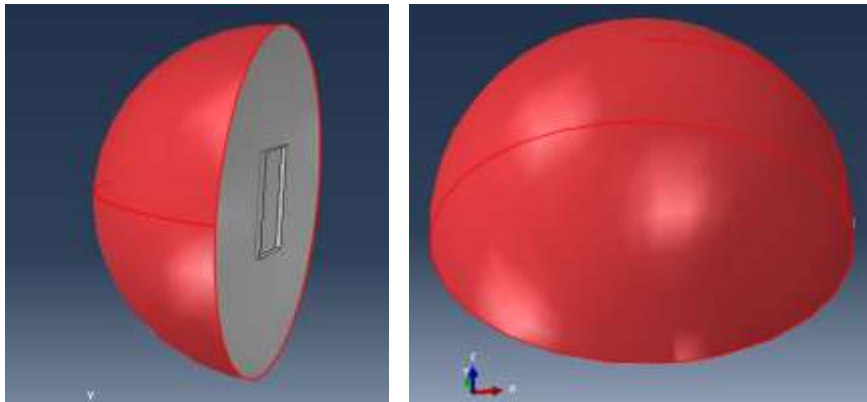|	Flow resistivity [Rayls/m]	Porosity	tortuosity	Thickness [mm]	Density [kg/m^2]
TC200 fiber	6400	0.99	1.1	29	0.24
TC200 scrim	150000	0.99	1.1	1	16

Simulation study

- The same mode shape appears (187 Hz and 200 Hz)
- The total mass of the structure was increased by the addition of the acoustical material

Simulation Study

Structural-Acoustic fully coupled simulation analysis

    - ABAQUS^{TM} was used.

    - Hemispherical sound field (radius = 1m) was attached to the panel and the fibrous material.

    - Non reflecting interaction was adopted around the hemispherical surface.

