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STUDY ON ACTIVE & PASSIVE VIBRATION REDUCTION TECHNOLOGY OF COMPOSITE FAN BLADES BASED ON MFC

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ABSTRACT

With the development of the performance of aeroengine fan blades, composite materials have become the first choice of fan blades. However, in the process of operation, the fan blades are excited by complex loads, which will produce obvious vibration. MFC(Macro Fiber Composite) has both positive and negative piezoelectric effects. Therefore, MFC can be used for active control and passive vibration reduction of composite fan blades.

Based on the macro fiber composite (MFC), the research on the active and passive vibration reduction technology of composite fan blades is carried out by using numerical simulation and experimental methods. The specific contents are as follows:

The analysis model of active and passive vibration reduction of composite fan blades is established, the natural frequency, mode, modal strain and other information of composite components are obtained, and the determination method of MFC sticking position based on modal strain distribution is established, the influence mapping of active and passive vibration reduction effect of MFC is analyzed, and the analysis model is verified by composite laminated plate structure.

The MFC passive vibration reduction test of composite fan blades was carried out, and the LC circuit based on MFC was designed and constructed.

INTRODUCTION

Many research has been made in active & passive vibration reduction technology.

Many scholars have conducted theoretical and experimental research on active vibration reduction based on MFC.

Park and Kim (2004) studied the vibration, thermal post buckling and flutter characteristics of composite plates embedded with MFC under aerodynamic, thermal and piezoelectric composite loads. The numerical calculation results show that MFC actuators can effectively improve the vibration performance of the panel.

Robert et al. (2008) analyzed the effectiveness of piezoelectric actuators in reducing the buffeting response of the ventral wing of a fighter by using numerical research methods. Through flutter analysis and existing flight test data, the most critical mode for active vibration control was determined. At the same time, it was proposed that piezoelectric actuators should be pasted in the area with the highest strain energy, and align with the main strain direction in the area.

James (2013) and others developed and analyzed the multi physical field finite element model of the forced vibration response of composite fan blades affixed with MFC and verified the accuracy of the model through experiments, which also proved that the piezoelectric active vibration reduction based on MFC can significantly reduce the vibration of aeroengine composite fan blades.

The shape and structure of fan blades in turbofan engines are complex, with special materials used, and are affected by various loads. Analyzing their vibration characteristics has important reference significance for the study of fan blade vibration reduction.

Yu et al. (2018) studied the complex vibration of the system under the combined action of rotor imbalance and asymmetry, impact load, friction between rotor and stator, etc. when the fan blades of turbofan engines fall off. The research shows that the impact load caused by blade falling off on the rotor will lead to a sharp increase in vibration amplitude and reaction force, and at the same time, the impact friction will restrict the rotor and limit the transient vibration amplitude of the rotor.

Kou et al. (2016) studied the vibration stress of single-stage fan blade affected by the interaction of aerodynamic and structural behavior at multiple critical speeds, carried out numerical analysis based on the fluid dynamics finite element model. They established a single channel model of fan blade, and simulated the aerodynamic pressure load that causes the vibration of fan blade. It is observed that the vibration stress at the critical speed is significantly higher than that at the adjacent speed, and the peak stress is determined by the combined effect of the critical speed, vibration mode and synchronous excitation.

In recent years, many studies related to reducing structural vibration have begun to apply piezoelectric shunt technology, especially for relatively large but precise machinery, such as turbomachinery fan blades, bladed discs, and so on.

Choi (2012) and others tested various piezoelectric composite patches with different material properties to determine the piezoelectric transducer with the best performance. Then, the selected piezoelectric composites were applied to the laminated composite fan blades, and vibration reduction tests were carried out at several speeds.

Zhou (2014) and others focused on the piezoelectric shunt damping applied to mistuned bladed discs with multiple piezoelectric ceramic plates. The sensor is fixed on the disc surface between adjacent blades. Therefore, due to the coupling of the blade disk, the blade vibration can be significantly reduced.

Mokrani (2015) recently considered the resonant shunt damping of a rotating periodic structure using piezoelectric arrays, and applied it to represent blade discs used in turbomachinery. He mentioned the possibility of using a pure passive resonant shunt because the required inductance of a resonant shunt tuned in the target mode is very low. However, the author does not specify the required resistance value of the shunt, which can be lower than that of the pure passive inductor.

O. Thierry et al. (2016) studied the low-frequency vibration reduction of composite fan blades of turbojet engines using piezoelectric devices. The purpose is to prolong the service life and avoid flutter by reducing the amplitude. Solutions considered in the work include the use of piezoelectric elements to connect to passive circuits commonly referred to as diverters. Using braided composites for fan blades, it can be planned to embed piezoelectric materials, for this structure, the results show that the pure passive resonant shunt can significantly reduce the vibration level of the second bending mode, and the good correlation between the experiment and simulation verifies the best fit finite element model.

Airodi and Ruzzene (2011) studied the effective bending stiffness of a periodic beam with distributed patches shunted by inductance and resistance. They demonstrated that the effective bending stiffness of the beam exhibits resonance characteristics near the resonant frequency of the circuit, just as the effective mass of a metamaterial with a mechanical resonator is near the resonant frequency. On this basis, the concept of local resonant piezoelectric metamaterials is extended to flat plates.

