HELICOPTER GEARBOX PERIODIC STRUT WITH GEOMETRICAL DISCONTINUITY FOR VIBRATION AND NOISE REDUCTION

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Abstract
A novel periodic strut for helicopter cabin noise reduction is presented in this paper. The strut uses mono-material with periodical changing geometry in the longitudinal direction.To design the strut, a theoretical model is built based on the lumped-mass method. On this basis, a parametric analysis is carried out. The results showed that if the static stiffness and the weight are constants, the stop band of the strut is mainly determined by the periodic number and the cell stiffness ratio. The strut can be quickly designed through reducing the periodic number and increasing the cell stiffness ratio to obtain a lower stop band. Subsequently, a sample strut is designed. Its translational and rotational transmissibility are simulated. It is found that except for the designed longitudinal direction, the strut has broadband vibration attenuation characteristics in the lateral direction. However, resonances appear in some frequencies due to the small damping, which would cause noise amplification. In spite of this, with the novel periodic struts, nearly 20dB noise attenuation appears in the cabin of the helicopter model between 500-2000 Hz.

1. INTRODUCTION

The vibration of mid- and high-frequency harmonic tones generated from gear meshing in the main gearbox is one of the dominant sources of helicopter cabin noise [1-3]. The typical frequency range locates between 500 Hz and 2000 Hz which influences human's subjective reaction greatly [1]. This vibration can be carried by the support struts between the main gearbox and the fuselage, and radiate structure-borne noise into the cabin [4-5]. Thus, the cabin noise can be controlled by suppressing the vibration transferred to the fuselage. An effective way is to embed periodic structures into the transmission paths, which have unique dynamic characteristics of restricting wave propagation within specific frequency bands. With proper design, the periodic structures can realize broadband vibration attenuation from the gearbox to the fuselage.

According to the concept, using metal and rubber materials, Szefi et al. [6-9] designed an axial layered isolator for helicopter gearbox with fluid elements embedded in the metal layers. Meanwhile, in 2003, Asiri et al. [10] first proposed the concept of gearbox periodic strut. The experimental results showed that the strut with periodical metal and rubber materials exhibits good broadband attenuation characteristics in the target frequency range [6-10].

However,when the periodic supporting structures were applied between the gearbox and the fuselage, realistic engineering environment should be considered. When in flight, the supporting structure would carry high strength pull-pull alternating loading generated by the rotor [9]. Furthermore, when not in flight, it would support the weight of the gearbox and rotor system. To work normally under this engineering environment, a kind of series/parallel compounded gearbox periodic strut for helicopter cabin noise reduction was proposed by the authors [11-13]. In Ref. 11, the compounded gearbox periodic strut was proposed with modeling and parametric analysis. In Ref. 12, sample design of the developed strut was carried out as shown in Fig. 1. The broadband vibration attenuation characteristics of the strut were experimentally studied together with the effect of

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pre-compression. In Ref. 13, from two aspects of fuselage vibration and cabin noise, under the excitation of a real gearbox as shown in Fig. 2, the efficiency of the compounded periodic struts was verified through simulation and experimental research.

However, this previous structure designed by the authors also brings some problems such as more complex structure, increased weight, difficult manufacturing, and rubber materials easy to aging, etc. Therefore, considering the mechanical properties of metal material and its plasticity in processing and manufacturing, the authors attempt to use mono-material to design a novel gearbox periodic strut to achieve the vibration and noise reduction within the frequency range of 500-2000 Hz. At the same time, the newly developed strut needs to meet the requirements of size, weight, stiffness, and intensity in the background helicopter.

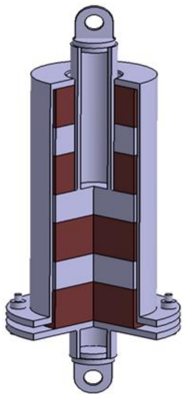


Figure 1. Schematic of the compounded periodic strut.