    - CPU time for simulation: Intel i7-4771 3.50Ghz, 16 GB RAM

        1) Panel only case: 1 hour 38 min

        2) Panel with fibrous material treated case: 14 hour 35 min

Sound field Modeling – Non reflecting boundary **Sound field mesh – Air properties**
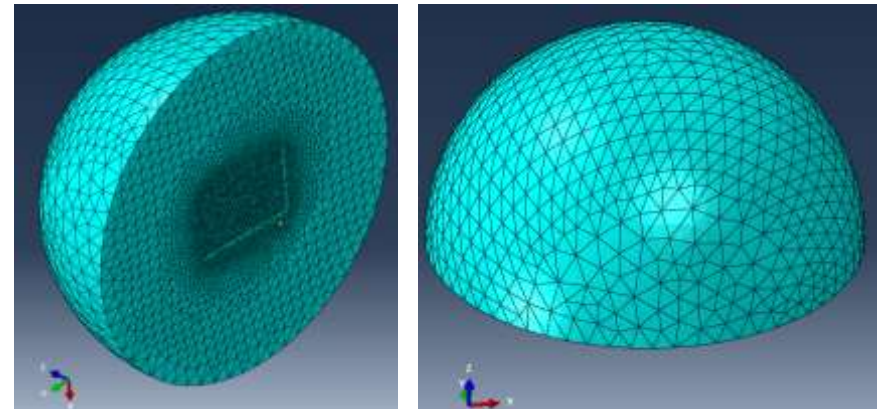
Simulation study
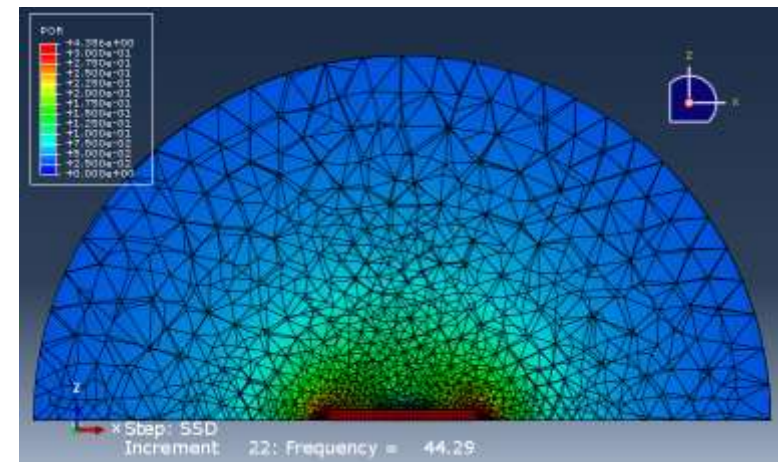
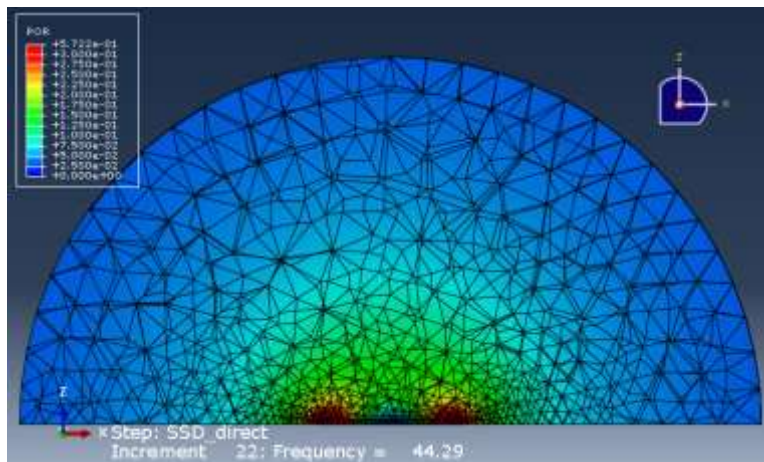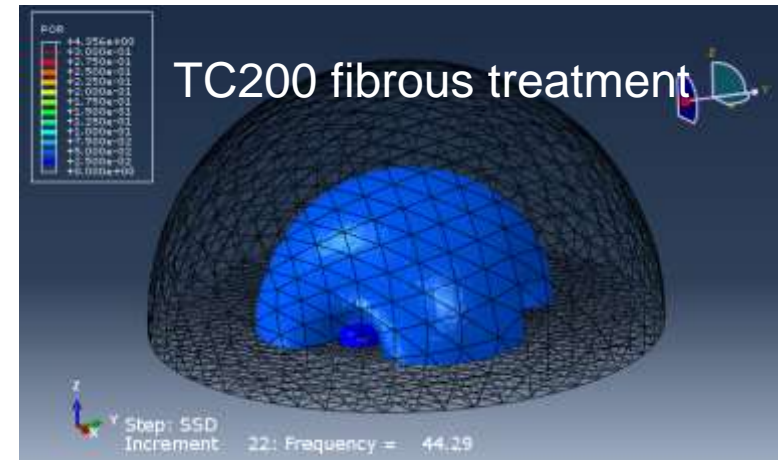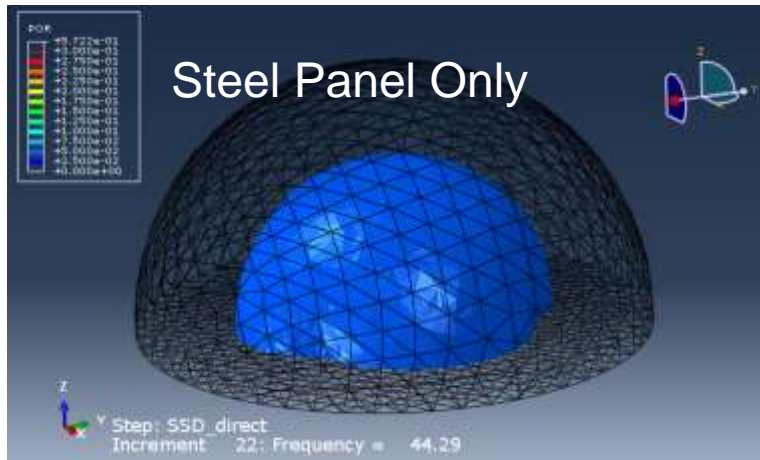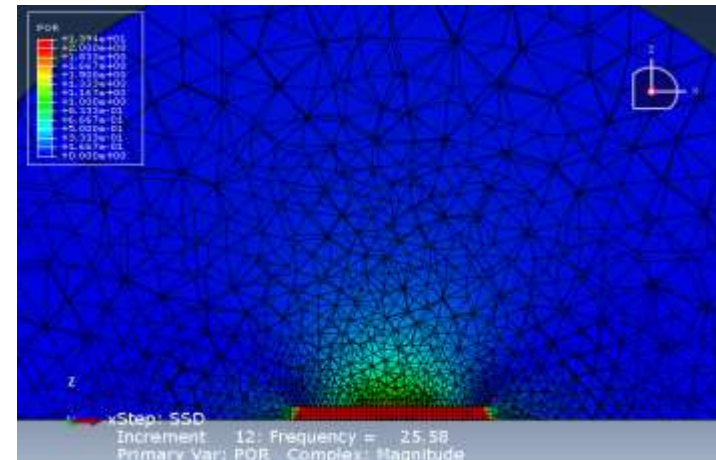**Acoustic Pressure distribution at Low Frequency – 23.71 Hz**