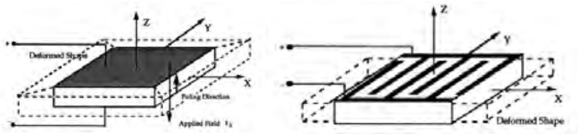
In order to expand the LR band gap in piezoelectric metamaterials, Wang et al. (2012,2016) proposed an amplifier resonator feedback circuit to achieve broadband gap. The concept of "rainbow trap" is discussed by gradually changing the diverter parameters in space. This design shows good results in increasing the band gap range, but the number of elements required to achieve a "rainbow trap" with an acceptable wave isolation level may be large. A shunt with multiple resonant frequencies is also proposed. Multiple resonance effects can be achieved by using multi branch shunt.

The above research realized the vibration reduction research for a variety of mechanical structures through experiments based on piezoelectric patches. The passive vibration reduction circuit based on a simple LC circuit or the use of piezoelectric shunt devices were studied to achieve vibration reduction in the low frequency band. However, the above research skips the finite element simulation and lacks the comparison between the experimental results and the finite element simulation results. This paper establishes a finite element simulation model based on MFC, extracts the vibration characteristics of the damping position through harmonic response analysis, and compares it with the results without MFC to obtain the dimensionless damping effect and damping amplitude, so as to improve the vibration characteristics analysis of composite blades under the excitation of piezoelectric sheets

METHODOLOGY

MFC

MFC uses piezoelectric ceramic composite fiber and interdigital electrode technology (IDEs) to combine, use directional drive to increase flexibility, improve the durability of PFC (Piezoelectric Fiber Composites), and use IDE (Interdigitated Electrode) to increase output strain for uniform electrodes. The free strain value of MFC is five times that of piezoelectric fiber composite with uniform electrode, which improves the driving performance of piezoelectric fiber composite. At the same time, the maximum free stress output in the fiber direction is about three times that in the transverse direction (1997).MFC-P1 interdigital interlaced electrodes are designed to generate electric field components on the driving plane of piezoelectric materials to improve the planar driving performance of piezoelectric materials. Traditional planar driving technology uses electric field to generate isotropic plane strain in X and Y directions through piezoelectric effects in the thickness direction (Z direction) and transverse direction (Y-axis, d_{31}) of the wafer, as shown in Figure 1-(a). As MFC is designed with interlaced electrodes, the longitudinal (X-axis, d_{33}) piezoelectric effect has a large electric field component in the X direction of the structure plane, as shown in Figure 1-(b). For most piezoelectric ceramics, the longitudinal piezoelectric effect can be significantly greater than the transverse piezoelectric effect, so the MFC interdigital cross electrode drive has a stronger planar drive. Longitudinal driving can not only increase the ability of free strain, but also greatly improve the ability of induced stress.



(a) d_{31} (b) d_{33}

FIGURE 1: Uniform Electrode Layer of Traditional Geometry Electrode & Fingertip Cross Electrode Interleaved Electrode Layer

PASSIVE VIBRATION REDUCTION METHOD

The structure obtained by parallel or series connection of inductance and piezoelectric patch is similar to mechanical resonator. In this study, MFC is used as a piezoelectric patch, which has an inherent capacitance C inside. Therefore, we just need to connect the MFC patch in series or parallel with capacitor C and inductor L of appropriate size and periodically combine on the surface of the object to form a resonant shunt, which can produce local resonant bandgap. It is worth noting that the inductance value should not be too large in general, so the series capacitor or parallel capacitor should be selected according to the size of the inductance. After the capacitors are connected in series, the capacitance value decreases, and after the capacitors are connected in parallel, the capacitance value increases. Change the value of circuit components (in the LC circuit, the inherent capacitance in MFC is not adjustable, so the adjustable circuit components in the LC circuit refer to the capacitance C and inductance L in parallel with MFC), so that the frequency of the circuit itself is close to the natural frequency of the target object to be damped, so that the target object can show resonance characteristics near the resonant frequency of the circuit.

For the LC circuit of the passive vibration reduction method, in the actual application process, all circuit components are determined and cannot be adjusted. Therefore, the use of a fixed inductance and a certain type of MFC chip has a limited range of resonance, that is, the band gap is narrow, and the vibration reduction effect for objects with multiple modes has a greater limitation, which can only produce effects for a single mode.

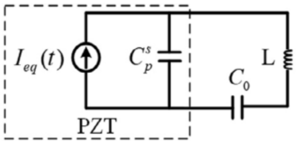


FIGURE 2: MFC And LC Circuit Connection

ACTIVE VIBRATION REDUCTION METHOD

As shown in the figure 3, several MFCs are pasted on the specific position of the composite fan blade, one of which is used as the sensor and the other as the damping driver. When the fan blade is excited to a certain extent, the MFC sensing module will sense the vibration state and transmit it to the control module. The control module will process the original signal generated by the signal generator after a certain amount of calculation to generate a control signal, which will be amplified by power and applied to the MFC driver. Because of the reverse piezoelectric effect, the MFC will generate response vibration to offset some of the original vibration, so as to achieve the effect of vibration reduction.

