中车株洲电力机车有限公司
CRRC ZHUZHOU LOCOMOTIVE CO., LTD.

Application of lightweight acoustic materials and structures in rail vehicles

Dengke Li

9th Nov. 2023

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01 Motivations & Background

Dengke Li

- Technical expert and senior engineer at CRRC ZHUZHOU LOCOMOTIVE Co., Ltd. (CRRC), member of the Environmental Physics Branch of the China Environmental Science Society, and member the Vibration and Noise Control Technical Committee at CRRC

Research areas:

- Rail vehilces vibration and noise control
- Novel noise reduction materials and structures

In 2018 receive a PhD degree from Chinese Academy of Sciences

Working with Camille Perrot under support of National CSC Program

In 2018 working as an acoustic engineer in CRRC

01 Motivations & Background

Noise inside cabin

Cabin of rail vehicles

Ultra High Voltage Transformer

Automobile

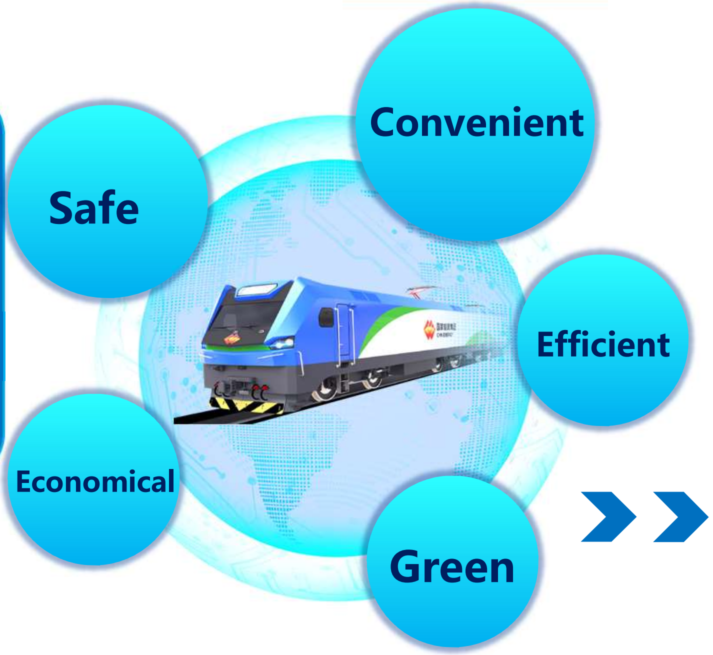
Underwater vehicles

Turboprop

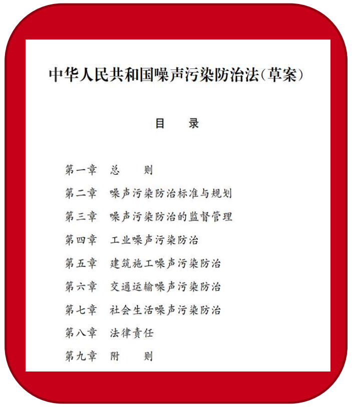
01 Motivations & Background

➢ Development trend of national transportation

- Building a safe, convenient, efficient, green, and economical modern integrated transportation system
- Vibration and noise in vehicles becomes increasingly prominent under high moving speeds and complex geographical and climatic conditions



High-speed & heavy loading



中华人民共和国噪声污染防治法(草案)

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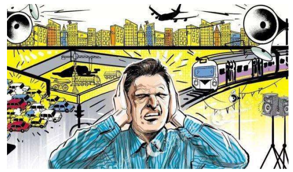
High Speed

Quiet Journey



Key problem of further development of high speed transportation

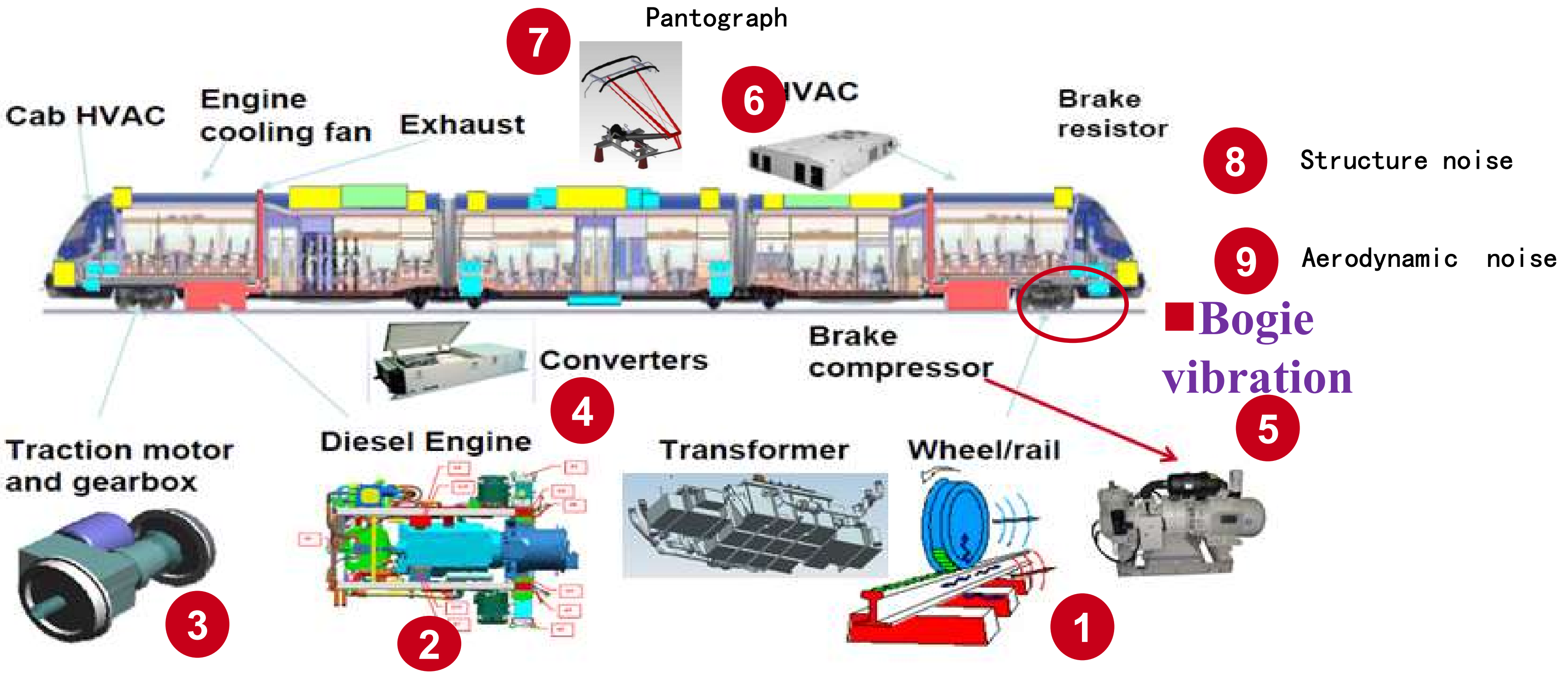
Ensuring acoustic comfort！！！

1. Where noise come from ?

2. What should do?

01 Motivations & Background

1.1 Main noise sources



Sources： Noise ＋ Vibration

1.2 Main vibration paths

Excitation sources - structure (path) - response

- **Sources**

–Suspension vibration transmission

–Primary suspension excitation (wheelset to bogie)

–Secondary suspension (Bogie to car body)

–Wheel/rail interaction system

–Suspended equipment（with excitation source）

–Air dynamic load

- **Transmission Paths(Structure-borne and airborne)**

–Suspension

–connection points between structures

–Car Body structure

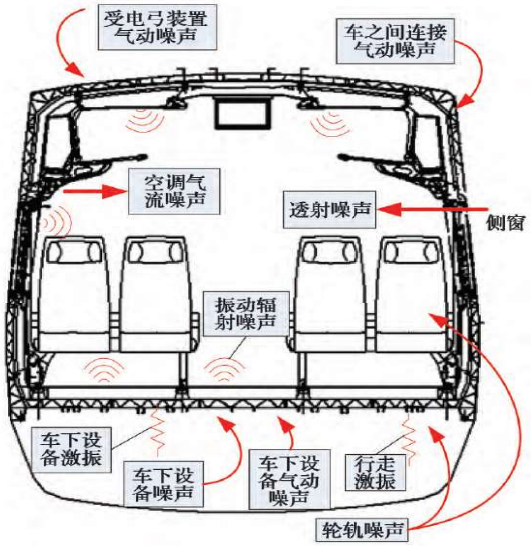
–Interior

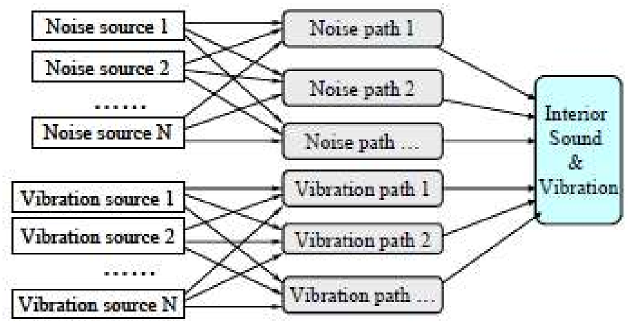
- **Response**

–Each part of car body

–Test points for ride quality index (steering wheel)

–Driver’ s and passenger's (seats)





01 Motivations & Background

- **Sources properties**

  Sound power
  of all noise sources

  System design

- **Body design**

  Transmission loss
  and absorption

  Air-borne sound

- **Interior design**

  Transmission and
  reverberation

  Air-borne sound

- **Vibration design**

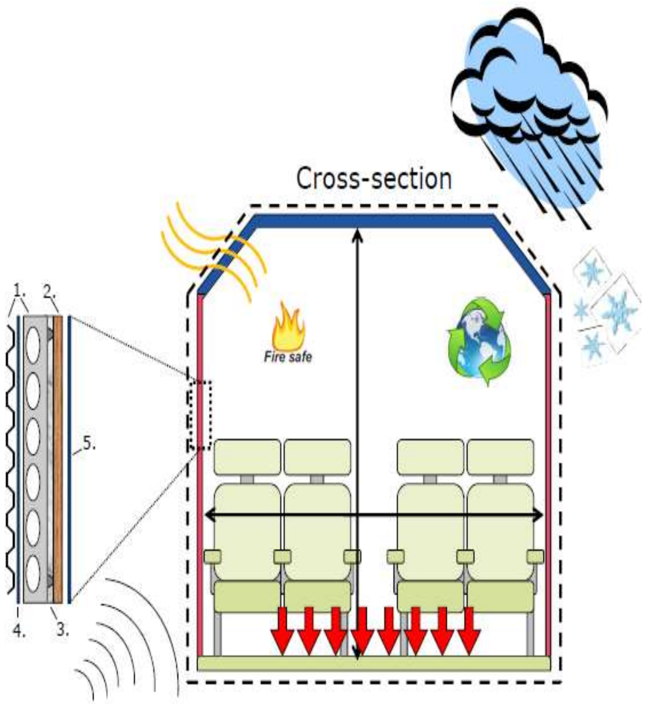
  Vibration
  isolation

  Structure-borne sound

Key requirement of acoustic materials

Considering multiple comprehensive performance factors

- ✓ Load-bearing capacity
- ✓ Vibration damping
- ✓ Noise reduction
- ✓ Fire resistance
- ✓ Thermal insulation
- ✓ Weather resistance



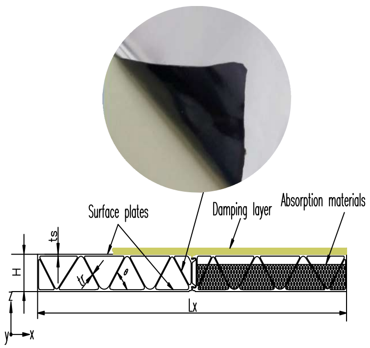
- Fireproof heat-resistant
- sound insulation, vibration reduction
- moisture resistance damp-proofing
- filtration dust absorption
- exterior decoration

Finding balanced solutions for various technical performance requirements !