- **Mode is close to inefficient sound radiator**
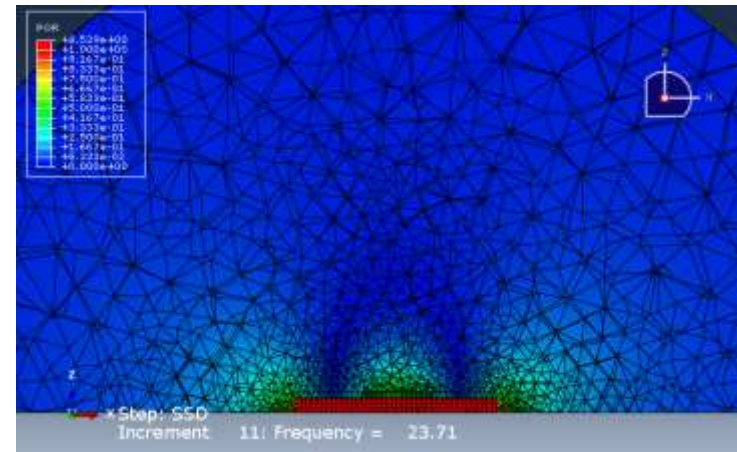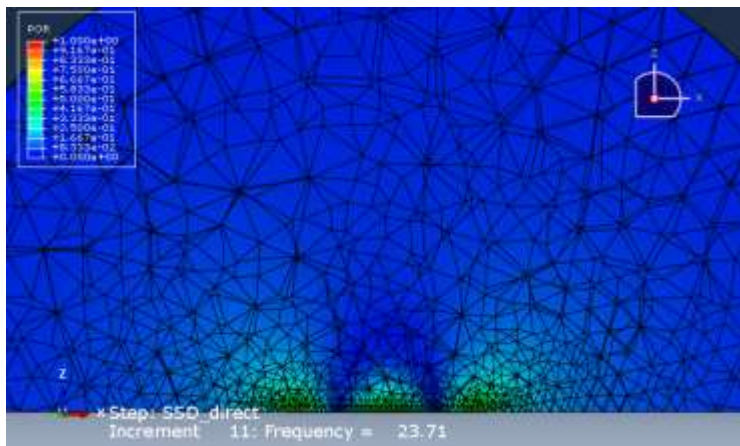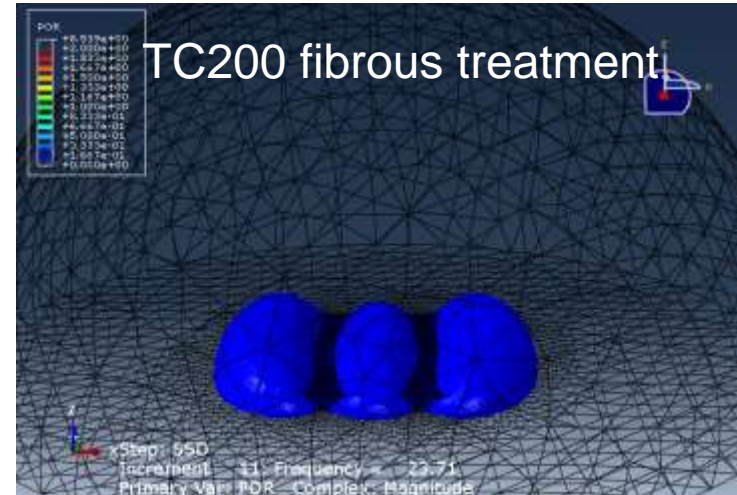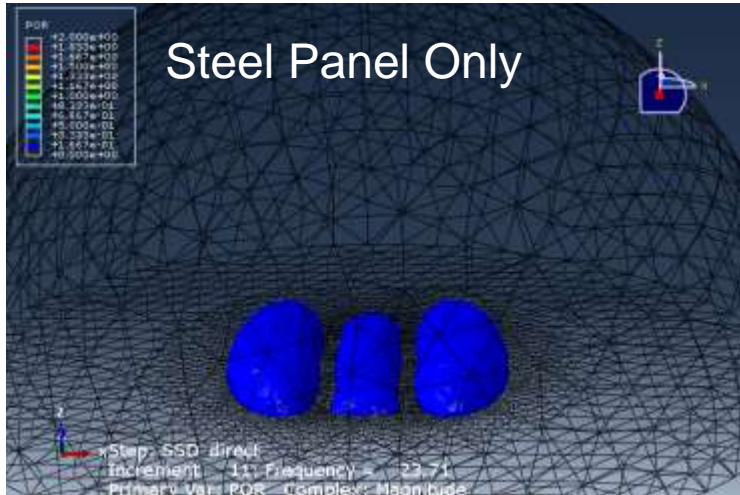
Simulation study