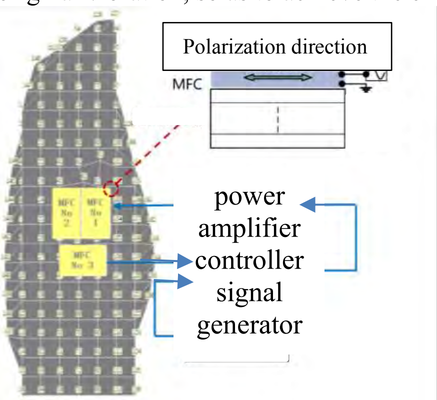


FIGURE 3: Schematic Diagram Of Active Vibration Reduction Based On MFC

According to the theory of structural dynamics, the continuous space structure can be described by a multi degree of freedom system, and the vibration response can be analyzed using the principle of modal superposition. The vibration energy is generally concentrated in the low order mode, so the vibration mode can be truncated and the low order mode that needs to be studied can be selected. The vibration response of a multi degree of freedom system can be expressed as a linear superposition of multiple vibration modes:

$$u = \sum_{i=1}^{N} u_i = \sum_{i=1}^{N} \xi_i \varphi_i \quad (1)$$

where, u_i is the modal displacement of the ith mode, N is the order of the considered mode, φ_i is the vibration mode vector, ξ_i is the modal coordinate, representing the contribution of each mode to the displacement response, equivalent to the participation factor or weighting factor.The electromechanical coupling model of the whole system is shown in the figure 4. The system is an n-degree of freedom spring mass damping system with MFC piezoelectric actuator. The force generated by MFC can be changed by changing the input voltage of MFC, so as to carry out vibration reduction control. Therefore, the project expects to study appropriate control methods to reduce the low order modal vibration of fan blades.

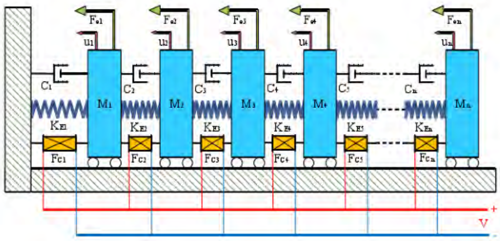


FIGURE 4: Multi-Mode Electromechanical Coupling Model

RESULTS AND DISCUSSION

ANALSYS OF ACTIVE VIBRATION REDUCTION OF COMPOSITE PLATE BASED ON MFC

Prepreg of the same model as the composite fan blade(epoxy carbon UD (230MPa)) shall be selected for the composite plate, and the ply design shall comply with the composite ply design requirements in the aviation field.

The size of the designed flat is 250mm long and 100mm wide, and the clamping area is selected as a 50mm wide area at one end of the panel surface.The layup design results and material parameters of laminate are shown in Tables 1 and 2.

TABLE 1 Layup Design Results of Composite Flat

Number of layers	Stacking Sequence	Thickness of single layer	Prepreg
12	(45/0/-45/0/90/0)s	0.186mm	epoxy carbon UD(230 GPa)

TABLE 2 Epoxy Carbon UD Material Parameters

Density (kg/m^3)	Young's modulus $\{E_1,E_2,E_3\}$ (Gpa)	Poisson's ratio $\{\nu_1,\nu_2,\nu_3\}$	shear modulus $\{G_1,G_2,G_3\}$ (Gpa)
1490	{121,8.6,8.6}	{0.27,0.4,0.27}	{4.7,3.1,4.7}

After the model is set, we limit the displacement of the clamping surface to 0, and apply an acceleration of 0.1g in the direction of plate thickness, then the characteristic frequency is calculated. The first six mode shapes are shown in Figure 5. It can be observed that the mode shapes of laminated plates are significantly different from those of single-layer plates.

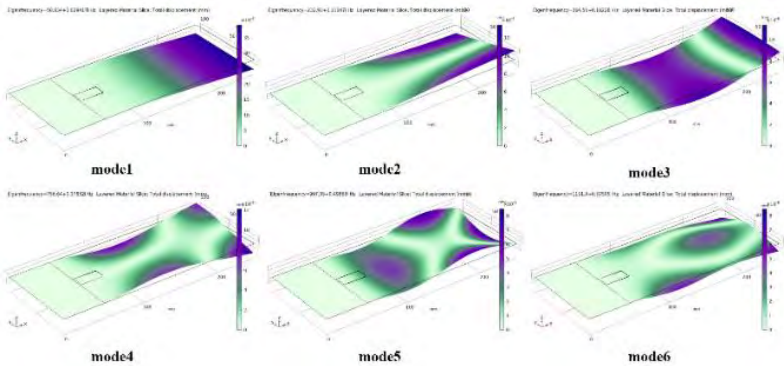


FIGURE 5: First Six Modal Shapes

Through cloud images, the nodal lines under each mode can be found, thereby analyzing the types of the first six modes mentioned above. We can obtain: the first order is the bending mode; The second order is the torsional mode; The third order is the bending mode; The fourth order mode is relatively complex, possessing both bending and torsion characteristics; The fifth order is the bending mode; The sixth order is the torsional mode.

Add MFC sheet solid model to the original shell model of composite plate, as shown in Figure 6. The mesh is changed to a free triangle element controlled by physical field, which has the coupling of electrostatic field and structural mechanics field in MFC area, and requires more intensive mesh division to obtain better solution accuracy. In the initial model, MFC was pasted on the root area of the plate close to the clamping surface, and it was symmetrical about the center of the plate.

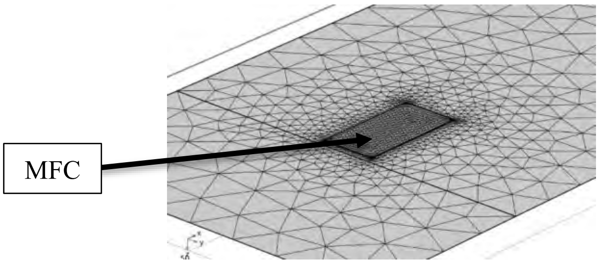


FIGURE 6: Geometric Structure And Mesh of MFC Laminate Coupling Model

After the coupling model is established, the modal calculation is carried out, and the calculated resonance frequency is compared with the previously calculated plate resonance frequency, as shown in Table 3. It is found that the damping resonance frequency changes slightly after the MFC is pasted. The main reason is that the introduction of MFC has changed the stiffness and mass matrix of the system. Because the volume and mass of MFC are small, the resonance frequency offset is also small, which is in line with the reality, and also shows that the connection between the two structures is correctly defined.