Damping

for high frequency noise and vibration

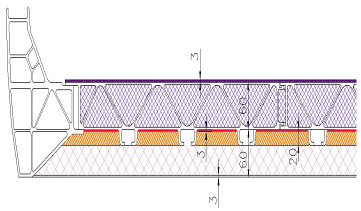
thin panel structure



Transmission loss

much depend on mass density

whole vehicle body areas



Absorption

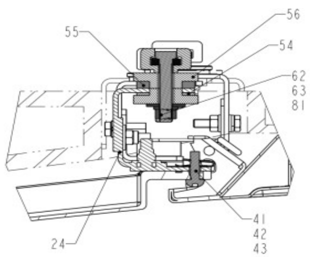
for middle and high frequency noise

interior porous materials and structures

Isolation

for low frequency vibration

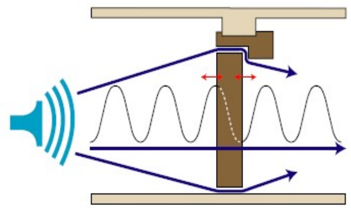
bogie area
Air conditioning compressor



Sealing

for high frequency noise

door and gangway



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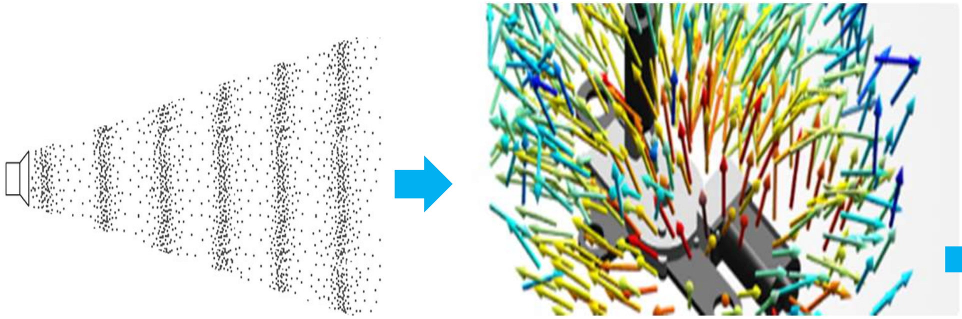
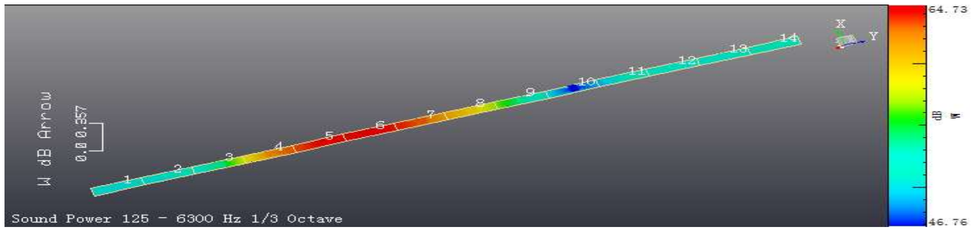
1.3 Noise identification

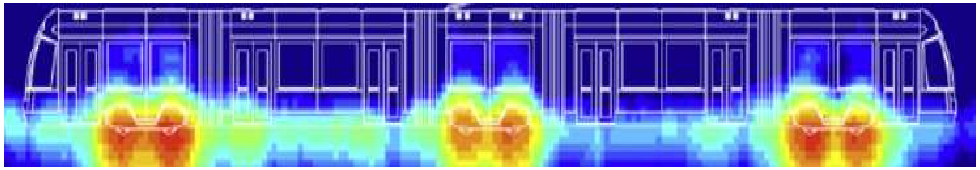
Noise source identification system by vector acoustic sensors



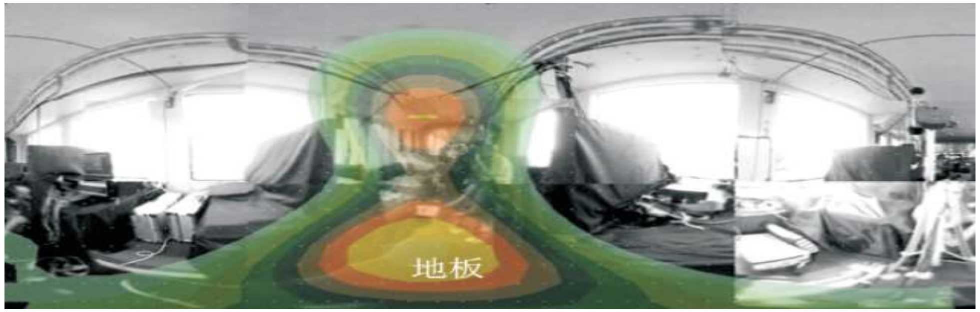


Airconditioning duct noise distribution

Noise source identification and transmission characteristics test



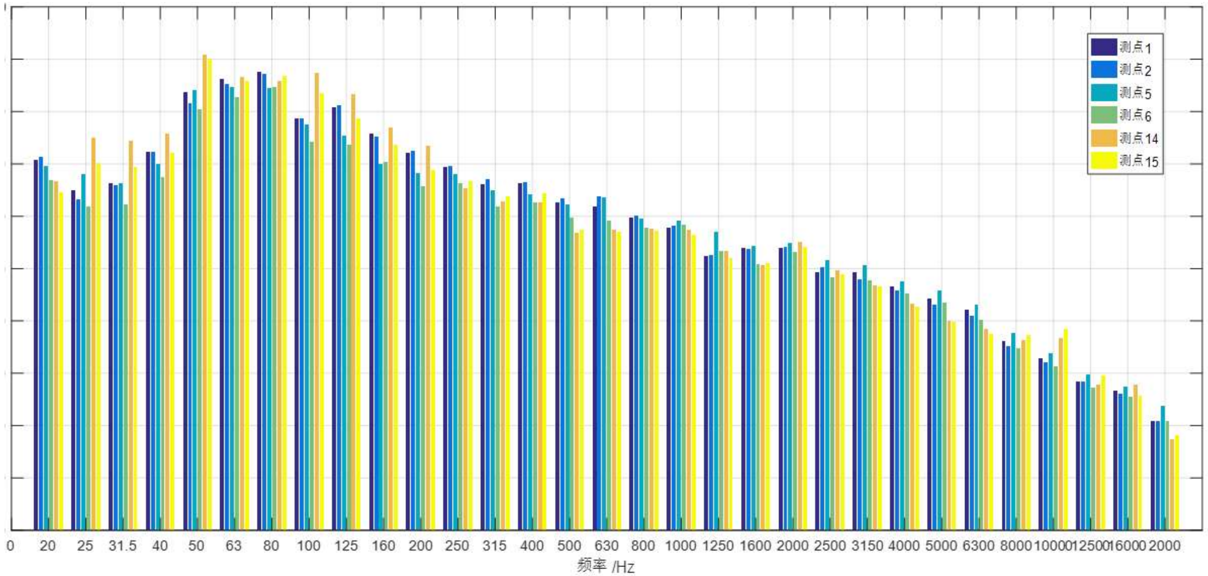
Exterior noise source identification



Interior noise source identification

01 Motivations & Background

1.4 Noise specturm



- 50-2000Hz dominates

Challenges：

- Improve low frequency sound insulation and absorption

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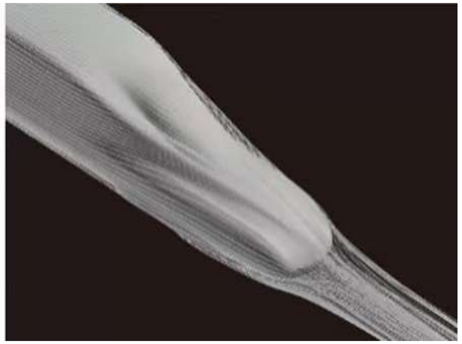
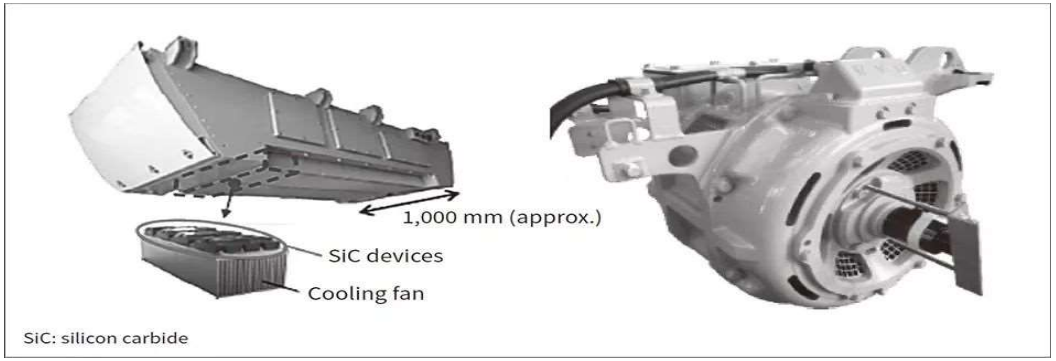
2.1 Japan_N700S

- a "super double wing" streamlined nose, which optimizes the aerodynamic performance of the high speed train
- minimizing the pressure wave into the tunnel, thus reducing the noise into the tunnel and improving the comfort of the train
- SiC device and high efficient induction motor is used to reduce the noise of transformer and motor



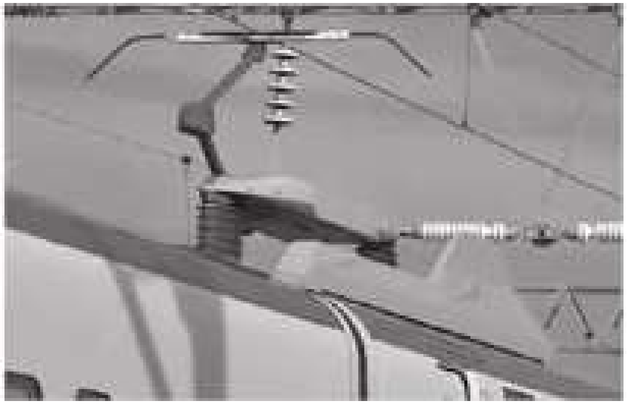


02 Research Progress

2.1 Japan_N700S

N700S

- ✓ Noise reduction material is used to reduce the noise from the bottom of the train
- ✓ Sound insulation barrier to reduce the radiation noise of the pantograph.
- ✓ High-performance sound absorption materials in cabin



2.2 Shift2 rail

Recent progress from Pro. David J. Thompson
- ➢ Mic array for beamforming
- ➢ Near-field array for WSE
- ➢ Static and dynamic vibration andnoise for ATPA (and MISO)

Field test to gather validation data

- One week in June 2016
- High speed track near Munich
- DB Schallmesswagen
- Wheel and rail roughness
- Modal analysis of wheel
- Track decay rate
- Wheel / rail vibration and noise
- Mic array for beamforming
- Near-field array for WSE
- Static and dynamic vibration and noise for ATPA (and MISO)

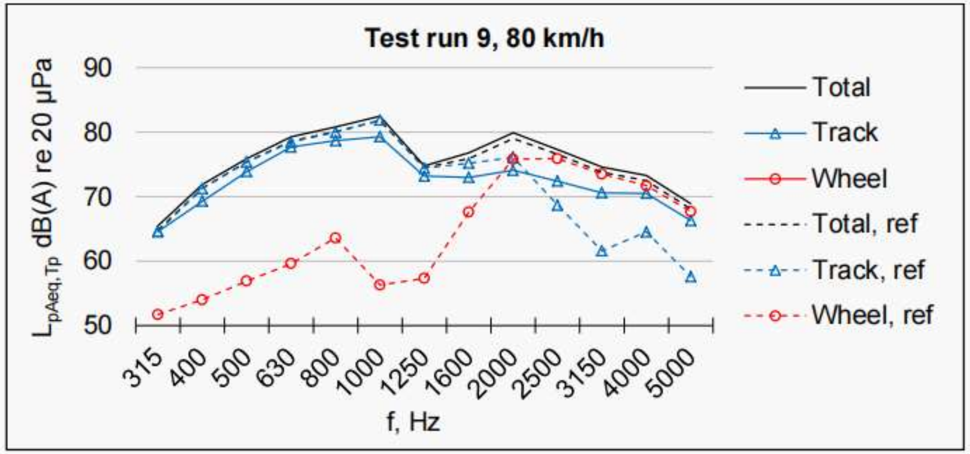
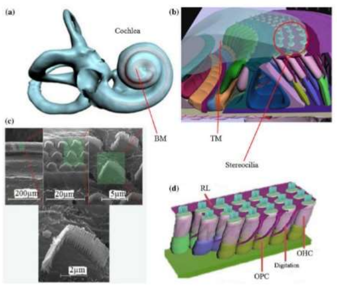