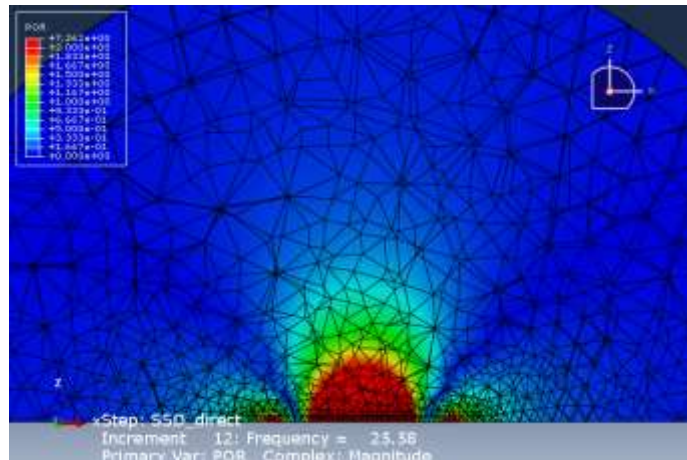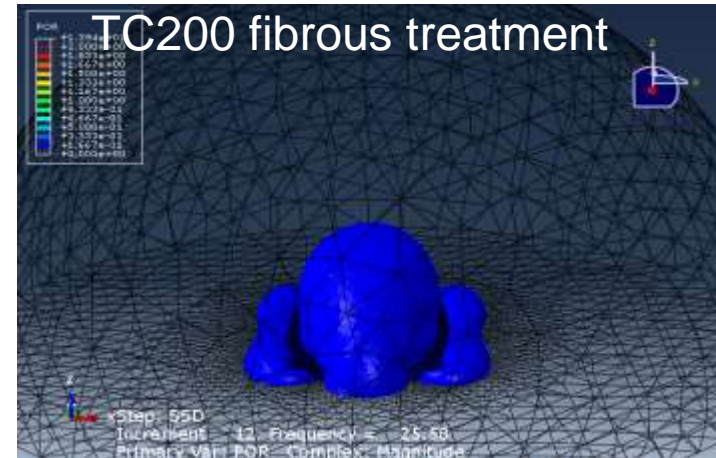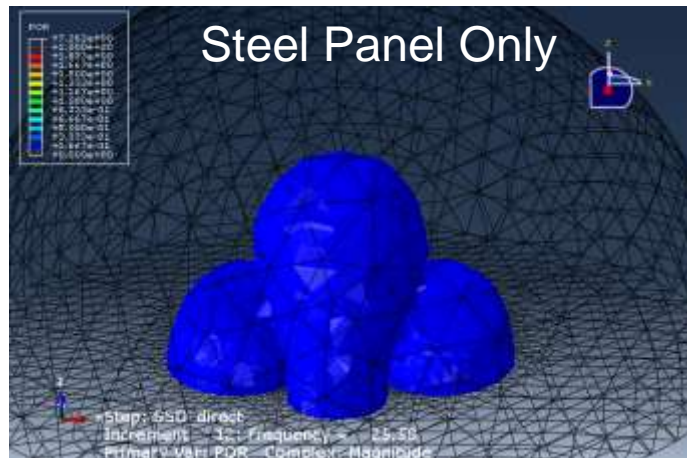**Acoustic Pressure distribution at Low Frequency – 44.29 Hz**