TABLE 3 Resonance Frequency of The System Before And After MFC Pasting

mode	Damping resonance frequency without MFC (Hz)	Damping resonance frequency with MFC (Hz)
1	58.834+0.029417i	59.295+0.029219i
2	232.93+0.11647i	233.58+0.11629i
3	364.56+0.18228i	365.86+0.18167i
4	756.64+0.37832i	758.70+0.37775i
5	997.35+0.49868i	998.99+0.49739i
6	1151.9+0.57595i	1152.2+0.57580i

INFLUENCE OF EXCITATION VOLTAGE AMPLITUDE AND PHASE

MFC uses its converse piezoelectric effect as the driver to reduce vibration. Its driving level is directly related to the excitation voltage. As shown by the calculation results of the aforementioned MFC model, the output strain of MFC is proportional to the amplitude of the excitation voltage. Therefore, the voltage applied at both ends of MFC naturally affects the vibration reduction effect.The voltage variation range in Figure 7 is 1000V to 10000V, and the step size is 1000V. The actual voltage amplitude should not be too high, which is only used as a reference for theoretical analysis. Two phenomena can be observed from the figure. First, the frequency of the anti resonant peak shifts, and second, the response value at the first order mode decreases first and then increases. The response values of the second and third order modes gradually increase with the increase of the voltage, and the response value is large almost always when there is no MFC effect.The anti resonance peak is often caused by the opposite phase and similar amplitude of the adjacent two modes near the frequency, which are nearly offset. Without MFC, there is no anti resonance peak between the first mode and the second mode. Because the response value generated by the first mode is large, the second mode is difficult to offset. When MFC is introduced, the first mode, the second mode and MFC may cancel each other at a certain frequency and form an anti resonant peak. Since the vibration of MFC changes with the voltage, the frequency of the anti resonant peak also changes. The response value at the anti resonance peak and its adjacent frequency band is almost 0, but it has little significance for the vibration reduction of the system, because the original vibration response value at this frequency is not large.

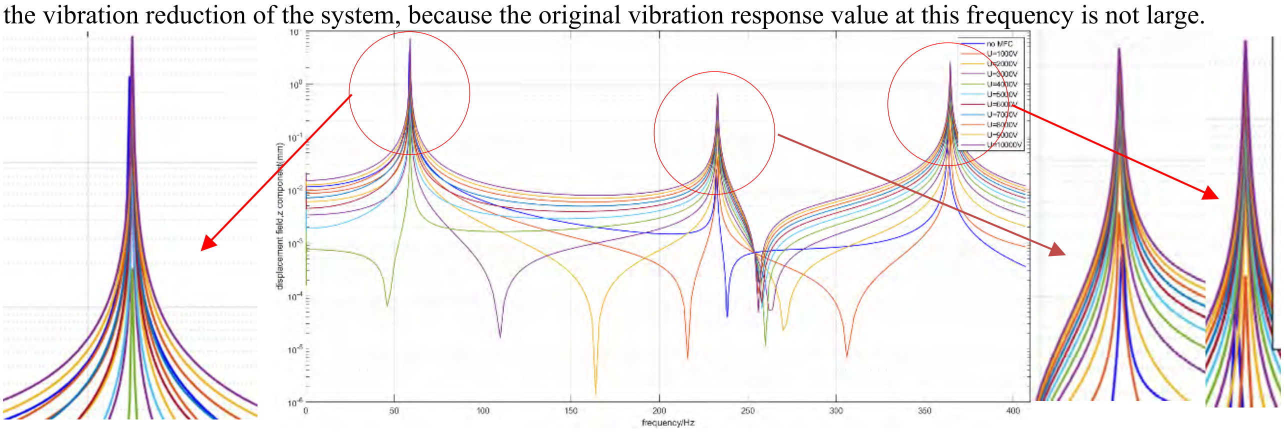


FIGURE 7: Frequency Response Under Excitation of Different Voltage Amplitude（1000V-10000V）

In order to further analyze the relationship between the response at the resonance frequency and the voltage amplitude, the frequency response of the system in the voltage range of 100V to 1500V and 3000V to 5000V was calculated with a parameterized scanning step size of 100V. It can be seen that under lower voltage excitation, the first mode is not significantly affected, while the responses at the second and third modes also decrease first and then increase with increasing voltage. At this pasting position, MFC has a corresponding excitation voltage value with the best vibration reduction effect for each mode. If the voltage value is too small or too large, it may cause the reaction effect of enhancing vibration. Therefore, the selection of excitation voltage amplitude is an important factor affecting the vibration reduction effect of MFC, and vibration reduction can be carried out separately for different modes by adjusting the voltage.