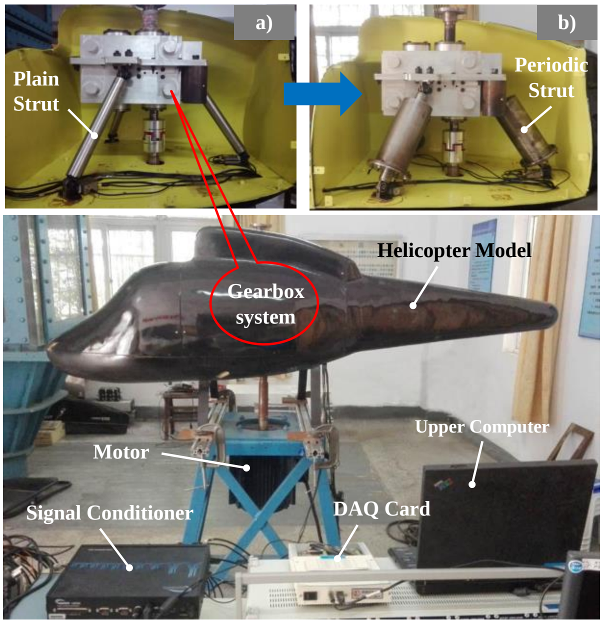


Figure 2. Experiments with the compounded periodic struts

Six sections are included. Section 1 gives a brief introduction. Section 2 introduces the novel periodic strut with geometrical discontinuity. Section 3 presents a brief dynamics model of this strut. On this basis, the influence of the parameters is analyzed in Section 4. A sample periodic strut is further designed under the requirements of a helicopter model, whose broadband vibration and noise attenuation characteristics are verified through simulation in Section 5. Section 6 summarizes the conclusions of the present study.

2. DESIGN OF THE NOVEL STRUT

Under the constraints of the requirements, this paper attempted to design a two-component periodic strut based on Bragg scattering principle. As shown in Fig. 3, using mono-material with periodical changing geometry in the longitudinal direction, the sub-cells A and B of the novel strut have two kinds of impedance characteristics.

To obtain an effective configuration, a large number of attempts were developed based on the previous research results in Ref. 11-13 and finally a configuration was obtained. The detail of the newly developed periodic strut is shown in Fig. 4. In a periodic cell, sub-cell A adopts a hollow metal cylinder structure which has high stiffness and strength characteristics. For sub-cell B, this paper attempts to adopt a new spiral structure, which is composed of a hollow or solid cylinder structure with several spiral grooves removed. When its two ends are stressed, the structure deforms along the spiral grooves. By selecting appropriate parameters, the stiffness and strength characteristics of sub-cell B can be adjusted in a wide range.

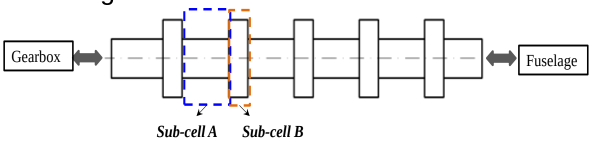
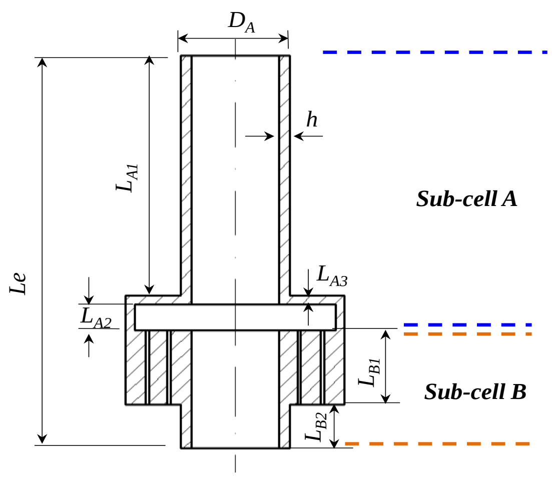


Figure 3. Schematic illustration of a periodic strut with geometrical discontinuity.



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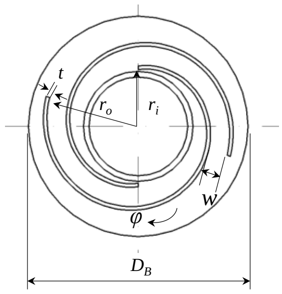

Figure 4. Schematic of the cell of the periodic mono-material strut.

3. MODELING OF THE NOVEL STRUT

3.1. Overview

To guide the design of this metal periodic strut, the dynamics model of the strut was built to obtain the mathematical relationship between the design parameters and the design requirements.