- **Mode is close to inefficient sound radiator**

Simulation study

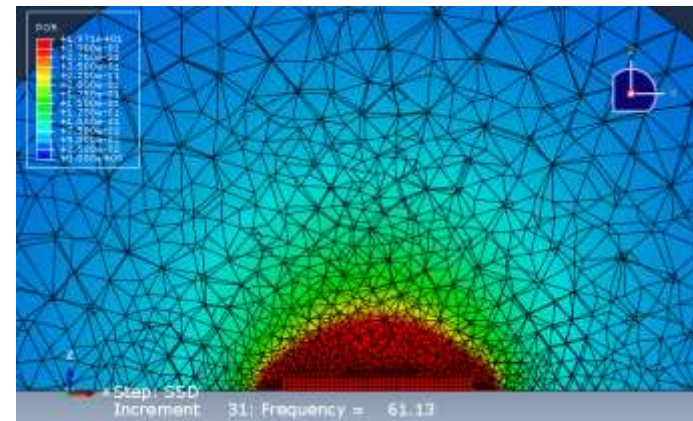**Acoustic Pressure distribution at Low Frequency – 25.58 Hz**

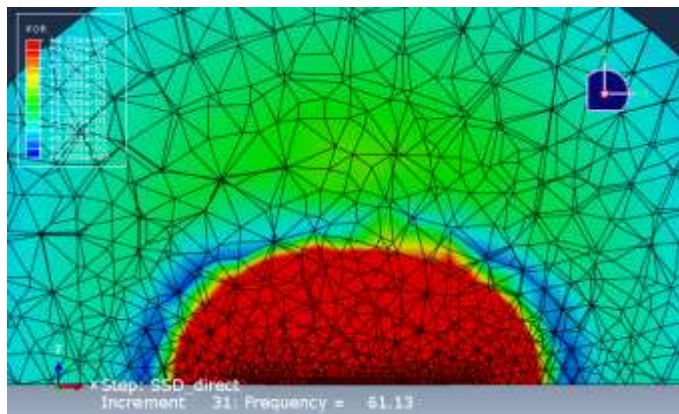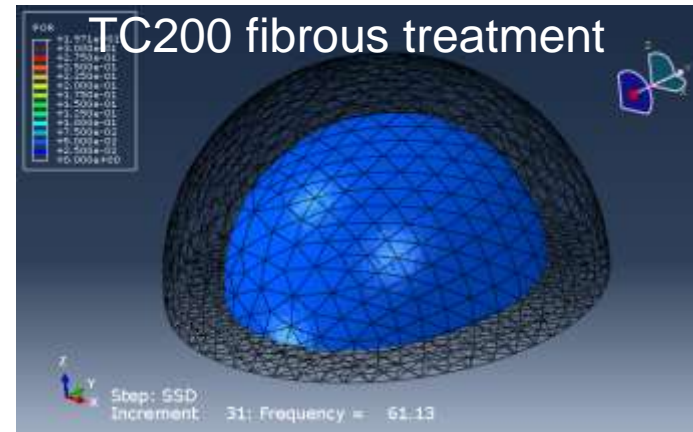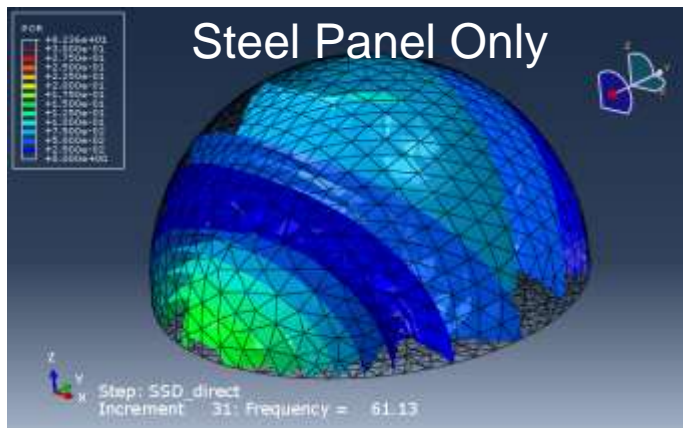   **- Mode is close to efficient sound radiator**
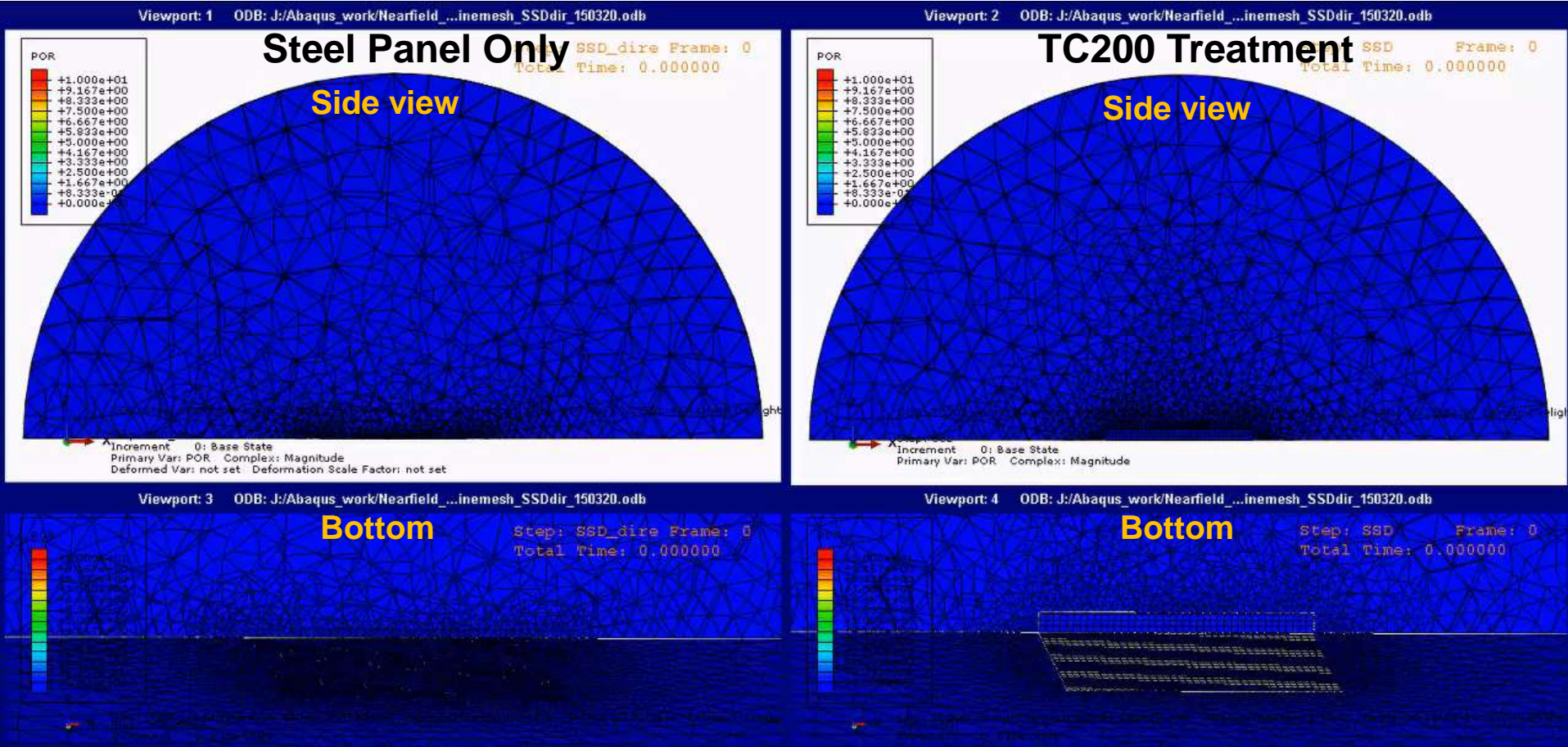
Simulation study

