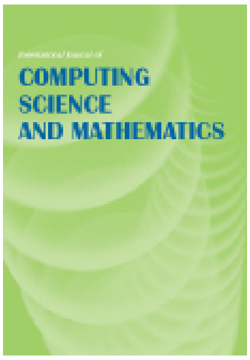

International Journal of Computing Science and Mathematics

ISSN online: 1752-5063 - ISSN print: 1752-5055
https://www.inderscience.com/ijcsm

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Jin Guo, Yunzhe Hou, Guanbing Cheng, Shuming Li

DOI: 10.1504/IJCSM.2023.10058371

Article History:

Received:	09 March 2023
Last revised:	01 June 2023
Accepted:	20 June 2023
Published online:	22 February 2024

Study on damage identification of aero-engine composite vane based on experimental modal parameters

Jin Guo

Laboratory of Aero-engine Modal Analysis,
College of Aeronautical Engineering,
Civil Aviation University of China,
Tianjin, 300300, China
Email: 1090799948@qq.com

Yunzhe Hou, Guanbing Cheng* and Shuming Li*

College of Aeronautical Engineering,
Civil Aviation University of China,
Tianjin, 300300, China
Email: 529386010@qq.com
Email: gbcheng@cauc.edu.cn
Email: smli@cauc.edu.cn
*Corresponding authors

Abstract: The paper took both undamaged and damaged composite plane plates as examples to effectuate the modal experiments by laser vibrometer measurement systems. The plate vibration modal parameters were identified. One coefficient was further constructed to locate the damage position. The experimental results show that the presence of the damage reduces the plate vibration amplitude due to its material deformation. Several orders of frequencies of the plate may not be identified only by the amplitude-frequency curves. For the plate, its first three orders natural frequencies are 67 Hz, 85 Hz and 105 Hz, respectively. The fourth and fifth orders ones correspond to 140 Hz and 225 Hz. The plate's natural frequencies decrease and its damping ratio increases in the presence of the damage on the plate. The plate damage does not change evidently its vibration shape features, but reduces its vibration amplitude. The constructed damage coefficient may identify the damage location in most cases.

Keywords: aero-engine; damaged composite plate; experiment modal; modal parameters; laser vibrometer; damage identification.

Reference to this paper should be made as follows: Guo, J., Hou, Y., Cheng, G. and Li, S. (2024) 'Study on damage identification of aero-engine composite vane based on experimental modal parameters', *Int. J. Computing Science and Mathematics*, Vol. 19, No. 1, pp.53–66.

Biographical notes: Jin Guo is a Master student in Aero-engine Modal Laboratory in College of Aeronautical Engineering, Civil Aviation University of China. His current research mainly includes structure dynamic computation and intelligent algorithm in complex computation.

Yunzhe Hou is a Master student in Aero-engine Modal Laboratory in College of Aeronautical Engineering, Civil Aviation University of China. His research interests are mechanical dynamic computation and intelligent diagnosis algorithm.

Guanbing Cheng is an Associated Professor of Aircraft Propulsion Engineering at Civil Aviation University of China, Tianjin, China. He received his PhD in Combustion and Detonation Laboratory of ENSMA in France in 2012. His current research mainly focuses on mechanical engineering, power plant intelligent diagnosis and combustion science.

Shuming Li is a Professor of Aircraft Propulsion Engineering at Civil Aviation University of China, Tianjin, China. He received his PhD in College of Mechanical Engineering, Tianjin University. His current research mainly focuses on mechanical engineering and power plant intelligent diagnosis.

1 Introduction

With the development of civil aviation industry, various composite materials are applied widely in the aero-engine components and accessories such as fan/compressor blade, casing and fluid tanks. To increase the force weight ratio of the aeroplane and ameliorate its performance, the aluminium alloy is used in the cold parts in early period instead of the steel alloys. In recent years, titanium alloy materials are predominant in the various parts in the airframe and aviation engine due to its lighter weight, high strength and longer lifespan. Therefore, several domestic and foreign aviation fabricants have adopted some advanced composite materials during the parts or components manufacture. This not only reduces the component weights, but also improves the part or engine overall performances. Nowadays, it is noticed that the fibre-reinforced composite materials are more popular with its advantage like specific modulus and stiffness in contrast with steel or aluminium alloys. The composite materials have complex structure and various types such as laminated, honeycomb and winding structures because of its layer method and material intrinsic properties in various directions (Zhou, 2008). For example, the civil aviation engine LEAP manufactured by CFM International adopts epoxy resin based composite material of its fan wide chord blade with higher strength, lighter weight and erosion resistance. Therefore, the engine fuel consumption and noise are reduced. As well-known, aviation engine functions in high temperature, pressure and revolution conditions. The application of composite materials also weakens the engine overall structural stiffness. The complicated environments such as the rotor unbalance, foreign object damage (FOD) and heat-solid-fluid coupling induce composite component vibration, and even lead to appearance of the damage (Yang et al., 2014). All these threaten greatly engine operation safety. The structural damage of the composite one changes its natural mechanic properties such as frequencies, damping ratio and vibration mode. Therefore, it is of interest to identify the damage location in the composite parts during its structural design and optimisation (Liu, 2021; Zhao et al., 2015).