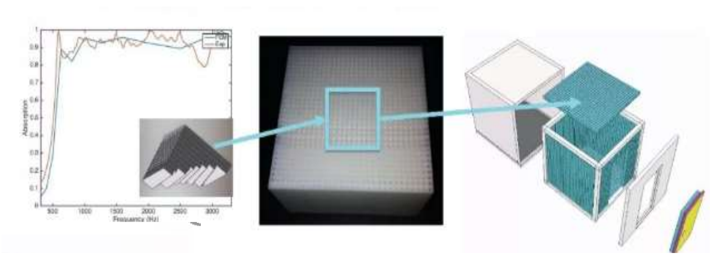
Figure 124. Noise separation result obtained with MISO method compared with reference result, Test run 9 at 80 km/h, Mic 3

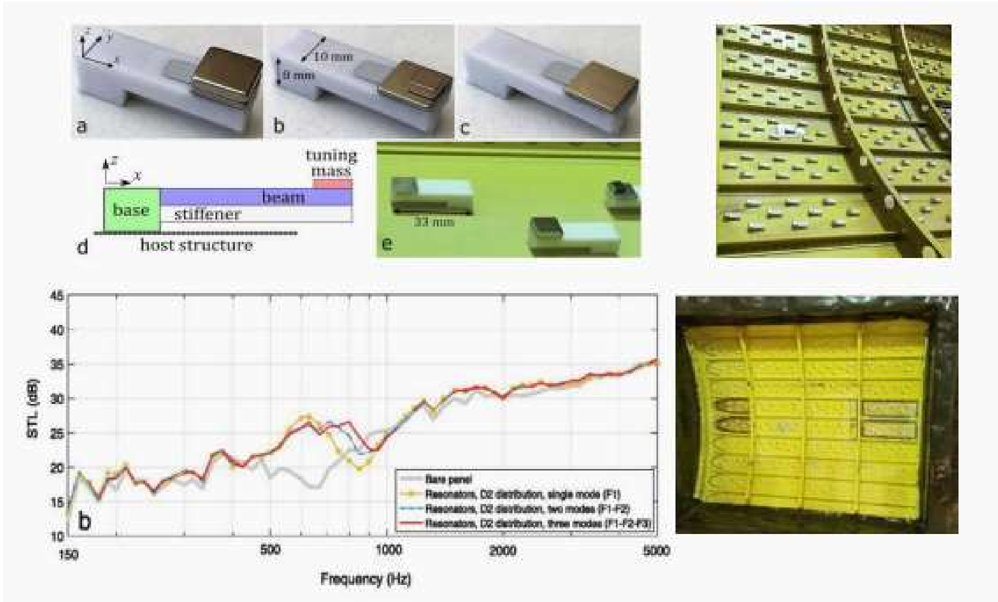
02 Research Progress

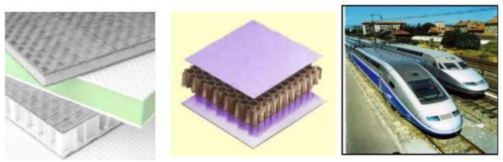
2.3 Canada

- New acoustic insulation materials in aircraft cabin





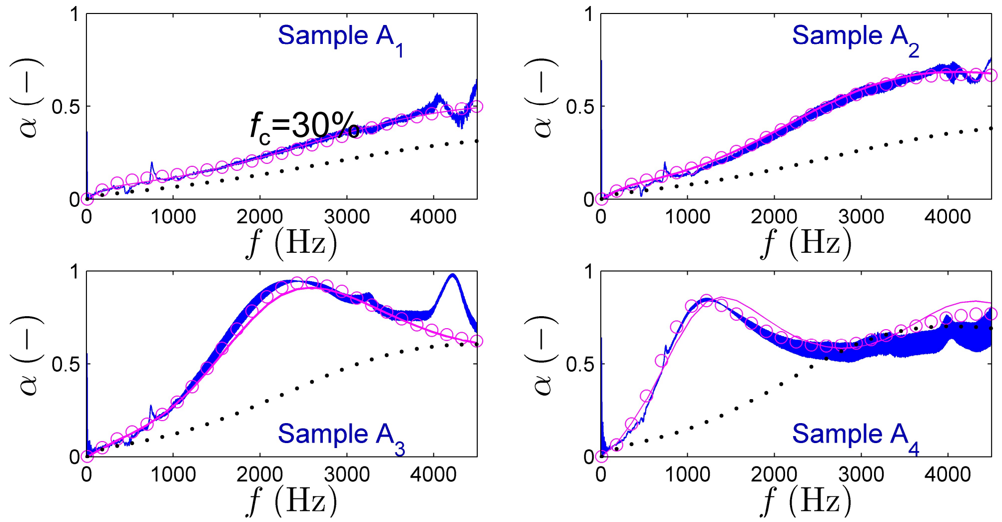




Large aircraft plate for improving TL at ring frequency range , total mass added from 5%-8.5% (Christophe Droz et al, jasa,2019)

2.4 France

Effects of membranes

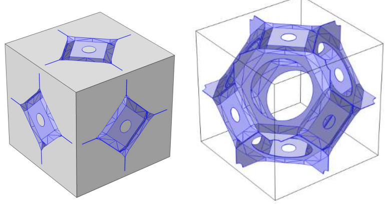


f_c=45%

f_c=65%

f_c=81%

Increasing of the membrane level f_c ↗ from 29%(A_1) to 81 % (A_4) has a strong influence on sound absorption.



By Van Hai Trinh，Camille Perrot；

- New acoustic absorptive materials for low frequency noise control

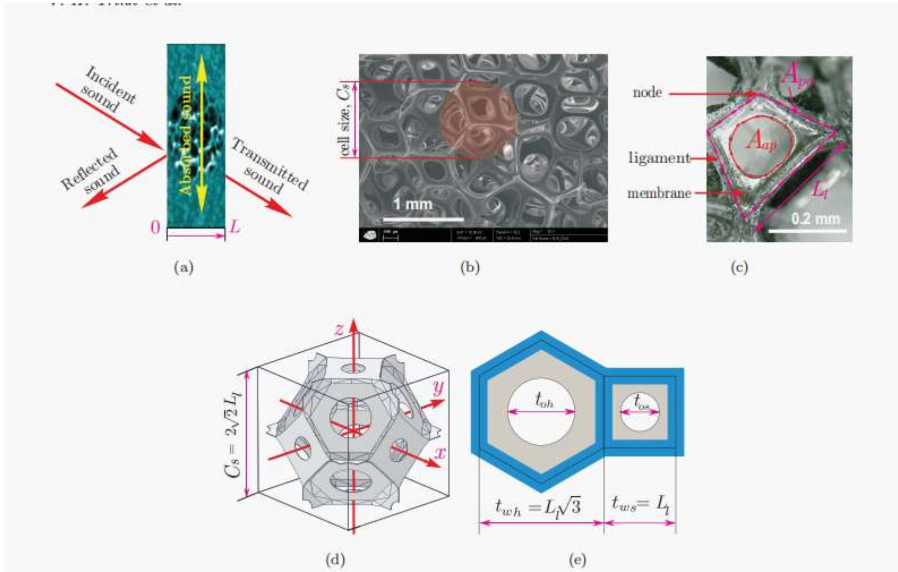


Fig. 1. Modelization of local microstructure of monodisperse foams with membranes. (a) Schematic diagram of a L-thick sound absorber. (b), (c) Morphology characterization of the real foam. (d), (e) Idealized periodic unit cell based on a C_s-size Kelvin model.

- **Vanhai Trinh, Dengke Li, Mu He, Xin Li**, Modeling sound absorption of graded foam absorbers via polynomial surrogate technique, Journal of Theoretical and Computational Acoustics, 2021.

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3.1 Sound absorption

➢ Concept of sound absorption

$$W_i = W_r + W_\alpha + W_t$$

Reflection coefficient $r = \frac{W_r}{W_i}$

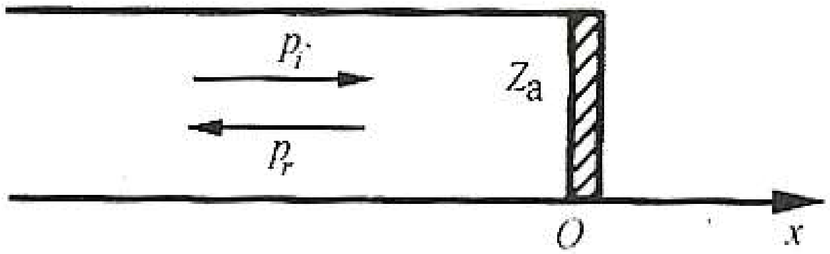
Transmission coefficient $\tau = \frac{W_t}{W_i}$

- Sound absorption coefficient is defined as

$$\alpha = 1 - r = 1 - \frac{W_r}{W_i} = \frac{W_\alpha + W_t}{W_i}, (0 \le \alpha \le 1)$$

3.1 Sound absorption

➢ Concept of sound absorption



$$p_i = p_{ai} e^{j(\omega t - kx)}$$

$$p_r = p_{ar} e^{j(\omega t + kx)}$$

$$r_p = \frac{p_{ar}}{p_{ai}} = |r_p| e^{j\sigma\pi}$$

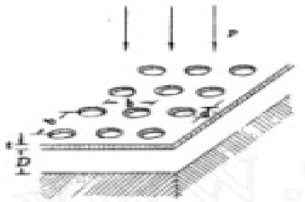
$$\alpha = 1 - r_I = 1 - r_p^{\ 2}$$

$$\alpha = \frac{4r}{(1+r)^2 + x^2}, \qquad r = \frac{R_s}{\rho_0 c_0}, x = \frac{X_s}{\rho_0 c_0}$$

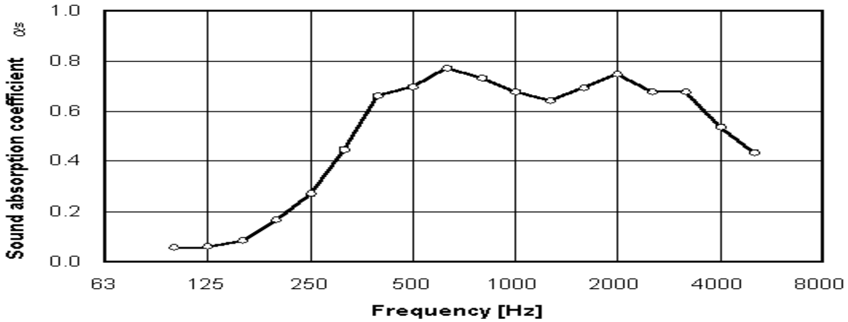
- To achieve a broadband sound absorption coefficient: $r \longrightarrow 1$, $x \longrightarrow 0$ or $(1 + r) < x$
- It is easy to increase damping to overdamping
- Difficult to minimize acoustic impedance

3.1 Sound absorption

● **MPP structure**



d<1mm,
φ=1~3%



Maa D. Y., J. Acoust. Soc. Am. 104(5), 1998.