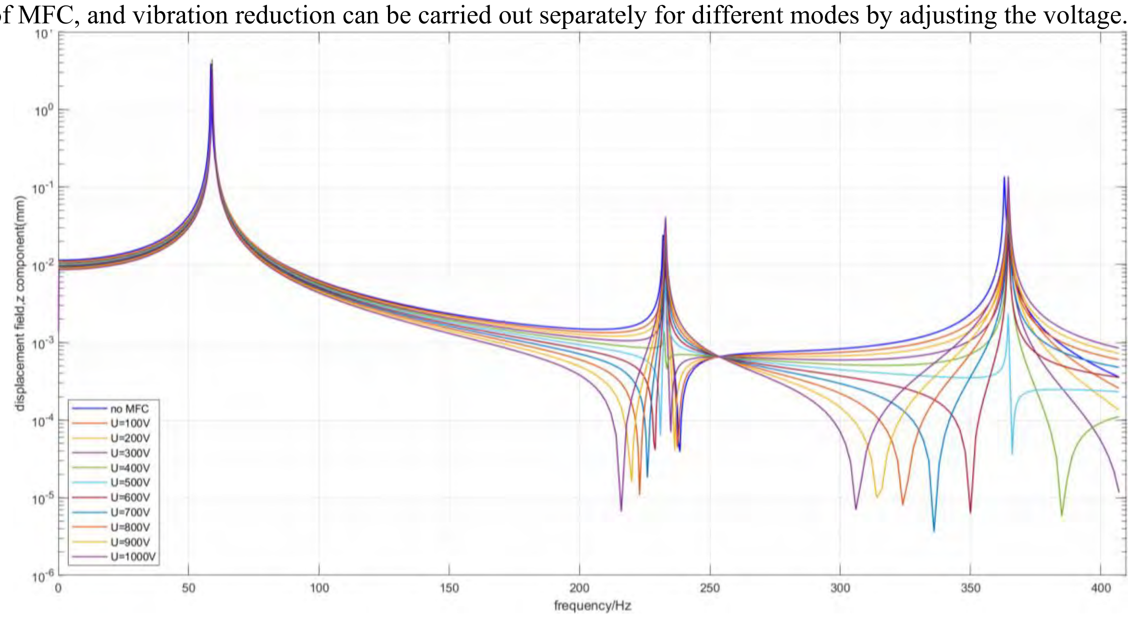
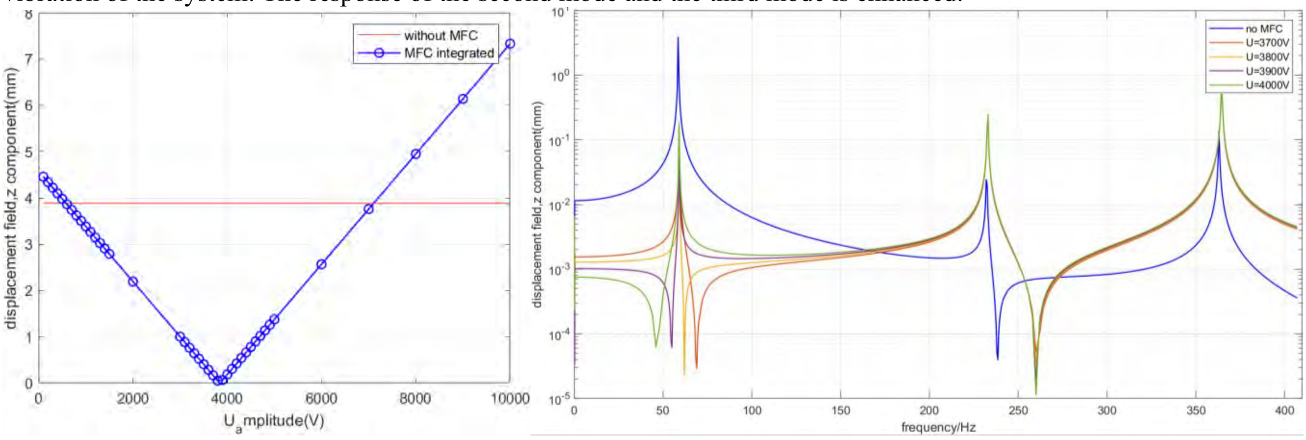


FIGURE 8: Frequency Response Under Excitation of Different Voltage Amplitudes (100V-1000V)

The first mode is selected for further discussion, and the change of response value at the first mode with the excitation voltage is plotted, as shown in Figure 9-(a). It can be observed that the reduction of response is proportional to the excitation voltage. According to the above MFC model calculation results, the strain value of MFC is proportional to the voltage, so it can be considered that the output strain size is proportional to the vibration reduction effect, and there may be a minimum point. The minimum point of the response at the pasting position of the MFC is at 3800V, and the response curve under 3700V to 4000V voltage excitation is drawn, as shown in Figure 9-(b). It can be seen that before the anti resonant peak gradually shifts from the first mode to the second mode, the vibration response also gradually increases, indicating that part of the output strain of the MFC counteracts the first mode strain at this time, and part of the output strain enhances the vibration of the system. The response of the second mode and the third mode is enhanced.



(a) (b)

FIGURE 9: Displacement Response Values Under Different Voltage Amplitudes At The First Mode & Response Curve Under Voltage With Better Vibration Reduction Effect of The First Mode

The results of the above calculation are all calculated when the phase difference between the MFC excitation signal and the plate vibration signal is 180°, that is, phi=pi. Since the laying direction of MFC is consistent with the length direction of the flat plate, under the setting conditions of the MFC ground terminal and voltage excitation terminal, when the phase angle is phi, it is exactly opposite to the structure. However, due to the restriction of other factors, it is unnecessarily effective to reduce vibration at the 180 ° phase angle. The response curves under each voltage when phi=0 are calculated, as shown in Figure 8. The change trend of the response curve with frequency has not changed compared with that before pasting MFC, but the response values have been expanded. If the analysis is consistent, when the phase angle is 0, MFC has an overall enhancement effect on the system response. It can be seen that the excitation phase of MFC is also an important factor affecting its damping effect.