3.2. Dynamics Model

If a sub-cell is considered as a bi-spring-mass system, the dynamical equation of motion for the sub-cells can be described as follows

(1) $$\begin{bmatrix} F_u \\ F_d \end{bmatrix}_A = \begin{bmatrix} k_A - m_{A1}\omega^2 & -k_A \\ -k_A & k_A - m_{A2}\omega^2 \end{bmatrix} \begin{bmatrix} u_u \\ u_d \end{bmatrix}_A$$

(2) $$\begin{bmatrix} F_u \\ F_d \end{bmatrix}_B = \begin{bmatrix} k_B - m_{B1}\omega^2 & -k_B \\ -k_B & k_B - m_{B2}\omega^2 \end{bmatrix} \begin{bmatrix} u_u \\ u_d \end{bmatrix}_B$$

where F and u define the force and displacement with subscripts u and d denoting the top and bottom sides of the structure. In addition, m_{A1}, m_{A2}, and k_A define the equivalent mass and stiffness of the sub-cell A; m_{B1}, m_{B2}, and k_B define the equivalent mass and stiffness of the sub-cell B.

In Eqs. (1) and (2), the mass has the relations to the geometric parameters as shown in Fig. 4

(3a) $$m_{A1} = \rho\left(A_1 L_{A1} + A_3 L_{A3} / 2\right)$$

(3b) $$m_{A2} = \rho\left(A_2 L_{A2} + A_3 L_{A3} / 2\right)$$

(3c) $$m_{B1} = \rho\left(A_3 - \pi n \varphi t \frac{\left(r_o^2 - r_i^2\right)}{\left(r_o - r_i\right)}\right) L_{B1} / 2$$

(3d) $$m_{B2} = m_{B1} + \rho A_1 L_{B2}$$

where $A_1 = \pi\left(D_A h - h^2\right)$, $A_2 = \pi\left(D_B h - h^2\right)$, $A_3 = \pi\left(D_B{}^2 - \left(D_A - 2h\right)^2\right) / 4$ are the cross-section areas of the minor-diameter cylinder, major-diameter cylinder, and the annular plate. ρ is the density of the mono-material.

Meanwhile, the stiffness can be calculatd using the experience formulas in Ref. 15.

Then, the transfer matrix of a sub-cell A can be obtained through the matrix transforming of Eq. (1)

(4)
$$\boldsymbol{T}_A = \begin{bmatrix} k_A{}^{-1}(k_A - m_{A1}\omega^2) & -k_A{}^{-1} \\ k_A - k_A{}^{-1}(k_A - m_{A2}\omega^2)(k_A - m_{A1}\omega^2) & k_A{}^{-1}(k_A - m_{A2}\omega^2) \end{bmatrix}$$

In the same way, we can get the transfer matrix of a sub-cell B which is defined as $\boldsymbol{T}_B$.

Then, the transfer matrix of a cell can be obtained

(5) $$\boldsymbol{T}_{cell} = \boldsymbol{T}_B \boldsymbol{T}_A$$

Finally, the transfer matrix of a periodic strut with N cells can be obtained

(6) $$\begin{bmatrix} u_d \\ F_d \end{bmatrix}_s = \left(\boldsymbol{T}_{cell}\right)^N \begin{bmatrix} u_u \\ F_u \end{bmatrix}_s = \begin{bmatrix} t_{11} & t_{12} \\ t_{21} & t_{22} \end{bmatrix} \begin{bmatrix} u_u \\ F_u \end{bmatrix}_s$$

where the subscript s denotes the parameter of the strut and N define the periodic number of the strut.

To obtain the stop band of the periodic strut, it is assumed that the propagation parameter is μ and

(7a) $$u_d = e^{\mu} u_u$$

(7b) $$F_d = e^{\mu} F_u$$

Substituting Eqs. (7) into Eq. (6), we have

(8) $$\cosh \mu = \left(t_{11} + t_{22}\right) / 2$$

The propagation parameter of the strut can be calculated by Eq. (8). However, when the strut is designed, the total mass M and the static stiffness K of the strut are usually limited where we have

(9) $$M = N\left(m_{A1} + m_{A2} + m_{B1} + m_{B2}\right)$$

(10) $$\frac{1}{K} = N\left(\frac{1}{k_A} + \frac{1}{k_B}\right)$$

In addition, several proportional coefficients of the cell as well as the sub-cell were introduced to represent the stiffness and mass distribution in a cell, including cell stiffness ratio $r_k = k_A / k_B$, cell mass ratio $r_m = \left(m_{A1} + m_{A2}\right) / \left(m_{B1} + m_{B2}\right)$, sub-cell

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A mass ratio $r_A = m_{A1} / m_{A2}$, and sub-cell B mass ratio $r_B = m_{B1} / m_{B2}$.