$$\rho \dot{u} - \frac{\eta}{r_1}\frac{\partial}{\partial r_1}\left(r_1 \frac{\partial}{\partial r_1} u\right) = \frac{\Delta p}{t}$$

$$Z_1 = \frac{\Delta p}{\bar{u}} = j\omega\rho t\left[1 - \frac{2}{x\sqrt{-j}}\frac{J_1\left(x\sqrt{-j}\right)}{J_0\left(x\sqrt{-j}\right)}\right]^{-1}$$

$$Z_1 = \frac{32\rho\mu t}{d^2}\sqrt{1+\frac{x^2}{32}} + j\omega\rho t\left(1+\frac{1}{\sqrt{3^2+\frac{x^2}{2}}}\right)$$

$$r = \frac{32\mu}{\rho c}\frac{t}{d^2}\left[\sqrt{1+\frac{x^2}{32}} + \frac{\sqrt{2}x}{8}\frac{d}{t}\right]$$

$$m = \frac{t}{\rho c}\left[1+\frac{1}{\sqrt{3^2+\frac{x^2}{2}}} + 0.85\frac{d}{t}\right]$$

$$Z_D = -j\rho c \cot\left(\frac{\omega D}{c}\right)$$

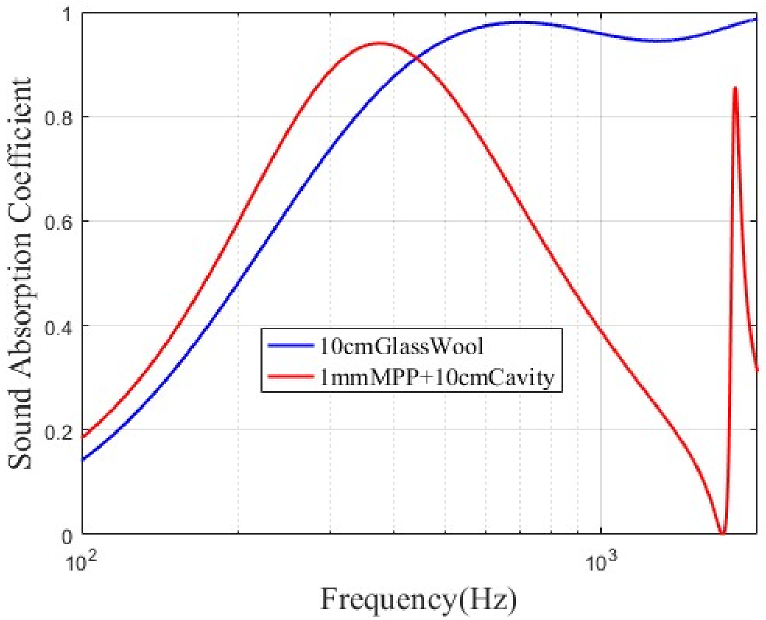
$$\alpha = \frac{4r}{(1+r)^2 + \left(\omega m - \cot\left(\frac{\omega D}{c}\right)\right)^2}$$

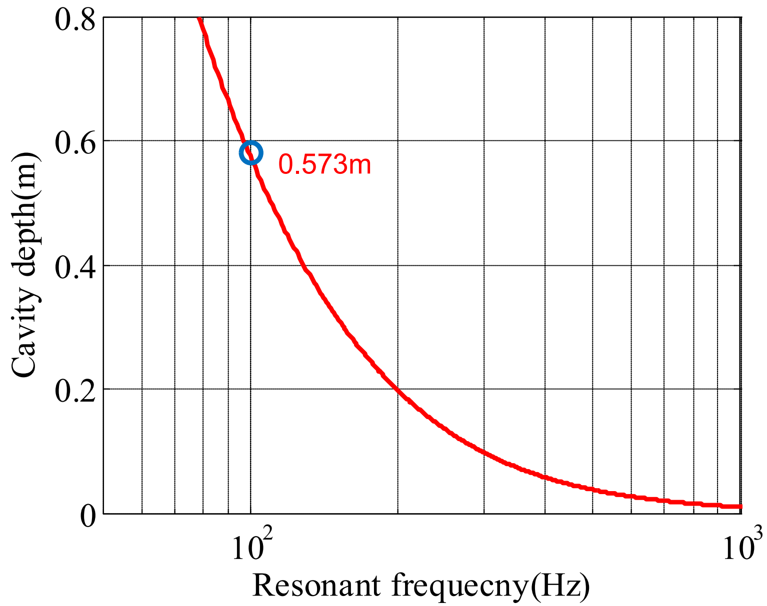
$$r \to 1, \left|\omega m - \cot\left(\frac{\omega D}{c}\right)\right| \to 0 \quad \alpha \to 1$$

$$f\uparrow \propto \sqrt{\frac{p\downarrow}{tD\uparrow}}$$

3.1 Sound absorption

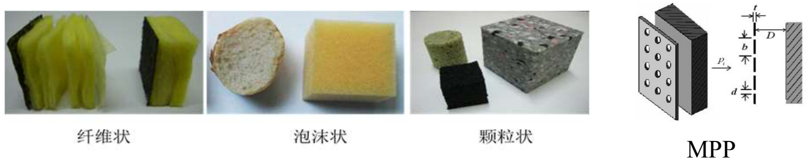
- **MPP structure**





$$f_0 = \frac{1}{2\pi}\frac{1}{\sqrt{\frac{D}{3c}\left(m+\frac{D}{3c}\right)}}$$

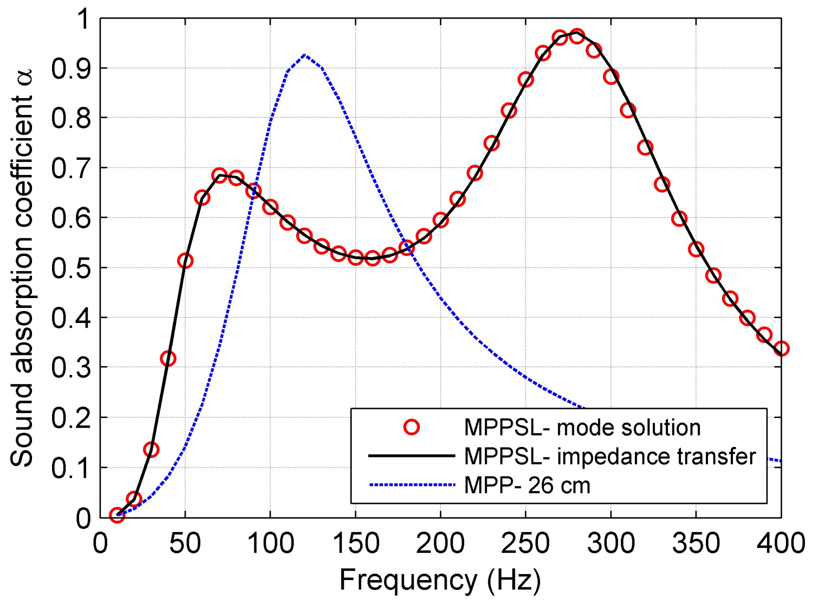
$$\Delta f = \frac{1}{2\pi}\frac{1+r}{\left(m+\frac{D}{3c}\right)}$$

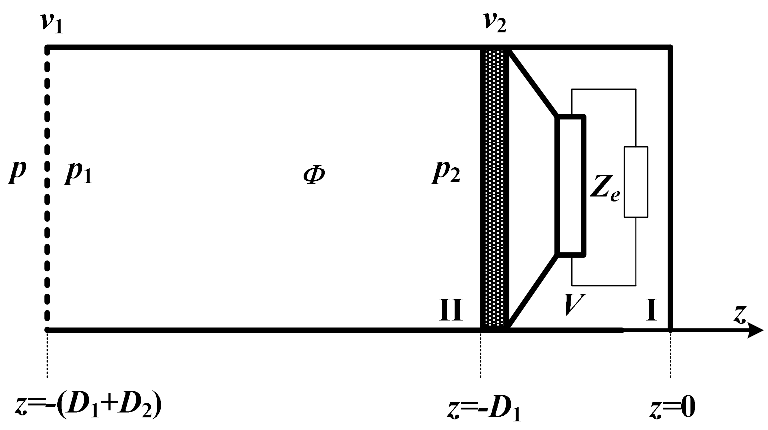


MPP requires a relative thick cavity, and the cavity depth will also narrow the absorption frequency band！

3.1 Sound absorption

● MPP combined structure



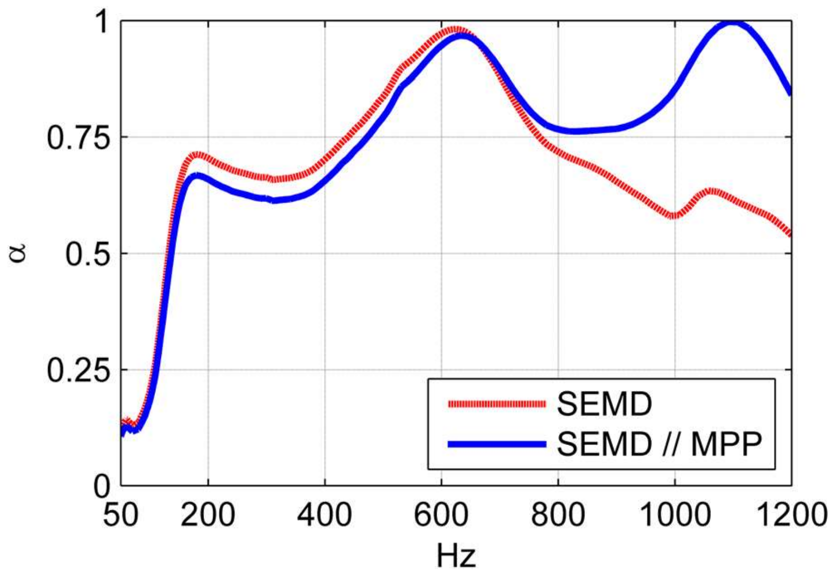


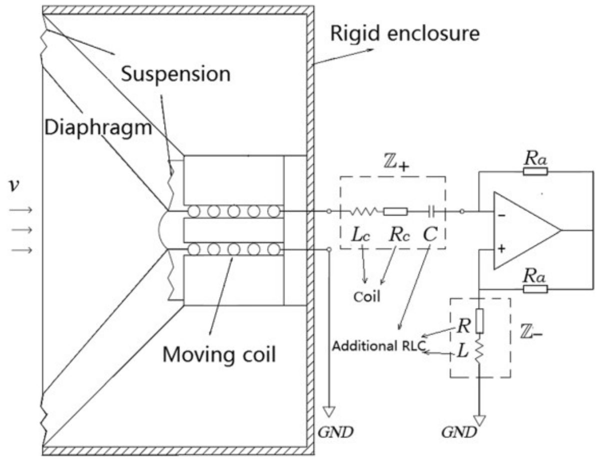
Tao JC, Jing RX, Qiu XJ. Sound absorption of a finite micro-perforated plate backed by a shunted loudspeaker. Journal of Acoustical Society of America 2013;135:231-238.

Acoustic absorption structure with microperforated panel and a bypass circuit in a speaker

3.1 Sound absorption

● MPP combined structure





- Yumin Zhang,Yum-Ji Chan,and Lixi Huang, Thin broadband noise absorption through acoustic reactance control by electro-mechanical coupling without sensor. J Acoust Soc Am 2014;135(5):2738–2745.

MPP combining with electro-acoustical coupled speaker

03 Engineering Application

3.1 Sound absorption

● Sonic crystals

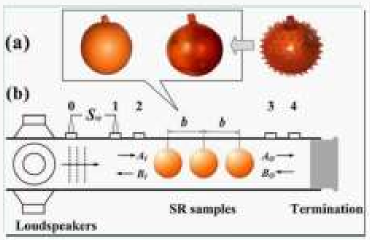
A. Using ‘elastic/soft ball’ to realize resonances (X. Jing, et al, Scientific Reports, 2015)

B. Using ‘Helmholtz Resonator’ to achieve resonances (M. Reynolds et al, ASA meeting,2013)

C. Using ‘membrane and mass’ to achieve cell resonances 2013 (P. Sheng et al, Phys. Rev. Lett. 2008)

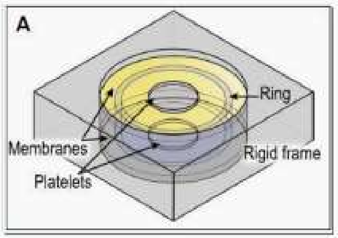
D. Using ‘double membrane and mass’ to achieve multiple resonances (P. Sheng et al, Phys. Rev. Lett. 2013).