Acoustic Pressure distribution at Low Frequency – 61.13 Hz

   - Mode is close to efficient sound radiator
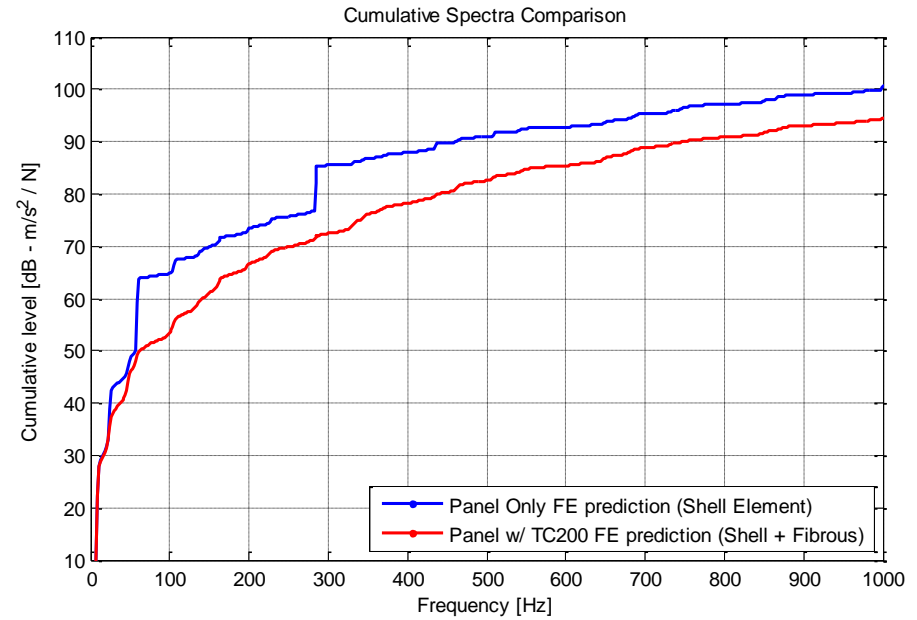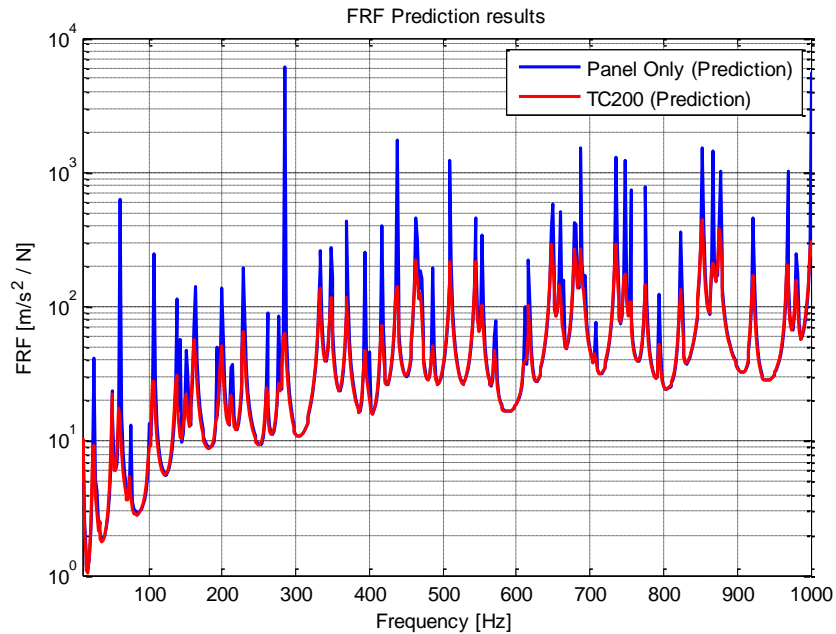
Simulation study