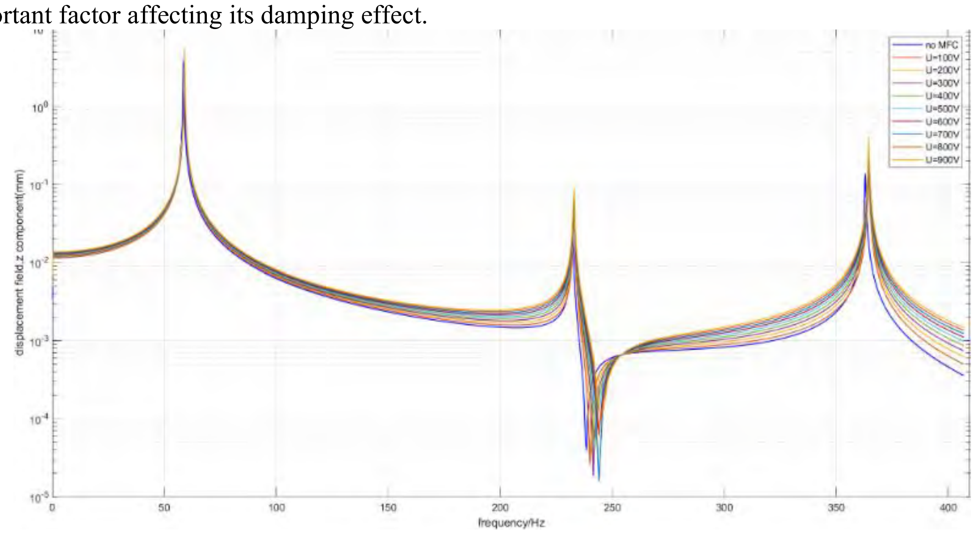


FIGURE 10: Frequency Domain Response Under Excitation Of Different Voltage Amplitude (phi=0°)

INFLUENCE OF LAYING POSITION AND ANGLE

It is found in the previous calculation that the vibration response of the structure is closely related to the output strain of MFC. Theory shows that the effect of MFC is also related to the size of the strain energy near the location where it is stuck. The location where the structural strain energy is large will often generate more intense vibration under the same excitation. Therefore, sticking MFC at the location where the strain energy is large can have better vibration reduction effect under the same voltage and phase. Firstly, the relationship between strain energy and strain distribution is analyzed from theoretical calculation.

Taking the model as a bending cantilever beam, z in the thickness direction of the plate can be assumed as follows:

$$\varepsilon_z = 0,\ \gamma_{yz} = \gamma_{zx} = 0 \qquad (2)$$

Then the strain of the thin plate can be expressed as:

$$\{\varepsilon\} = \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{Bmatrix} = \begin{Bmatrix} \frac{\partial u}{\partial x} \\ \frac{\partial v}{\partial y} \\ \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \end{Bmatrix} = -z \begin{Bmatrix} \frac{\partial^2 w}{\partial x^2} \\ \frac{\partial^2 w}{\partial y^2} \\ 2\frac{\partial^2 w}{\partial x \partial y} \end{Bmatrix} \qquad (3)$$

where, w is the deflection on the same uniform discovery, and the stress component of the beam is:

$$\{\sigma\} = \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \end{Bmatrix} = [D]\{\varepsilon\} = \frac{E}{1-\mu 2} \begin{bmatrix} 1 & \mu & 0 \\ \mu & 1 & 0 \\ 0 & 0 & \frac{1-\mu}{2} \end{bmatrix} \{\varepsilon\} \qquad (4)$$

Therefore, the calculation formula of strain energy can be obtained:

$$U_m^e = \int_{V_m^e} \frac{1}{2} \{\varepsilon\}^T \{\sigma\} dV = \int_{V_m^e} \frac{1}{2} \{\varepsilon\}^T [D_m] \{\varepsilon\} dV \qquad (5)$$

The calculation results of strain energy show that the greater the strain value at a certain location of the structure, the greater the strain energy density. To find the location of the maximum strain energy, only determine the location of the maximum strain distribution.Observe the strain distribution cloud diagram of the first order mode. The maximum area of strain distribution is concentrated at the corners. Therefore, the MFC is laid at the corners and compared with the situation that MFC was laid in the middle before, as shown in Figure 11.

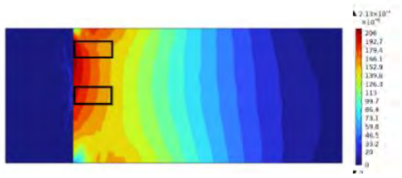
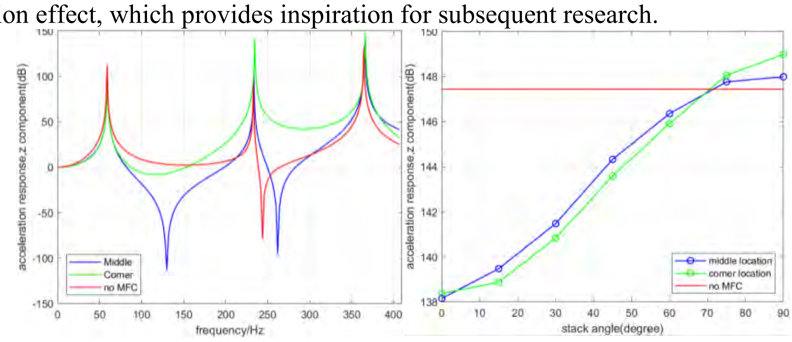