Substituting these four proportional coefficients to Eq. (8), we can get the relationship between these coefficients and the propagation parameter

$$\cosh \mu = 1 - \frac{1}{2N^2}\Omega + \frac{1}{2N^4}\left(\frac{\Delta_1}{\Delta_2}\right)\Omega^2 \tag{11}$$

where

$$\Delta_1 = \left[r_m\left(r_B+1\right)+r_B\left(r_A+1\right)\right]\left[r_m r_A\left(r_B+1\right)+\left(r_A+1\right)\right] \tag{12}$$

$$\Delta_2 = r_k^{\,2}\left(\frac{1}{r_k}+1\right)^2\left(r_m+1\right)^2\left(r_A+1\right)^2\left(r_B+1\right)^2 \tag{13}$$

and

$$\Omega = \frac{M}{K}\omega^2 \tag{14}$$

is the dimensionless frequency parameter of the strut.

Then, the frequency range of the first stop band can be deduced, including the beginning stop-band frequency (BF) and the end stop-band frequency (EF)

$$\Omega_b = \frac{N^2\Delta_2}{2\Delta_1}\left(1-\sqrt{1-16\frac{\Delta_1}{\Delta_2}}\right) \tag{15a}$$

$$\Omega_e = \frac{N^2\Delta_2}{2\Delta_1}\left(1+\sqrt{1-16\frac{\Delta_1}{\Delta_2}}\right) \tag{15b}$$

4. DESIGN OF THE SAMPLE STRUT

4.1. Overview

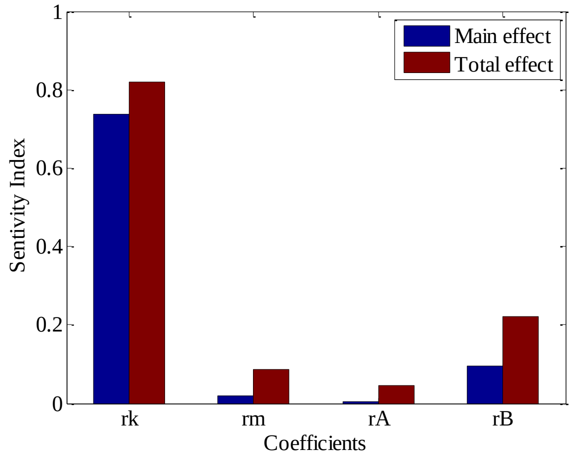
Based on the model shown above, parameters influence is studied, which is conducive to select and adjust the design parameters. Furthermore, a sample strut was designed with a set of suitable material and geometric parameters.

4.2. Parameters influence

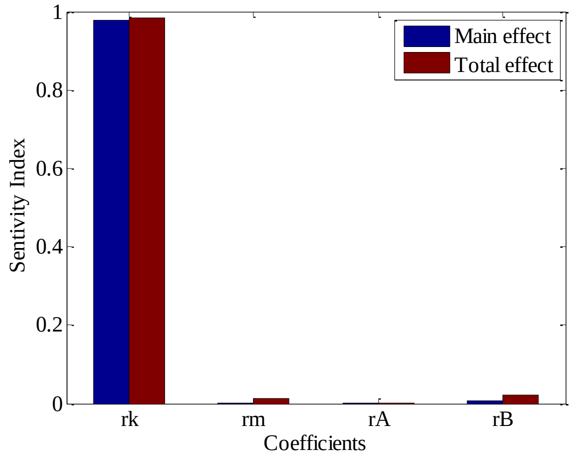
As shown in Eq. (15), if the total stiffness K and the total mass M are constants, the stop band of the strut is directly determined by the periodic number and the proportional coefficients of the cell as well as the sub-cell. The less periodic number is needed to meet the low-frequency vibration attenuation requirement.

For the proportional coefficients, coupling between r_k, r_m, r_A, and r_B are found based Eqs. (12) and (13). Therefore, the sensitivity analysis is carried out to find out the dominant factors for the stop band of the novel strut. Fig. 5 shows the sensitivity index of r_k, r_m, r_A, and r_B to the BF and EF of the first stop band, respectively. It can be seen from the figure that, regardless of first-order sensitivity or global sensitivity, r_k plays an absolutely dominant role on the stop band frequencies. Therefore, it can be used as the main design parameter in the design process.