Sound pressure radiation in the air

Simulation study

FRF Prediction results

Cumulative Spectra Comparison
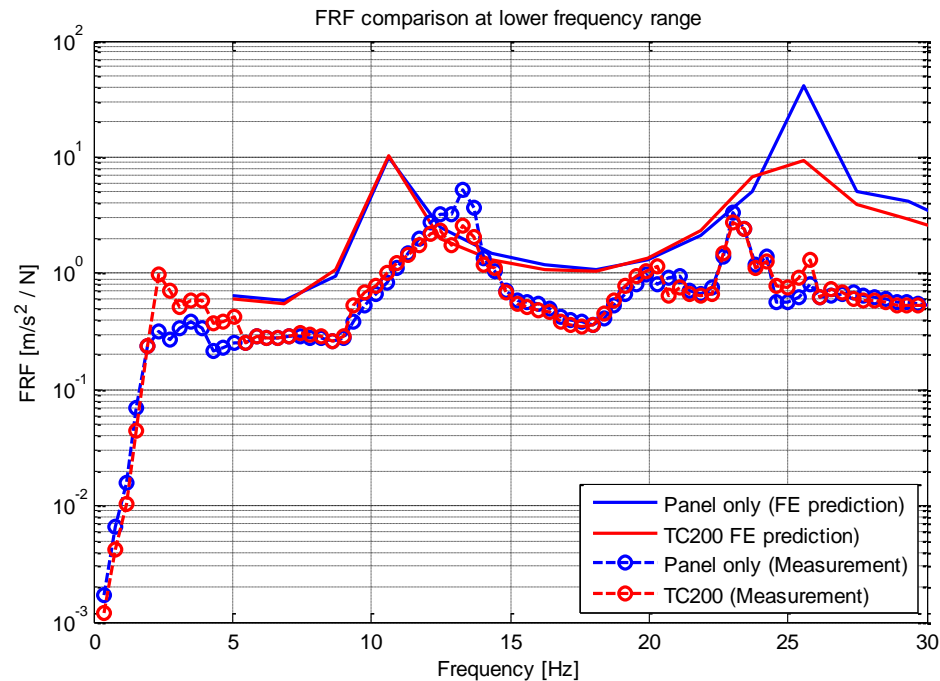
- • Sound absorbing materials resulted in damping of the structural vibration of the steel panel

At low frequency region

- Comparison between measurement and simulation at low frequency range(0 – 30 Hz)
- Mode frequencies of prediction model are slightly different compared with those of measurements due to different boundary conditions

FRF comparison at lower frequency range

Conclusion

- Sound absorbing material can be used to reduce structural vibration of a panel structure.

- A method to determine the appropriate thickness of a sound absorbing layer for structural damping purposes was proposed based on an analysis using the wavenumber spectrum.

- A finite element model to predict the structural damping effect of the sound absorbing material was suggested using the limp porous material model.

Future work

- More detailed simulation studies will be conducted that will consider different properties of fibrous materials and the structures.

- The loading effects of the sound field radiated through the fibrous layer will be studied.