At present, the structural damage identification is still more important on basis of classical modal parameters such as modal shape, natural frequency and damping ratio

(Yang et al., 2023; Deng et al., 2022; Duvnjak et al., 2021; Zhang et al., 2022). The modal parameter analysing methods are divided into frequency domain method, time domain one and time-frequency one. The frequency domain method is based on mutual independence for each order structural mode. The structural vibration is decomposed into a series of structural modal components (An, 2013). In structural dynamic frequency domain analysing, the concerned data is generally obtained by means of the frequency response function, for example, the graphical method in frequency domain identification. The time domain signal is Fourier transformed to obtain an amplitude-frequency curve from which the structural modal parameters may be roughly identified. Subsequently, the frequency domain analysing method was developed and the frequency response functions are expressed by mode parameter equation and parametric least squares to identify various modal parameters. Therefore, it is still important to evaluate qualitatively the structural parameters change with the undamaged and damaged condition. Various scientists, institutions and agencies devote further the efforts in this field (Zhou et al., 2019; Ghamami et al., 2020; Yang et al., 2021). Meng et al. (2013) initiated experimental modal analysis on composite plate structure by multi-point excitation method and an accelerator. They successfully obtained the modal parameters of the composite structure. Samyal et al. (2019) investigated effect of composite fibre orientation on the vibration patterns by finite element simulation. They found that the sensors must be optimally positioned to capture all the vibration characteristics. Yang et al. (2008) carried out the vibration modal tests on cantilevered composite laminate structures to analyse variation of vibration characteristics under different temperature conditions. They used the stochastic subspace method to identify the plate nature frequency, damping ratio and structural vibration modes.

Therefore, obtaining the modal parameters of composite plates based on experimental modal analysis has been widely developed and applied, which is convenient for the subsequent research of identifying structural damage or damage location by modal parameters (Yang and Peng, 2012; Yang et al., 2017). Antunes et al. (2020) discusses the experimental modal analysis used to compare the vibration modal parameters of a composite laminate with a mathematical model. The vibration modal analysis method was employed to determine the inherent frequencies and modal shapes of the structural vibration with high precision, while accounting for the various boundary conditions of the composite laminate. By examining the vibration modal parameters of the composite laminate, this study aimed to identify its dynamic behaviour and characteristics, including its resonance frequencies, damping ratios, and mode shapes. Khatir et al. (2021) has proposed an improved method for identifying damage in composite structures. By using the frequency response function of the structure as a starting point, and an improved artificial neural network algorithm to quantify the extent of the damage, the study found that damaged components could be predicted with high accuracy. These results suggest that the proposed approach could lead to more efficient and effective maintenance and repair strategies for composite structures. Khoo et al. (2004) used impact hammer and laser vibrometer to analyse its frequency response on wall structure. Modal analysis techniques were used to evaluate the structural modal parameter sensitivities to the wall damaged conditions. The changements of several parameters such as modal vibrations and modal residuals before and after the structural damage to quantificationally locate of the damage. Cao et al. (2017) compared reinforced concrete structure and fibre reinforced composite material, and compared natural frequency and damping parameters, and concluded that damping is more sensitive to damage than natural frequency and mode shape, and the damage is associated with the reduction of local stiffness of the structure, but there are still some identification errors in damping damage identification. Zhang et al. (2022) research progress in signal processing techniques for structural vibration damage identification is summarised, pointing out that experimental studies need to consider how to avoid the influence of environmental factors and test equipment on the experimental results, so that any damage present in the structure can be better identified, located and quantified. Patil and Reddy (2022) adopted accelerators to obtain the frequency response functions on the free-state composite plates. The modal assurance criterion (MAC) and the coordinated multimodal assurance criterion (COMAC) were established to correlate the structural modal parameters between undamaged and damaged plates, respectively. They found that that COMAC is more sensitive to the presence of minor damage in the plate structure. Hunady and Hagara (2017) used a three-dimensional digital image device to measure the full-field spatial displacement and strain of one composite plates. One force hammer was used to excite the plates to obtain the natural frequency and vibration shape by analysing the structure modified enhanced frequency response function. The structural modal vibration mode was more accurately estimated. Abdulkarem et al. (2017) applied the wavelet transform method to decompose the plate structure modal vibration parameters. The plate finite element models were considered within various boundary conditions such as cantilever, both ends fixtures and fixed four sides. They concluded that the wavelet decomposition method on the plate vibration mode differences is not affected by the boundary distortion and further validate the identification of the damage location on the plate structures. Pedram et al. (2017) proposed one damage detection method for the plate and shell structures by using a power spectral density (PSD) function. The structural finite element models were modified by deriving the sensitivity equations in order to identify the structure damage by the change of the modal parameters like natural frequencies and vibration modes.