E. Using ‘shunt circuit’ to achieve mechanical-circuit resonances(Hao Zhang, et al, Phys. Rev. Lett. 2016)

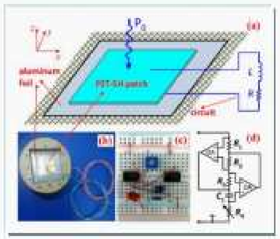


A

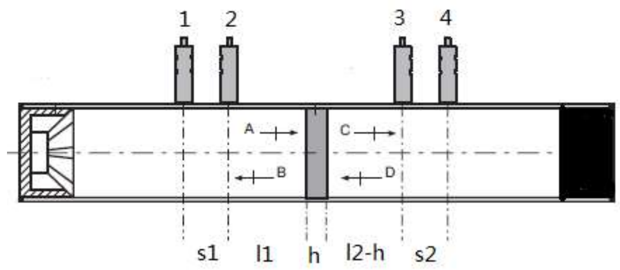
B



C



D

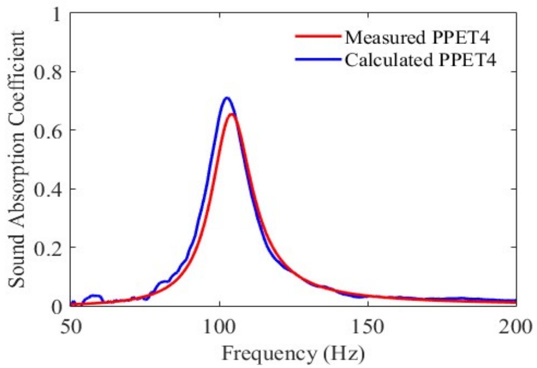


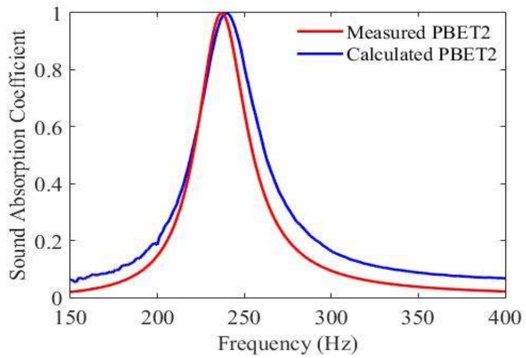
sound absorptive performance are tested by impedance tube

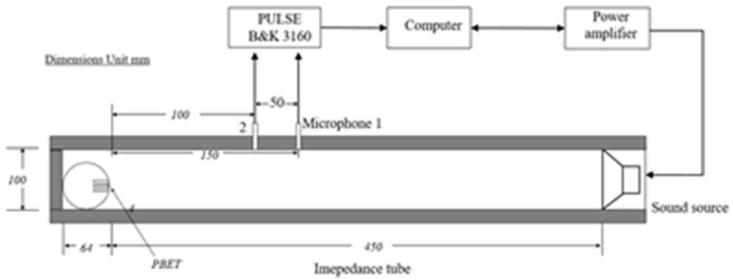
03 Engineering Application

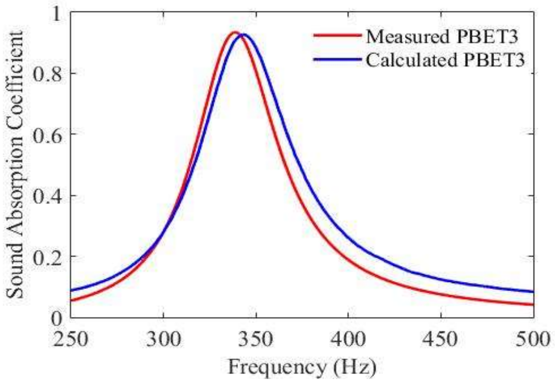
3.1 Resonant acoustic absorber

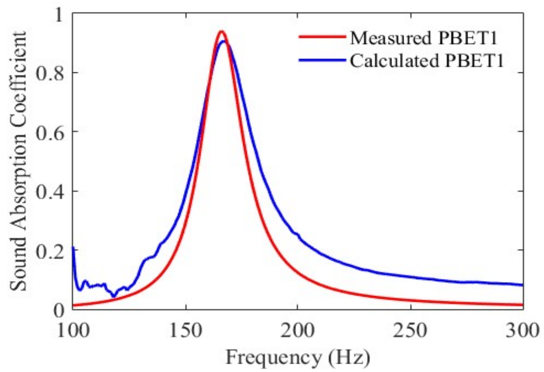
- **PBET for low frequency sound absorption**

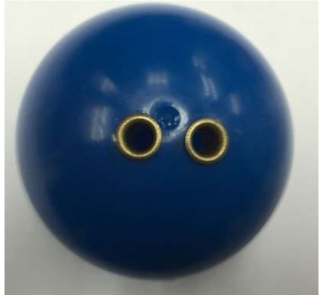










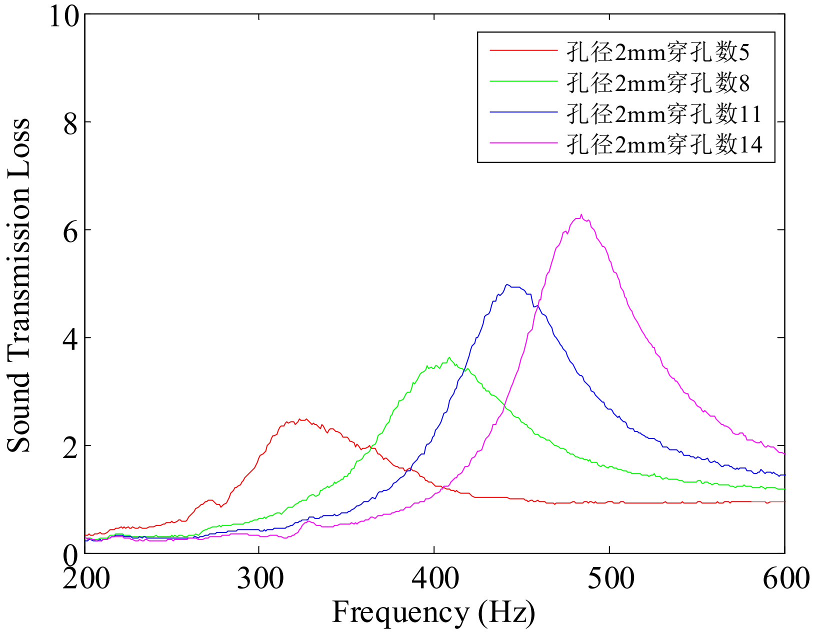


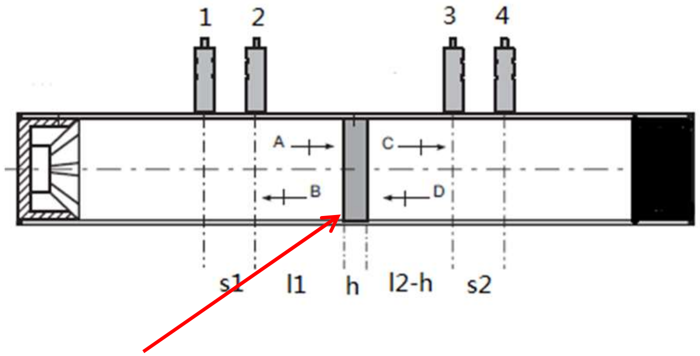
Samples

3.1 Resonant acoustic absorber

● PBET for low frequency sound transmisstion

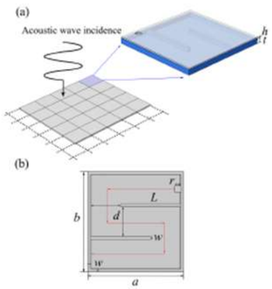
ASTM standard E2611-09: Standard test method for measurement of normal incidence sound transmission of acoustical materials based on the transfer matrix method



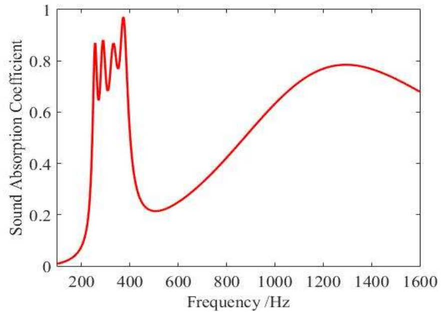


3.1 Resonant acoustic absorber——PBET

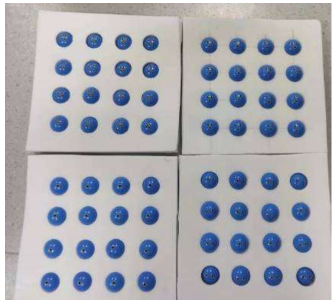
- A thin layer of low frequency noise reduction material
- Wideband noise reduction is achived by multiple resonant element



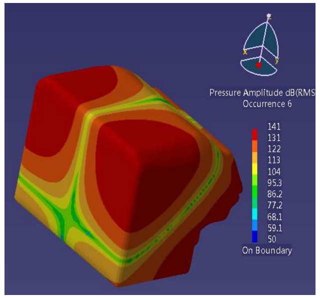
Coupling element



Sound absorption cofficient

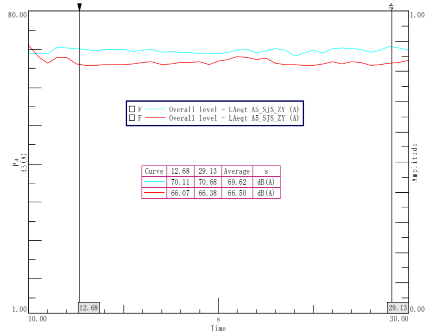


Samples



Low frequency mode inside cabin

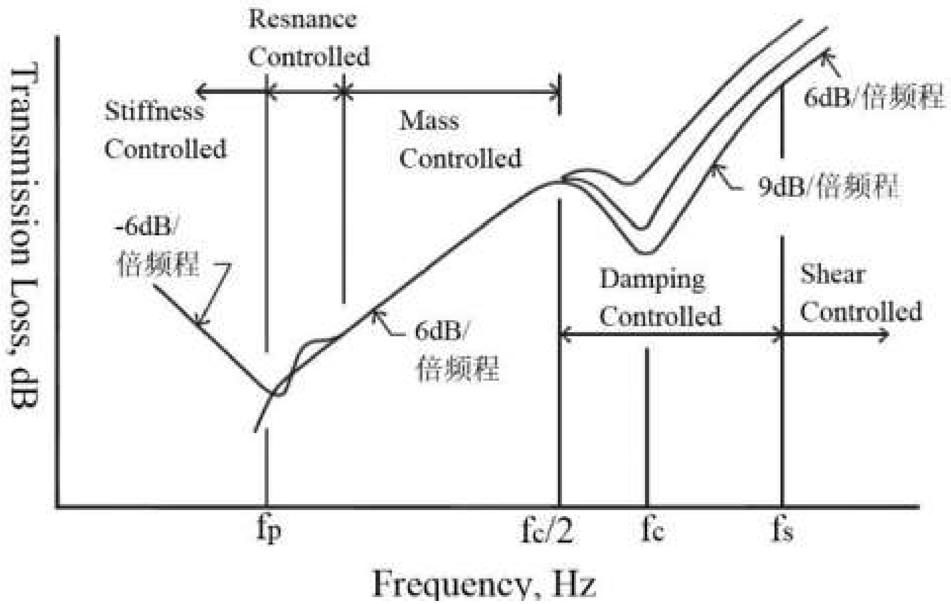
Test in locomotive



Noise reduction performance above 3dB

3.2 Sound transmisson improvement of thin plate

- **Transmission loss & Mass Law**

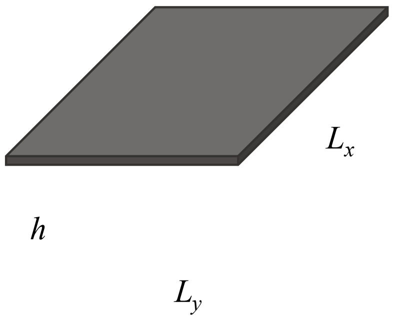


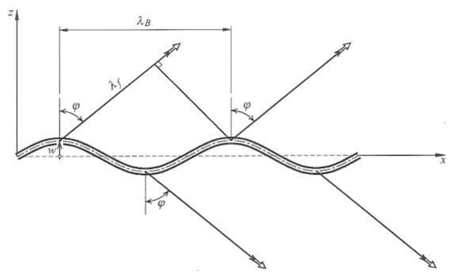
$$TL = 20\log_{10} m + 20\log_{10} f - 20\log_{10}(\rho_0 c/\pi) \quad \text{dB}$$
$$= 20\log_{10}(mf) - 42 \quad \text{dB}$$

To improve the TL without increasing the weight seems an impossible task !