Based on the research results in Ref. 15, under the constraint of the static stiffness and the weight, with the increase of r_k, the BF decreases gently while the EF increases nearly linearly, leading to an obvious increase of the stop-band width. Therefore, it is necessary to select the cell stiffness ratio as large as possible to achieve vibration attenuation in the targeted frequency range.



(a) Beginning frequency



(b) End frequency

Figure 5. The sensitivity of the proportional coefficients on the stop band frequencies.

Hence, if the static stiffness and the weight of the strut are constants in the design process, we can quickly design the strut through choosing the

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suitable periodic number and cell stiffness ratio. To obtain a lower stop band, we can achieve the parameters by reducing the periodic number and increasing the cell stiffness ratio.

4.3. Design fo the Sample Strut

Based on the above parametric study, a sample strut was further designed based on the detail design requirements of the background helicopter. Actually, the design requirements are the same as those of the compounded gearbox periodic strut in Ref. 13.

To meet the requirements, high strength aviation alloy steel 30CrMnSi is chosen whose yield strength is 885×106Pa. Its material properties include elasticity modulus $E=210\times10^{10}$Pa, density $\rho=7860$kg/m^3, and Poisson's ratio v=0.3. In addition, a set of appropriate geometric parameters is shown in Table 1. The periodic number N is 2.

With the chosen parameters, the weight of the novel strut is 1.875kg meeting the weight requirement, meanwhile, the maximal length (220 mm) and diameter (80 mm) meet the space requirement.

Table 1. Geometric dimensions of the sample strut.

Parameters	Values	Parameters	Values
D_A /D_B	30mm/ 80 mm	L_{B2}	40 mm
Le	110 mm	r_i/ r_o	16 mm/ 35 mm
L_{A1}	40 mm	h	3 mm
L_{A2}	5 mm	t	1.5 mm
L_{A3}	5 mm	φ	0.86
L_{B1}	20 mm	n	5

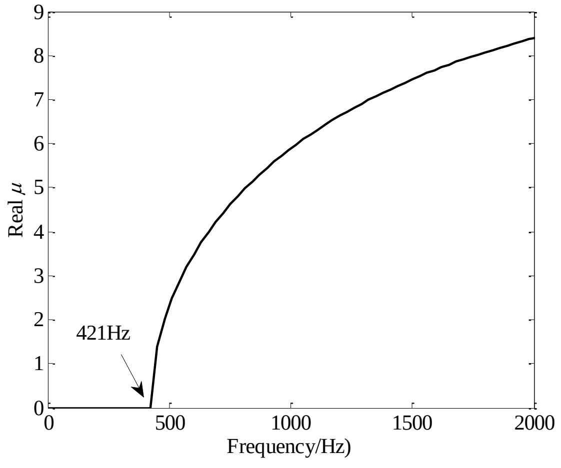

Figure 6. The longitudinal attenuation coefficient of the novel periodic strut.

The parameters of the strut were then submitted to the dynamical model to get the real part of the vibration propagation coefficient as presented in Fig. 6. The curve indicates the attenuation zone and attenuation level in the longitudinal direction of the strut. As the curves suggest, the novel periodic strut presents vibration attenuation between 421-2000 Hz. As the frequency increasing, the attenuation amplitude increases rapidly.

5. VIBRATION AND NOISE PERFORMANCE

5.1. Overview

To further observe the vibration and noise attenuation characteristics of the novel strut, a finite element analysis tool was used to calculate the vibration and noise level in the stop band.

5.2. Vibration Characteristics of a Strut

For the supporting strut of the gearbox, the longitudinal translation direction is the dominant vibration transmission path, which is thus the design direction of the stop band. To verify the design result, the longitudinal displacement transmissibility of the strut was calculated. As shown in Fig. 7, the vibration through the strut could be attenuated between 541-2221 Hz and be amplified at 181 Hz and 481 Hz. The stop band is a little different from the theoretical prediction. This is because, in the analysis tool, the strut is a 3D structure which is different from the theoretical model in Section 3.

However, as the curves suggest, the novel periodic strut still presents excellent broadband vibration attenuation characteristic in the first stop band, where the maximum reduction can exceed 40 dB.