From the literatures all-mentioned, we find that the structural modal parameter analysing is still one effective method during its damage identification. It is noticed that the mass of the accelerators may change the structure stiffness and lead to the measurement errors. Therefore, an experiment modal test was carried out on the undamaged and damaged conditions in present paper to obtain its structural modal parameters such as frequencies, damping ratios and vibration modes. The classical impact hammer and laser vibrometer were used in the experiments. The development of new techniques and algorithms, as well as advances in sensor technology, have made it possible to identify structural damage in a more accurate and reliable way. The change of those parameters was used to identify the location of the damage on the aero-engine static vane assumed as one composite plate. The experimental results may be provided with the references for the engine blade or vane structure design, optimisation and diagnosis.

2 Modal basic theory

For one composite plane plate, the basic modal analysis theory is respected. The plate vibrating dynamic characteristics may be described by some differential equations as follows (Tan, 2019).

$$MX + CX + KX = F \tag{1}$$

where $\boldsymbol{M}$, $\boldsymbol{C}$ and $\boldsymbol{K}$ are mass, damping and stiffness matrix of the system. $\boldsymbol{X}$ and $\boldsymbol{F}$ are displacement response vectors and excitation force vectors for each point. Laplace transform of equation (1) can be expressed

$$(S^2 + M + sc + K)X(s) = F(s) \tag{2}$$

$\boldsymbol{X}(s)$ and $\boldsymbol{F}(s)$ in equation (2) are Laplace transform of the displacement of the test point and the excitation force. The inverse of the impedance matrix $\boldsymbol{Z(s)}$ is the transfer function matrix as equation (3) shows

$$H(s) = [Z(s)]^{-1} = (S^2 M + sC + K)^{-1} \tag{3}$$

For linear time invariant system, its pole is in the left half plane of the complex plane, therefore, $\boldsymbol{s}$ can be replaced by $\boldsymbol{j\omega}$, the impedance matrix and frequency response function matrix in Fourier domain can be expressed

$$Z(\omega) = (K - \omega^2 M + j\omega C)^{-1} \tag{4}$$

Thus the motion equation of the system is formulated

$$(K - \omega^2 M + j\omega C)X(\omega) = F(\omega) \tag{5}$$

Then the corresponding point of any $\boldsymbol{i}$ is

$$\begin{aligned} x_i(\omega) &= \varphi_{i_1} q_1(\omega) + \varphi_{i_2} q_2(\omega) + \cdots + \varphi_{i_N} q_N(\omega) \\ &= \sum_{r=1}^{N} \varphi_{i_r} q_r(\omega) \end{aligned} \tag{6}$$

In equation (6), $\boldsymbol{\varphi_i}$ is the ***ith*** measurement point and $\boldsymbol{r}$ is the mode shape coefficient. The column vector composed of vibration mode coefficients of numbers N of measurement points is

$$\varphi_r = \begin{Bmatrix} \varphi_1 \\ \varphi_2 \\ \vdots \\ \varphi_N \end{Bmatrix}_r \tag{7}$$

$\boldsymbol{\varphi_i}$ is the $\boldsymbol{r}$**th** mode vector. The matrix composed of the modal vectors is called the modal matrix as shown

$$\varphi_i = \{\varphi_1 \quad \varphi_2 \quad \cdots \quad \varphi_N\}_i \tag{8}$$

3 Experimental facilities and procedures

3.1 Experiment setups

Figure 1 shows the experiment schematic of vibration mode test on one static composite vane of one aero-engine. The vibration measurement system mainly contains excitation subsystem, response reception one and signal dynamic acquisition and analysing one. The excitation system consists of one classic impact hammer TEST IH-05 with the sensitivity of 2.3 mV/EU. One piezoelectric sensor was embedded inside one end of the hammer, the vibration impact force signal was transmitted by the signal wires outside another end of the hammer. Figure 2 is an example of one typical excitation signal in experiment test. The signal pick-up system includes one laser vibrometer system Julight VSM-4000 with one sensitivity of 1000 mV/EU. The laser spot was projected directly on one surface of the composite vane. Figure 3 shows an example of the vibration response signal. One dynamic signal acquisition and analysing system contains transformer JM5890, signal data acquisition unit VSM-DAQ with four channels and software units like Signal and Modal ones in order to filter, amplifier and analyse the vibration signals.