FIGURE 11: Comparison of MFC Laying Positions

The acceleration response curves of the system at the two paving positions are calculated, as shown in Figure 12-(a). There is a significant difference between the two results. Under the first-order modal frequency, the response of MFC at the corners is slightly smaller than that of MFC at the middle. After introducing the influence of the paving angle and doing fine scanning, the phenomenon of small response at the corners can also be observed, as shown in Figure 12-(b), indicating that the damping effect of MFC can be optimized by sticking it at the position with large strain. The reason for the small difference between the two is that the voltage used in the calculation is small, and the vibration reduction effect is not obvious. In addition, the difference of strain distribution between the two positions is not obvious, which will be improved in the subsequent vibration reduction design for multiple modes. It is worth noting that under different placement positions of MFC, the second mode and the third mode also have obvious changes, indicating that the mode itself also has an impact on the vibration reduction effect, which provides inspiration for subsequent research.



(a) (b)

FIGURE 12: Frequency Sweeping Results Under Different Laying Positions of MFC & Response of Mfc At Different Laying Positions And Laying Angles At The First Mode

INFLUENCE OF MODE

The interaction between modes often occurs in the above calculation. When MFC has a damping effect on one of the modes, it will often have an exciting effect on one of the modes, as shown in Figure 13. The modal shape is also the key factor to determine the strain distribution under the modal, so the modal is an important reference for MFC vibration reduction design.

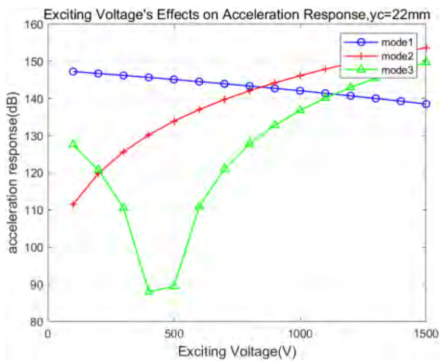


FIGURE 13: Influence of MFC On The First Three Modes Under Different Voltages

ANALSYS OF PASSIVE VIBRATION REDUCTION OF COMPOSITE FAN BLADE BASED ON MFC

FINITE ELEMENT SIMULATION

The blade finite element model used in this study is from the composite fan blade designed and manufactured by Zhu Qichen (2018).The three-dimensional model of composite wide chord fan blade and its various modes are shown in figure 14, which is designed by the research group itself. The composite swept fan blade is about 0.72m high, and the maximum chord length is about 0.33m. The straight tenon dovetail design is adopted.

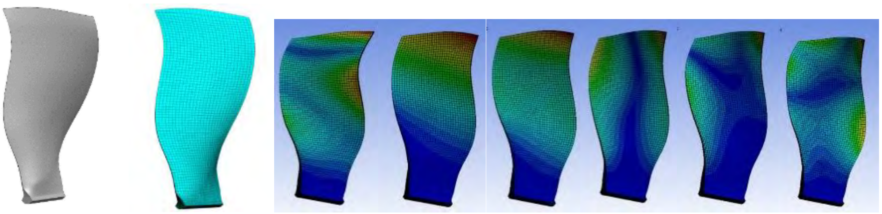


FIGURE 14: Composite Fan Blade Model & Various Modes of Composite Fan Blade（1-6）

TABLE 4 Natural Frequencies Corresponding To The First Six Modes Of Composite Fan Blades

Mode	natural frequency/Hz
Mode 1	55.705
Mode 2	146.89
Mode 3	284.92
Mode 4	409.78
Mode 5	558.2
Mode 6	676

Many researchers have studied the finite element simulation calculation based on MFC entities, but on the one hand, due to the large amount of calculation, it is difficult to find a simulation method that can fully describe the characteristics of MFC entities; on the other hand, the finite element model with excessively fine size has a long period and low efficiency in the application process, which is not suitable for practical engineering. Therefore, some researchers proposed that some simplified MFC finite element models ignore some non critical characteristics, grasp the MFC characteristics most relevant to structural performance, and conduct static characteristics and modal analysis of the MFC simplified model.

It is found in the study that MFC piezoelectric fiber composites have unique corresponding stress size for MFC under a certain voltage excitation through their own piezoelectric parameter characteristics. This stress size can be converted into pressure conditions through calculation, and the definition of piezoelectric module can be skipped directly. The static pressure conditions can be given for the part of the blade surface pre pasted with MFC through approximate calculation. The pressure corresponding to the stress is distributed on the grid where the suction surface of the composite blade should be pasted with MFC piezoelectric sheets. Therefore, the MFC entity is ignored, and only the mechanical action of MFC piezoelectric sheet on the composite blade surface under voltage excitation is considered, so as to analyze the similarities and differences between the modal and amplitude harmonic response curves when the MFC is not pasted.