03 Engineering Application

3.2 Sound transmisson improvement of thin plate





$$R = 10\log\left(\frac{1}{\tau}\right) \quad avec \quad \tau = \frac{W_2}{W_1}$$

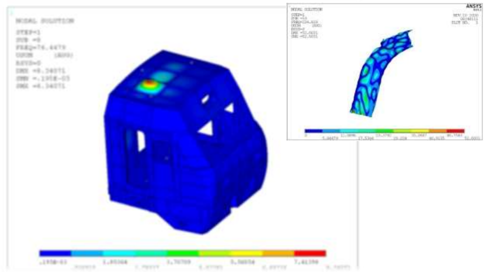
$$W_1 = I_1 S = \frac{\langle p_1^2\rangle_{t,s}}{4\rho_0 c_0} S$$

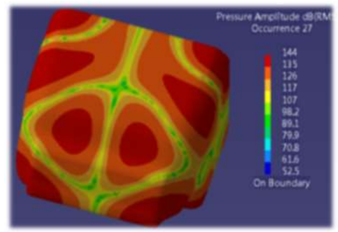
$$W_2 = \frac{\langle p_2^2\rangle_{t,s}}{4\rho_0 c_0} A$$

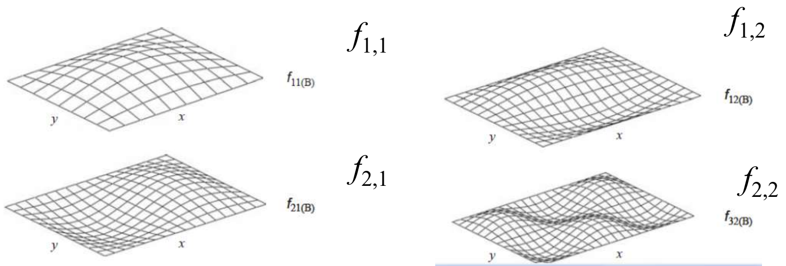
$$Ainsi \quad \frac{1}{\tau} = \frac{W_1}{W_2} = \frac{\langle p_1^2\rangle_{t,s}}{\langle p_2^2\rangle_{t,s}} \frac{S}{A}$$

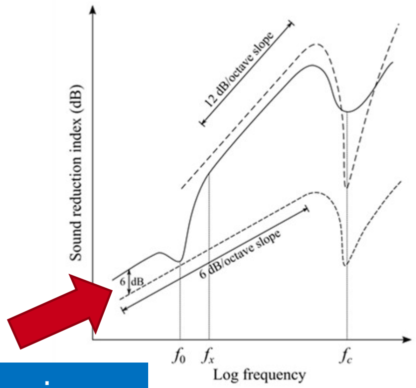
$$R = 10\log\left(\langle p_1^2\rangle\right) - 10\log\left(\langle p_2^2\rangle\right) + 10\log\left(\frac{S}{A}\right)$$

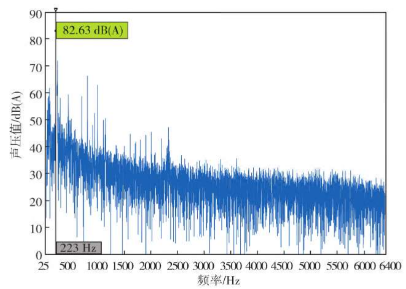
$$R = Lp_1 - Lp_2 + \log\left(\frac{S}{A}\right)$$

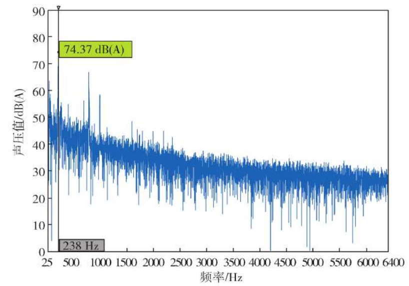












Before

Add damping and stiffener plate

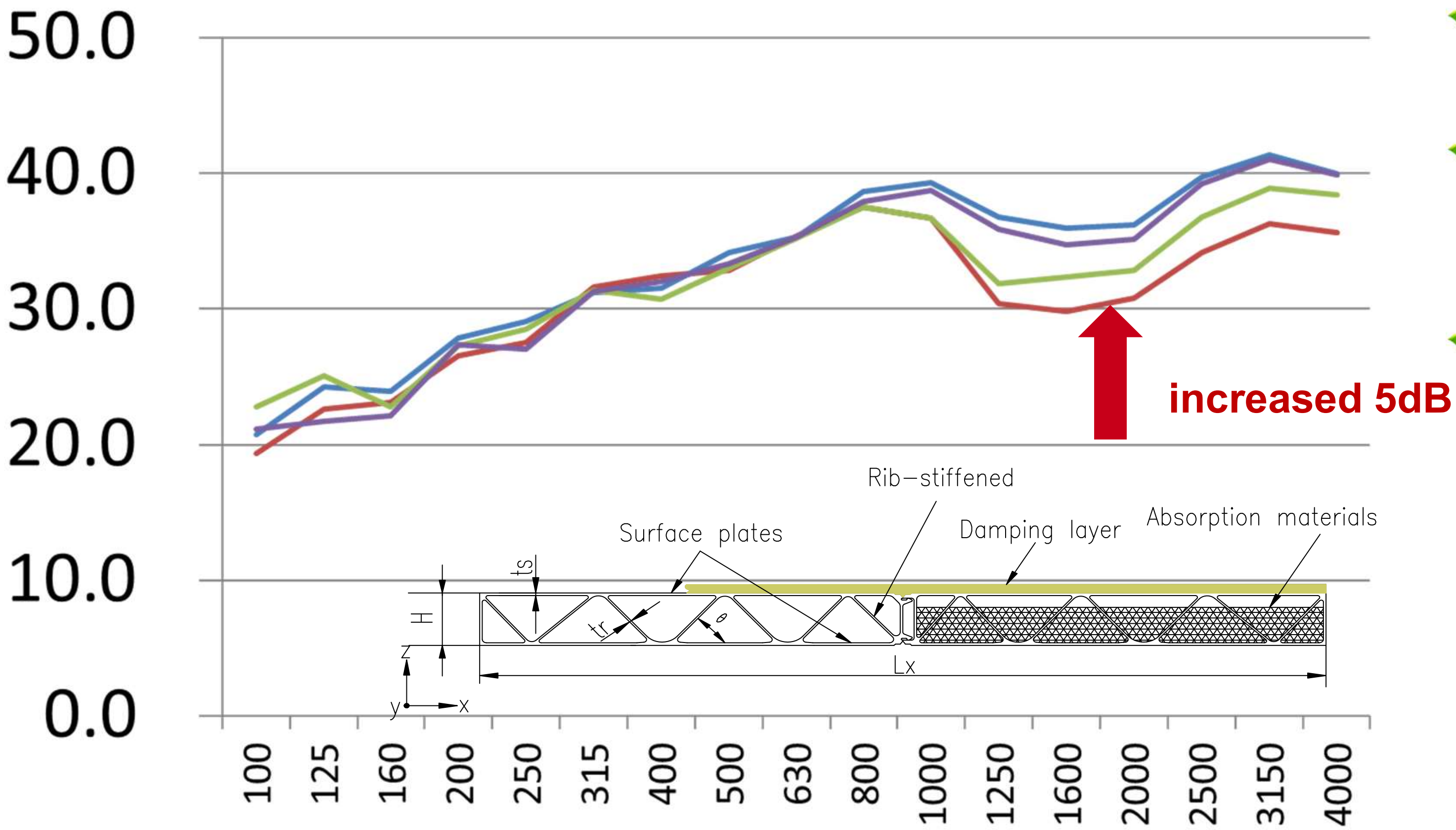
$$f_{m,n} = \frac{c}{2}\sqrt{\left(\frac{m}{L_x}\right)^2 + \left(\frac{n}{L_y}\right)^2}$$

Stiffness&Damping

c: velocity of sound in the material (m / s)
Lx, Ly: plate dimensions (m)
m, n: order of the eigen modes

- Stiffness and damping factors are considered to enhance low frequency transmission
- Multi Mode-frequency coupling response is also studied

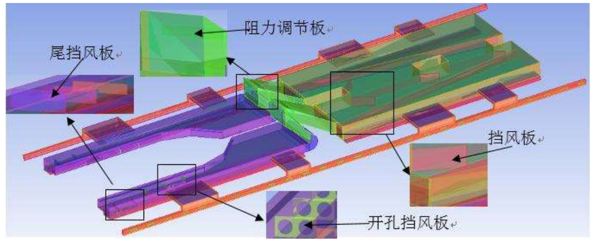
3.3 Sound absorbing materials

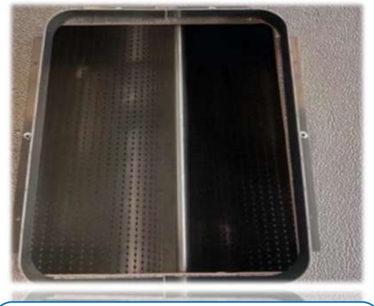


- Design for high frequency sound reduction
- Lightweight acoustic structure design with porous foam material
- The weight of the composite panel increases by approximately 0.2-0.3kg per square meter

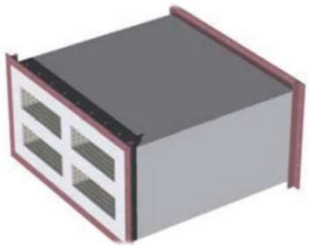
03 Engineering Application

3.4 Acoustic mulffer

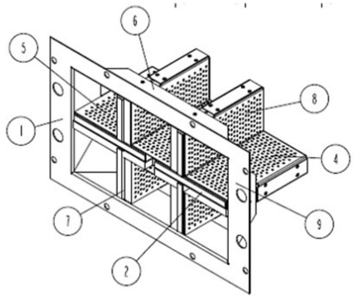




Supply airduct



Return airduct



MPP with sound absorbing materials

- Sound absorbing structures is both designed for **supply and return airduct**
- Noise control performance above 3 dB

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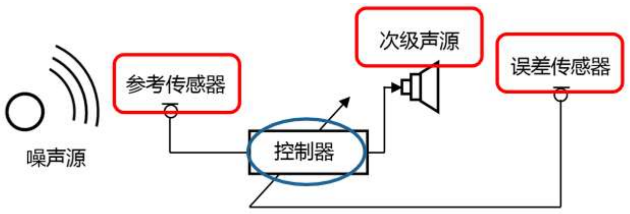
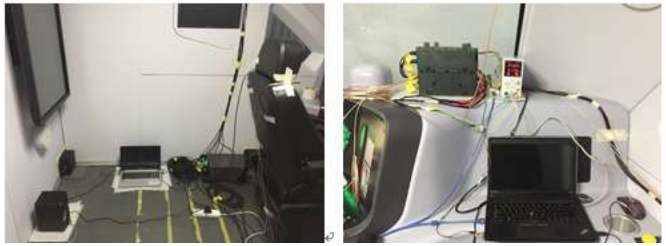
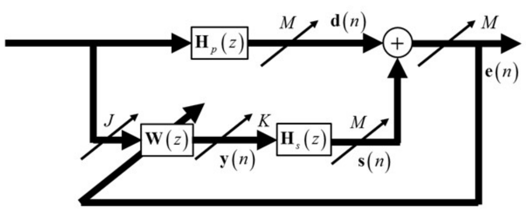
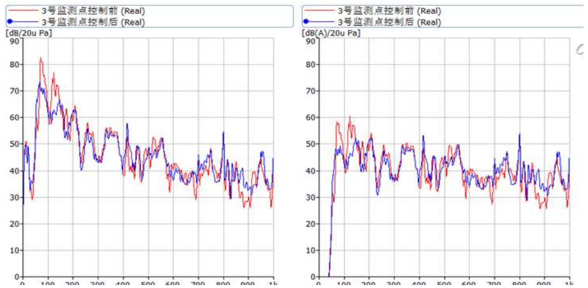
3.5 Active noise control

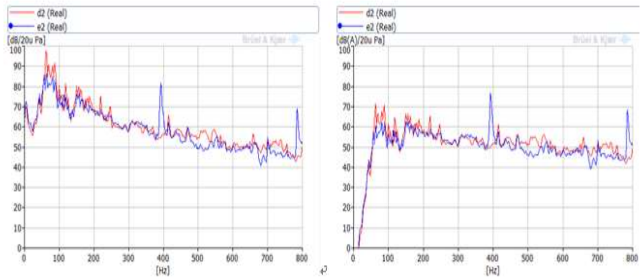
➤ Active noise reduction in locomotive cabin