Meanwhile, transmissibility at other directions was also calculated to monitor the vibration performance of the strut. The vibration directions include lateral translation, lateral rotation, and longitudinal rotation as shown in Figs. 8 and 9.

It can be seen from the figures that the novel strut also has different stop bands in other vibration transmission paths. For example, in the direction of longitudinal rotation, the stop band has a range of 511-1651 Hz. Compared with that in the translational direction, the BF is the same while the EF is much lower. This finding is a result of the stiffness of the supporting strut in the torsion direction which is limited by the shear modulus and polar moment of inertia, however, the longitudinal translation stiffness is determined by the elasticity modulus and the cross-sectional area. For a normal structure, like a cylindrical structure, the torsional stiffness is usually smaller than the tensile stiffness. However, for the novel strut, the spiral structure has shear deformation like that with a torque when it carried a

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longitudinal force. Hence, the special working form made the strut has a special transfer performance.

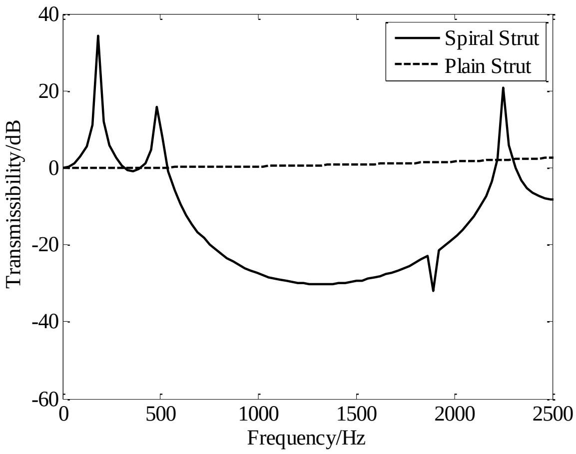


Figure 7. Longitudinal translation transmissibility of the novel periodic strut.

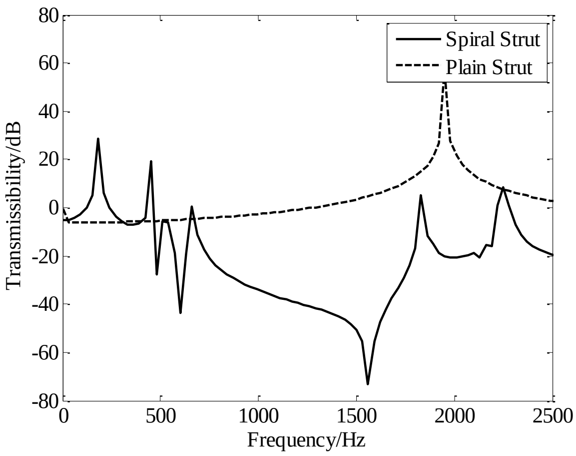
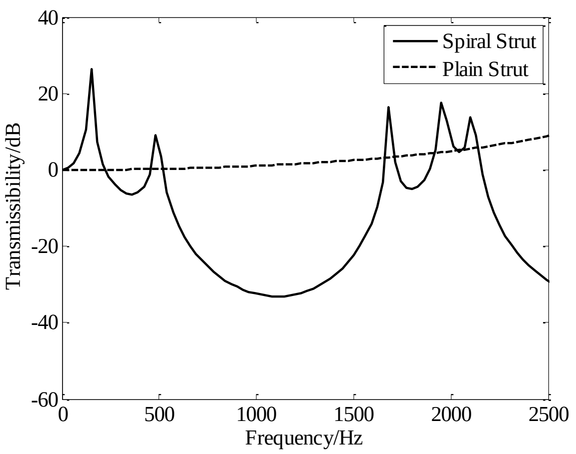
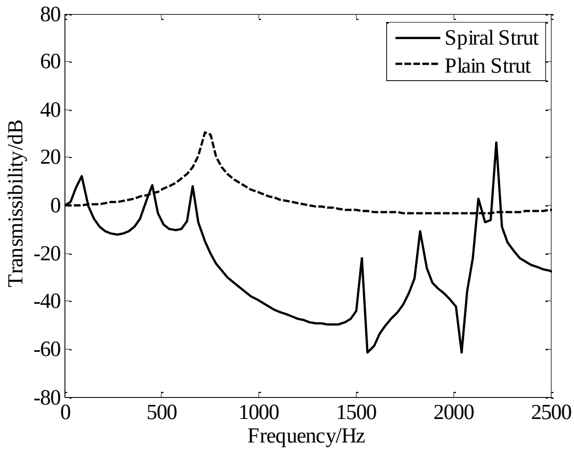


Figure 8. Lateral translation transmissibility of the novel periodic strut.