Figure 1 Experimental mode schematic of vibration test on the composite vane (see online version for colours)

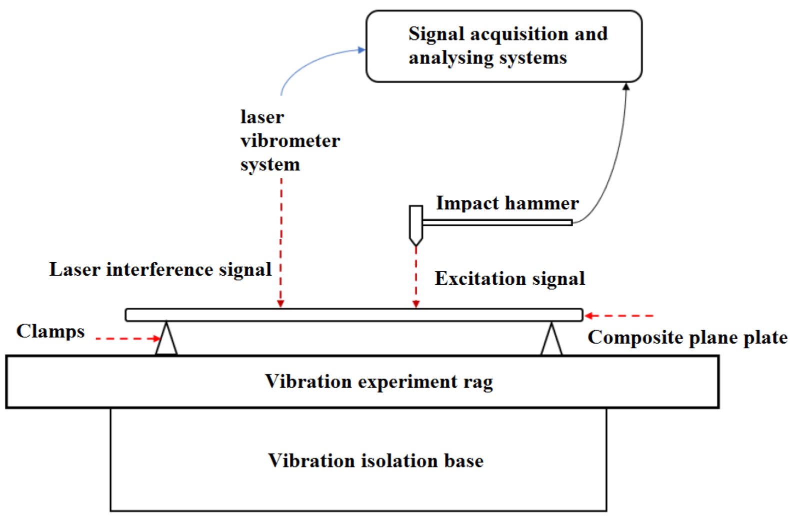
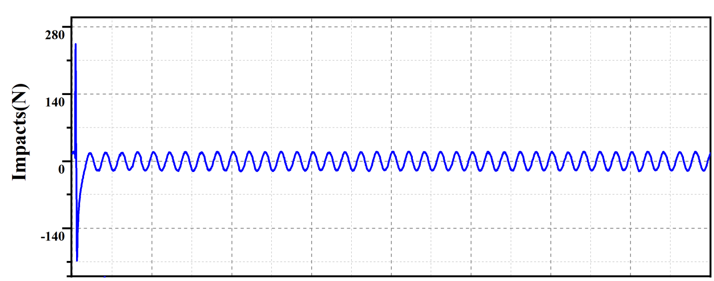


Figure 2 Example of vibration excitation signal (see online version for colours)



3.2 Composite vane model

The static vane of aero-engine was assumed as one plane composite plate, as shown in Figure 4. The plate length is 500 mm, its width is 200 mm and its thickness is about 1.75 mm. The composite plate was made from carbon fibre epoxy resin prepreg T300. The plate was stacked by 14 commutative layers with the layer angels 0 and 90 degrees respectively. The plate tensile strength is 743 MPa and its modulus of elasticity for tension is 50.5 GPa. The plate compressive strength is 337.5 MPa and its modulus of elasticity in compression is 33.2 GPa. The content of carbon is about 93%. The layer angel 0 degree was used to reduce effects of the axial loads on the composite plate and the one 90 degree was adopted to attenuate effects of the transversal loads on the plate and also control the plate Poisson Ratio along its directions.

Figure 3 Example of vibration response signal (see online version for colours)

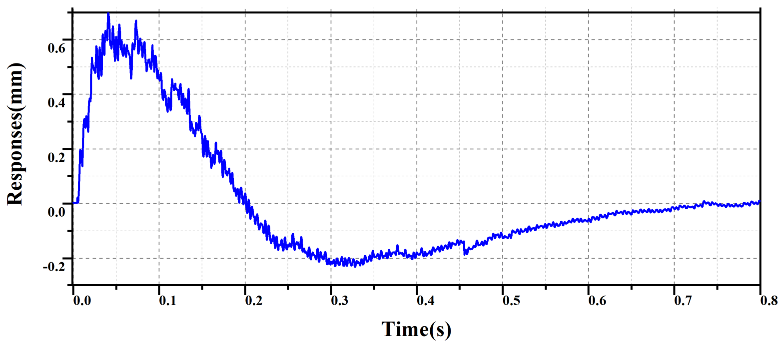
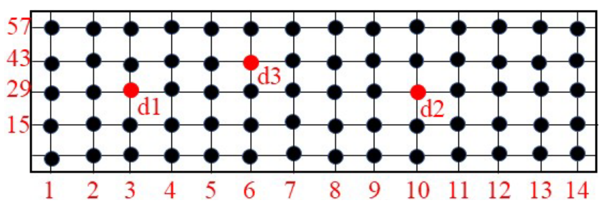