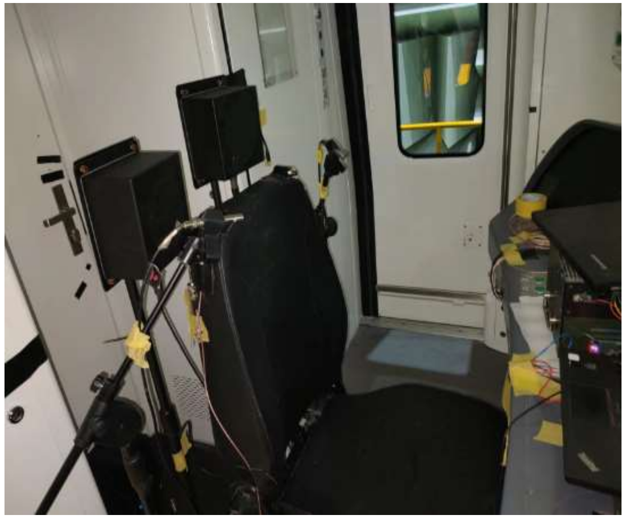


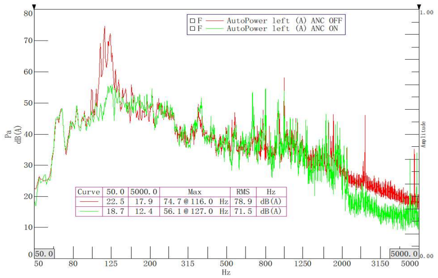


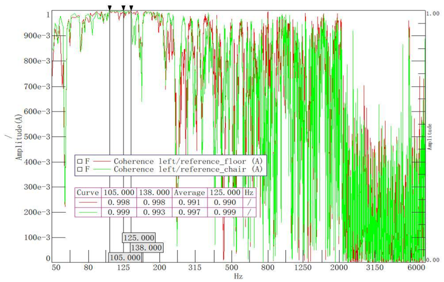
- Both the driver’s ear and standing area are considered
- Noise control performance above 3 dB

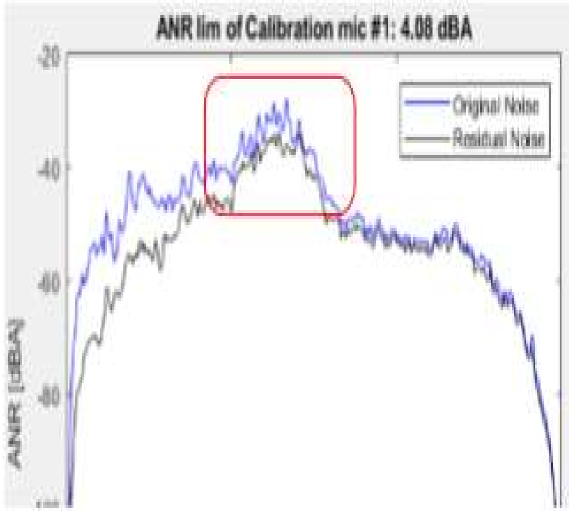
3.5 Active noise control

➢ Active noise reduction in metro cabin





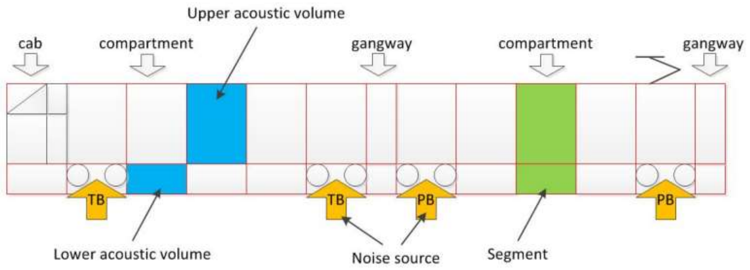


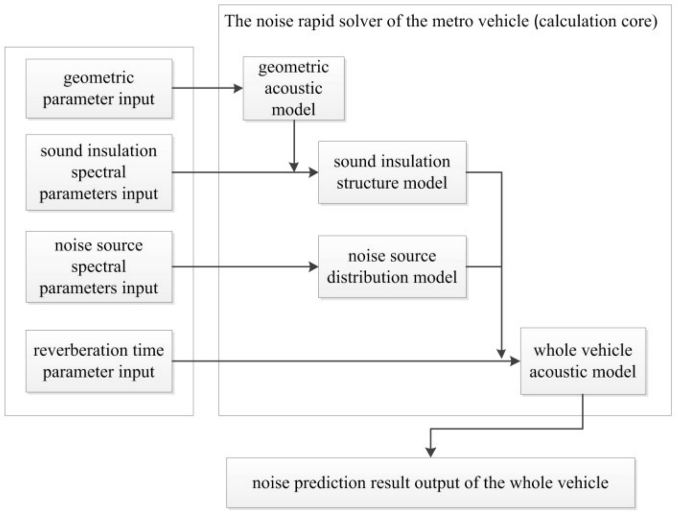


- the near-field active noise reduction reaches 7.6dB (A)
- the far-field active (0.4m) noise reduction reaches 6.0dB (A)

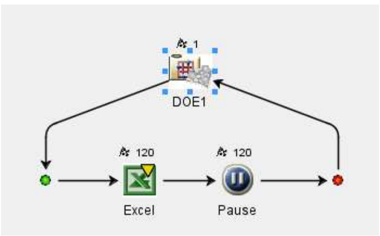
3.6 Noise calculation and optimization

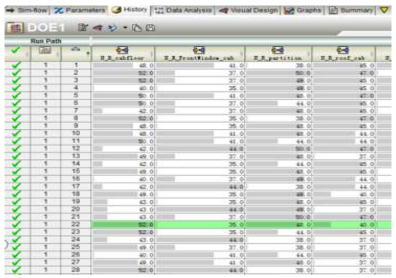
➢ Parameterized vehicle noise rapid solver

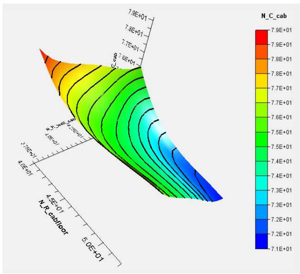


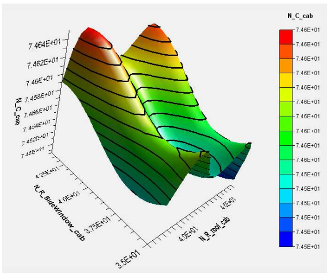


➢ Automatic optimization calculation









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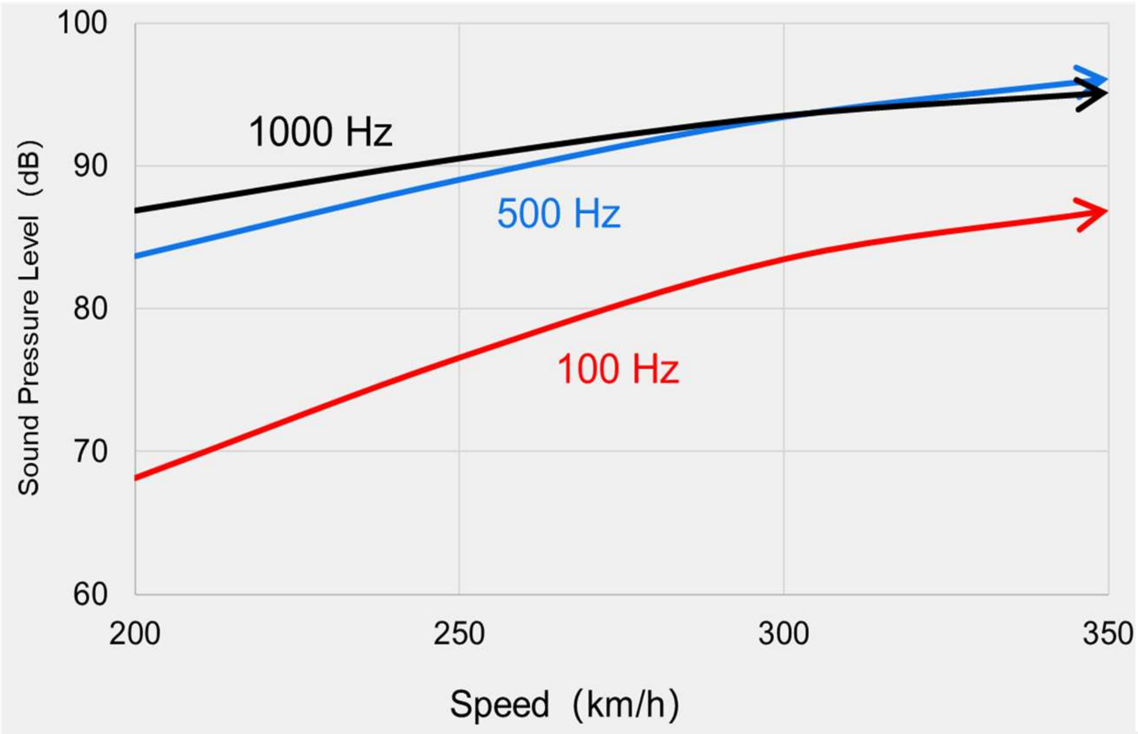
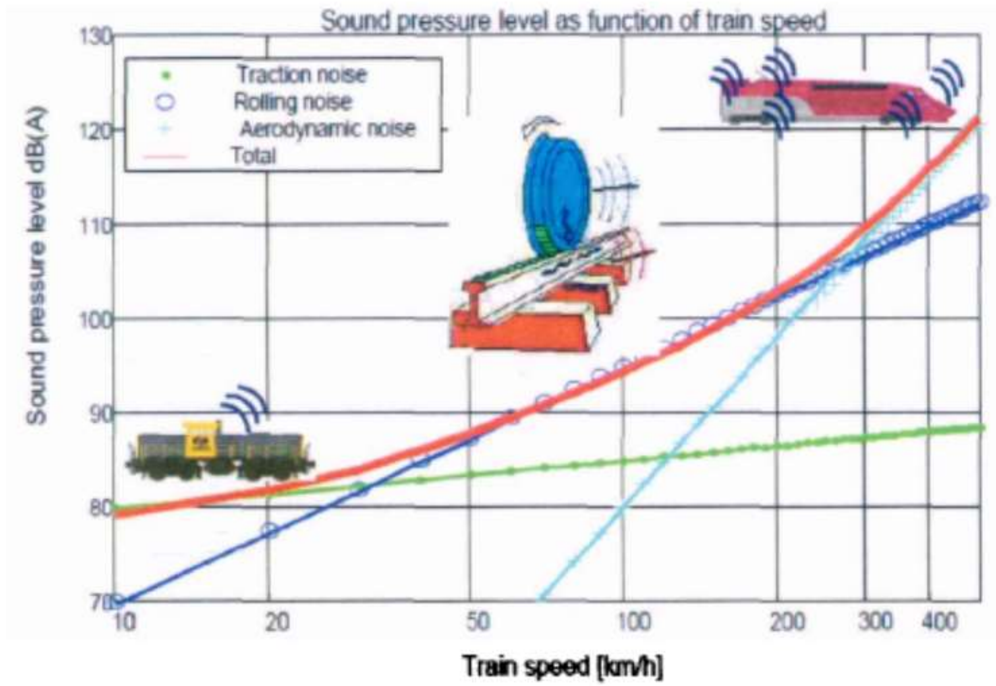
4.1 Wheel-rail noise and aerodynamic noise

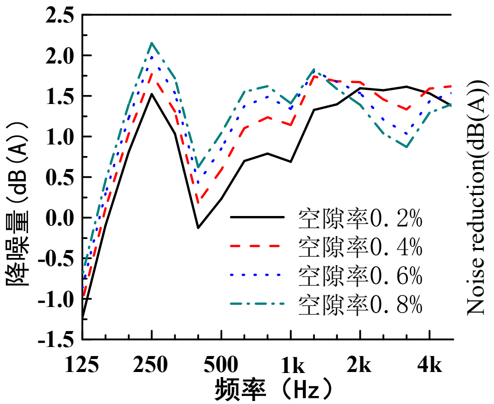




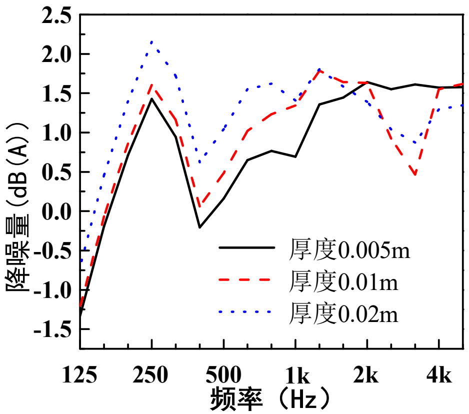
- With the increase of the running speed V, the sound pressure level of the wheel-rail noise and the running speed of the train satisfy 30log10V;
- the sound pressure level of aerodynamic noise and the train running speed satisfy 60log10V.