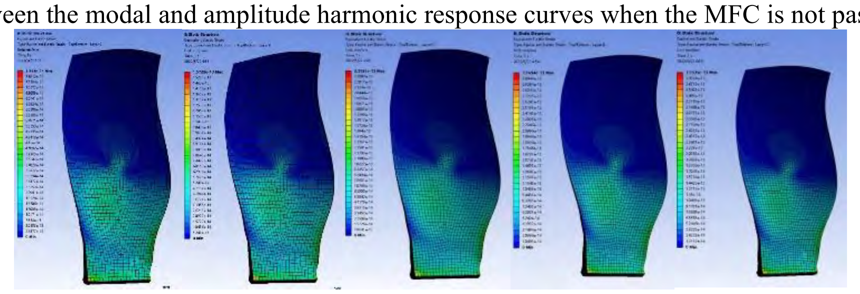


FIGURE 15: Strain Of Composite Fan Blade Under Single Different Voltage Excitation（300V-1500V）

According to the strain diagram of the static structure, the greater the pressure applied to the blade, the greater the strain result obtained. For different pressure constraints, the strain distribution on the blades is relatively similar, and the light blue grid area protruding above is the area where the pressure is applied, which is in line with expectations from the results. Next, modal analysis and harmonic response analysis can be conducted on the static structure to compare whether the composite blade model under the action of the MFC exciter has effectively achieved vibration reduction.

In order to verify the damping effect, the damping ratio can be calculated. The damping ratio of composite blade can be calculated by half power method. Through the sweep frequency curve of harmonic response, the damping ratio of the whole structure under different voltage excitation is calculated, and the change of damping without MFC is compared.

$$\xi = \frac{f_2 - f_1}{f_2 + f_1} \quad (6)$$

ξ is the structural damping ratio, f_2 and f_1 is the frequency corresponding to half power, and $f_2 > f_1$

TABLE 5 MFC Vibration Characteristic Parameters Under Different Voltages

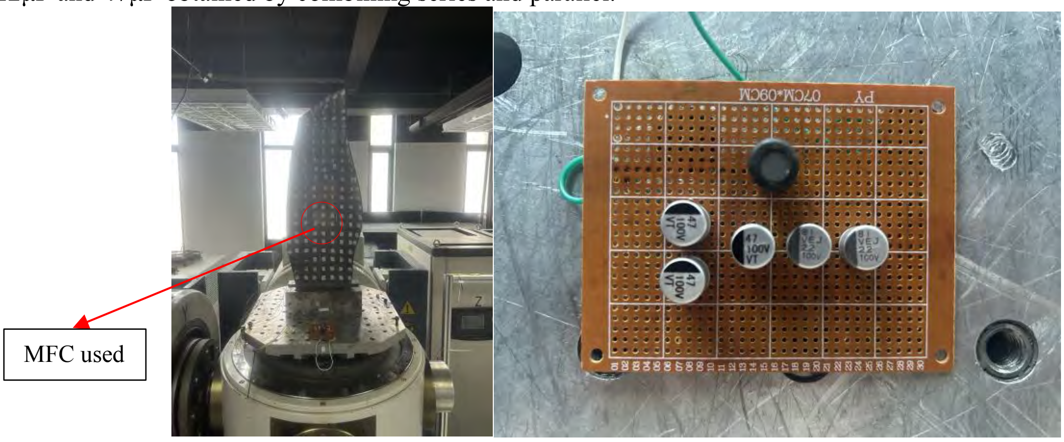
U (V)	A_{max} (mm)	$\frac{\sqrt{2}}{2}A_{max}$ (mm)	f_2 (Hz)	f_1 (Hz)	ξ
0	1.8926	1.3383	55.7143	55.6970	0.00015528
300	1.7701	1.2516	55.7144	55.6967	0.00015887
600	1.7658	1.2486	55.7143	55.6962	0.00016246
900	1.7543	1.2405	55.7146	55.6964	0.00016336
1200	1.7405	1.2307	55.7153	55.6965	0.00016874
1500	1.7256	1.2202	55.7149	55.6958	0.00017144

It is not difficult to see that, under the excitation of MFC piezoelectric sheet, the amplitude of resonance frequency decreases significantly. While increasing the voltage value applied at both ends of the MFC vibrator can also effectively reduce the amplitude under the resonance frequency, but the decreasing effect is relatively weak compared with the change effect of amplitude with and without MFC vibrator. According to the table data, we can conclude that the higher the voltage, the better the effect; However, the highest voltage within the working range of MFC, 1500V, isn't chosen as the optimal voltage, because excessively high voltage may damage the damping circuit. Hence, 300V is chosen as the optimal voltage.

The amplitude under the natural frequency is reduced by more than 5%, which is higher than expected from the simulation results.

Physical experiment

The setup of experiment is shown in the figure 14. Three MFCs were tested for their vibration reduction effect, and it was found that the lower one had the most obvious effect. Therefore, the LC circuit was chosen to apply to this MFC. Determine the value of circuit components according to the following formula, with an inductance of 1H and a capacitance of 22μF and 47μF obtained by combining series and parallel.



(a) (b)

Figure 16 Setup of Experiment & LC Circuit

Two different testing methods were used in the experiment to verify the damping effect of the passive damping LC circuit. The first method is random frequency scanning, with a frequency range of 0-800Hz, to obtain the maximum amplitude at the first mode before and after the LC circuit is applied. The second type is stationary frequency, which selects the first mode frequency(obtained by Random frequency scanning, 53.83Hz) and the same acceleration as the simulation at 0.1g. Measure the maximum amplitude at the first mode before and after the LC circuit is applied. The results are shown in Table 6.

Table 6 The max displacement of two conditions

	With LC Circuit	Without LC Circuit	vibration attenuation effect
Random frequency scanning	194.8μm	205.3μm	5.11%
stationary frequency	2.875mm	3.012mm	4.55%

CONCLUSIONS

Based on the designed composite plate finite element model and the existing composite fan blade finite element model, the calculation results show that both the active vibration reduction method and the passive vibration reduction method have a certain effect, and the active vibration reduction effect is better but the implementation method is more complex. This study has generally achieved the desired effect.

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