(a) Longitudinal



(b) Lateral

Figure 9. Rotational transmissibility of the novel periodic strut.

In addition, except for the low-frequency resonances below 500 Hz, several additional resonant peaks appear in the lateral direction between 500 Hz and 2000 Hz. These peaks may act on the cabin noise due to the small damping of metal material. This low-frequency amplification will limit the application of the novel strut. From this perspective, damping should be introduced in the advanced design of the novel strut to prohibit the resonances.

5.3. Noise Characteristics

To further verify the noise attenuation characteristics of the novel periodic strut, the noise level of the background helicopter model was calculated. The gearbox in the model was originally supported by three hollow cylindrical struts as shown in Fig. 2. The newly designed struts were used to substitute the plain struts. By the comparison of the changing of the cabin noise level, the noise attenuation can be predicted. The finite element model is presented in Fig. 10.

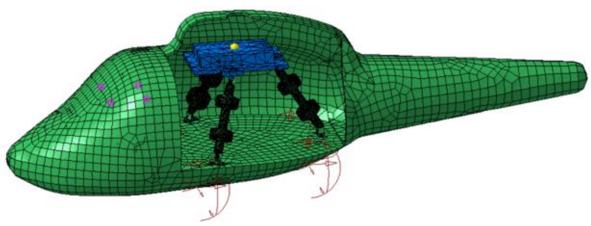


Figure 10. Finite element model of the background helicopter model with the novel struts.

The noise prediction results have been presented in Fig. 11. As the curve shows, the noise attenuation happens in the frequency range of 500-2000 Hz at the four selected noise measure positions. The beginning frequency appears at nearly 490 Hz. The maximum noise attenuation can exceed 20dB.

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However, noise is amplified in some frequencies especially in the range of 1100-1600 Hz. The reasons can be attributed to the resonances in the lateral or torsional directions of the strut as shown in Section 5.2. This is a disadvantage for the novel designed strut. Hence, both the longitudinal and lateral directions should be considered in the parameter design of the strut in the next stage. Also, there should be some damping added in the strut to improve its vibration and noise attenuation characteristics, especially in the high frequency.

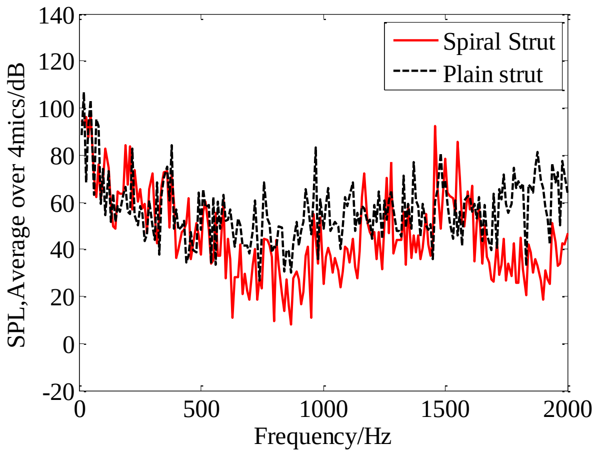

Figure 11. Cabin noise attenuation of the novel periodic strut.

6. CONCLUSION

A novel periodic strut for helicopter cabin noise reduction is presented in this paper. Based on the built theoretical model, parametric influence law was analyzed and found that if the static stiffness and the weight are constants, the stop band of the strut is mainly determined by the periodic number and the cell stiffness ratio. The strut can be quickly designed through reducing the periodic number and increasing the cell stiffness ratio to obtain a lower stop band.

In addition, simulation results show that the novel strut has excellent potential to prohibit the structural noise in the helicopter cabin generated by the main gearbox gear meshing vibration between 500-2000 Hz. However, the amplification will appear in some frequencies due to the small damping of metal material. Hence, all the translational and rotational directions should be considered in the parametric design of the strut. At the same time, damping would be introduced to advance the vibration and noise performance of the novel strut in the interest frequency range.

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