04 Chanlenges & Oppotunities

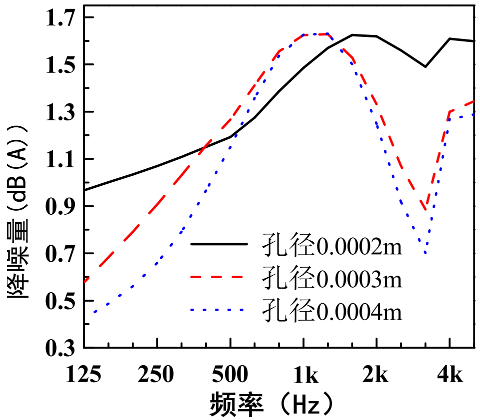
4.1 New sound absorbing materials for wheel-rail noise



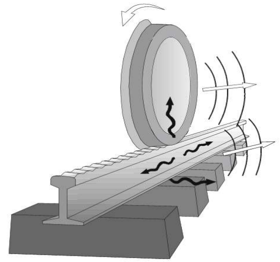
Porosity



Thickness



Pore diameter

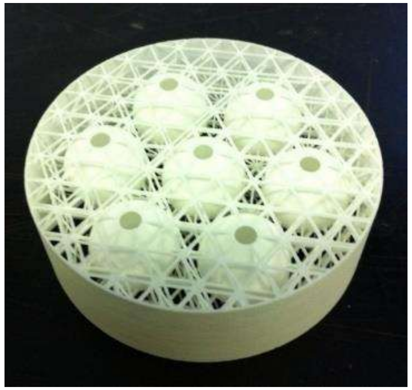


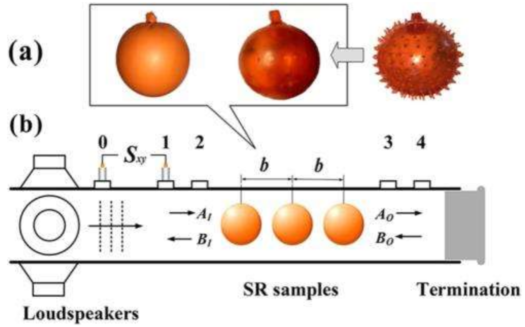
Around 680Hz

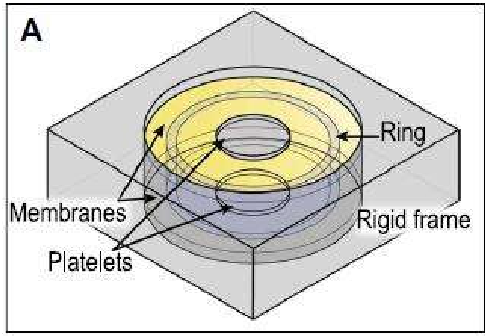
Rail-wheel Noise

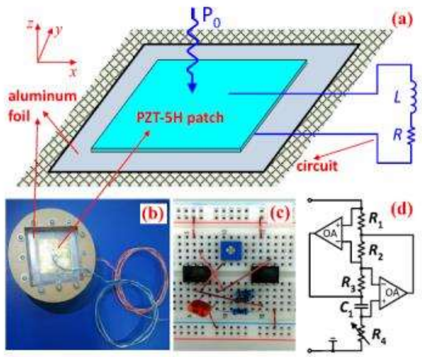
4.2 Acoustic metamaterials

Shark skin resistance reduction material







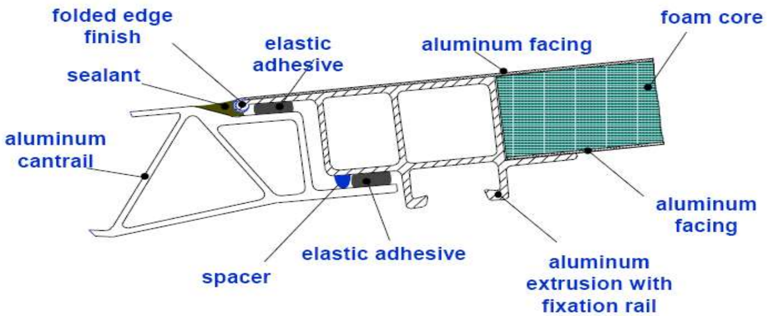


Acoustic metamaterials designed for low frequency sound absorption

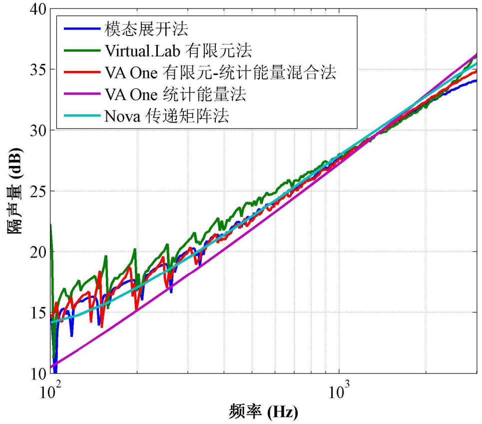
04 Chanlenges & Oppotunities

4.3 Lightweight sound insulation structure

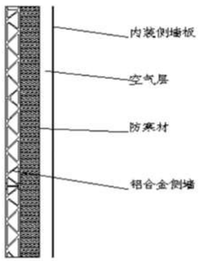
Traditional

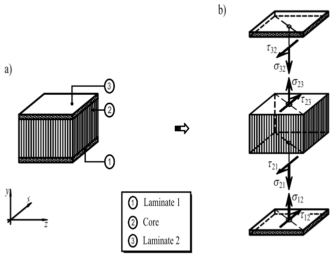


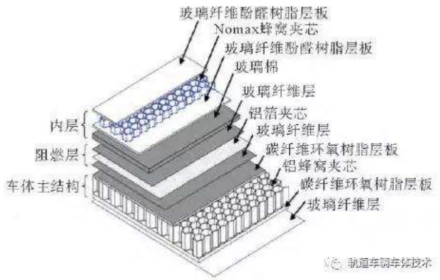
Lightweight

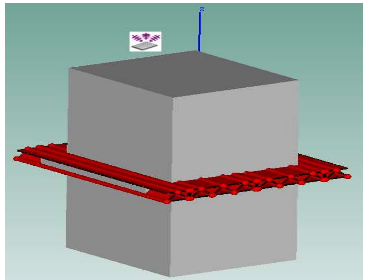


TL









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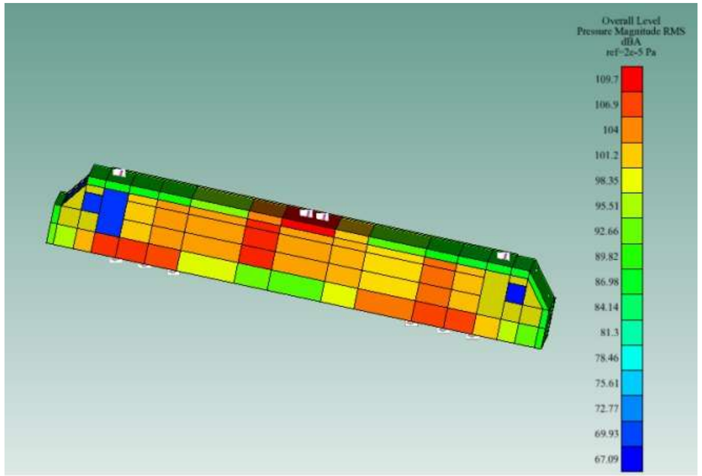
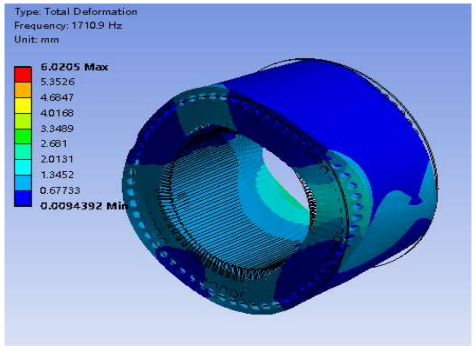
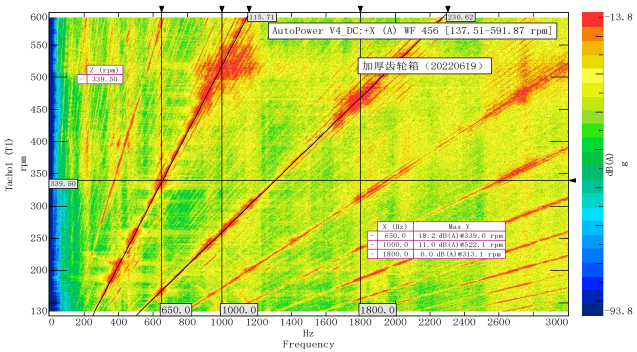
4.4 Low-noise design of gearbox and motor







- Gearbox noise test and optimization
- Motor noise test and optimization

04 Chanlenges & Oppotunities

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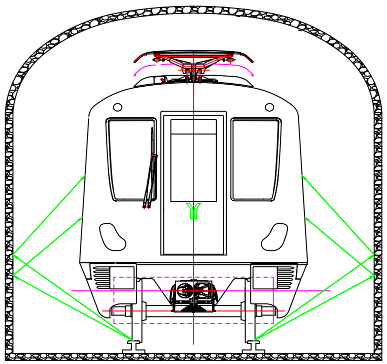
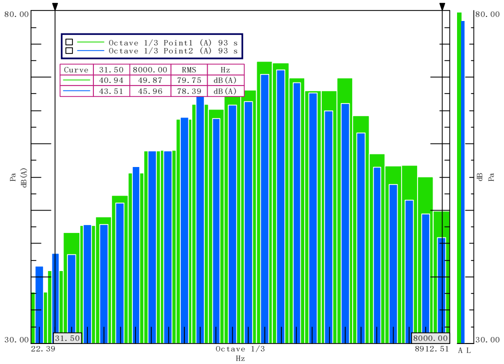
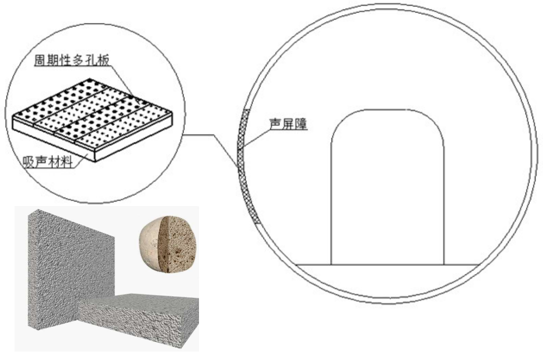
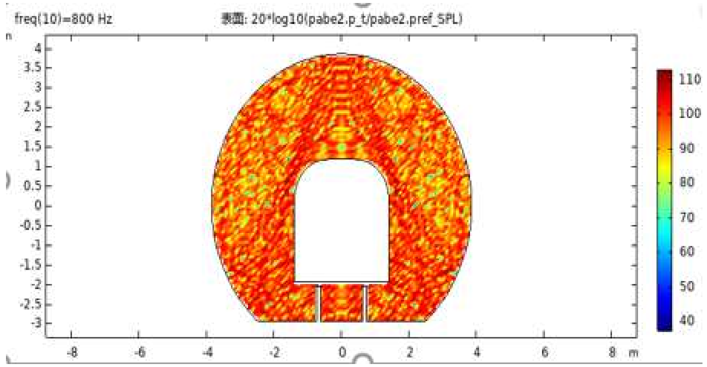
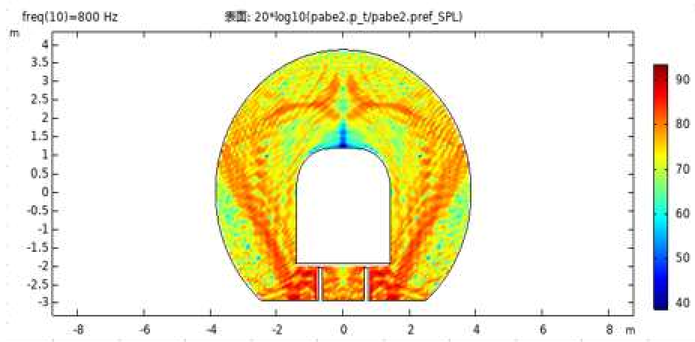
4.5 Noise control in tunnel condition











- New sound absorbing materials to control reverberation time in tunnel
- Design frequency at 200-2000 Hz

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