Figure 4 Physical model of the assumed composite plate (see online version for colours)



3.3 Procedures

To carry out the experimental modal test on both the undamaged and damaged composite plane plate, we marked 70 measurement points as seen in Figure 4. There are 14 points in its longitudinal direction and five points in its transversal one. The both ends of the plate were fixed by the clamps above the experiment base. The damaged points d1, d2 and d3 were appointed as No. 31, No. 38 and No. 48 on the plate. The vibration pick-up laser spots were placed on those three points. Each test, we move the impact hammer. To reduce measurement errors as far as possible, each test was repeated three times. Therefore, we have 70 arrays of the vibration signals such as the time-domain excitation signal, response one and the amplitude-frequency characteristic curves for each damaged point.

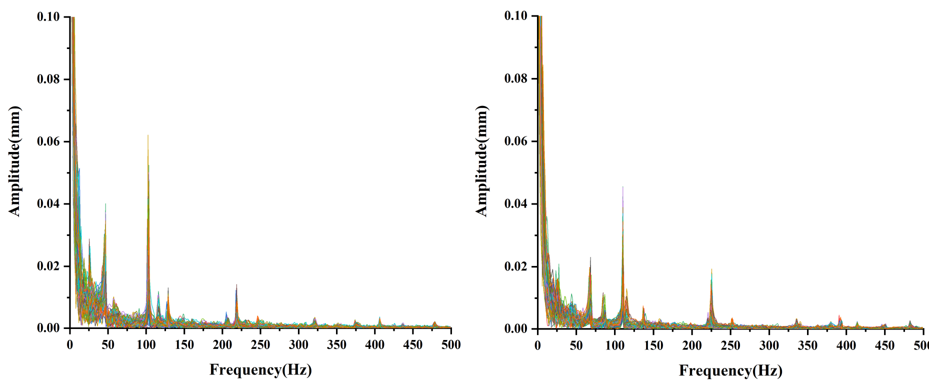
4 Results and discussion

4.1 Amplitude frequency response

Figure 5(a) and (b) represent the amplitude frequency response curves for the undamaged and damaged composite plane plates respectively when the laser spot was placed No. 31

measurement point. The horizontal ordinate denotes the frequencies of the plate and the vertical one shows its vibration amplitudes. From both figures, we see that the frequencies of the plate vary from 0 Hz to 500 Hz. For the undamaged plate, the first order frequency is probably 55–65 Hz, but the second one does not identify clearly. The third frequency of the plate is about 100 Hz. The fourth one is about 125 Hz. The next order frequency is on the order of 225 Hz. For the damaged plate, its first order frequency is about 65 Hz, the next four orders ones correspond 90 Hz, 110 Hz, and 140 Hz as well as 225 Hz. Therefore, the presence of the damaged plate reduces its vibration amplitude and increases its resonant frequencies because of the deformation of the plate.

Figure 5 Amplitude-frequency curves on the composite plate originated from no. 31 measurement point: (a) undamaged composite plate and (b) damaged composite plate (see online version for colours)



(a) (b)

Similarly, Figure 6(a) and (b) indicate the vibration amplitude frequency response curves for the undamaged and damaged plates when the laser spot was placed No.38 measurement point. Hence, we see that for the undamaged plate, the first order frequency is nearly 50 Hz, but the second and third ones cannot be verified evidently. The fourth one is about 140 Hz. The fifth order frequency corresponds 225 Hz. However, we see the same tendencies on the plate first five orders frequency variation. So, it seems that the plane plate damage cannot change its vibration frequencies, but it attenuates the fourth order vibration amplitude of the plane plate.

When the vibrometer laser focused on No.48 measurement point, we can obtain the vibration amplitude frequency diagrams shown in Figure 7(a) and (b) for the undamaged and damaged plates. We notice that for the presence of plate damage does not basically change its vibration frequencies. Its first three resonant frequencies correspond 50 Hz, 65 Hz and 90 Hz. The fourth and fifth order ones are about 145 Hz and 230 Hz. But for the undamaged plate, the third order frequency cannot be identified. The plate damage does not change evidently its vibration amplitude in contrast with ones in No. 31 and No. 38 measurement points.

Figure 6 Amplitude-frequency curves on the composite plate originated from no.38 measurement point: (a) undamaged composite plate and (b) damaged composite plate (see online version for colours)

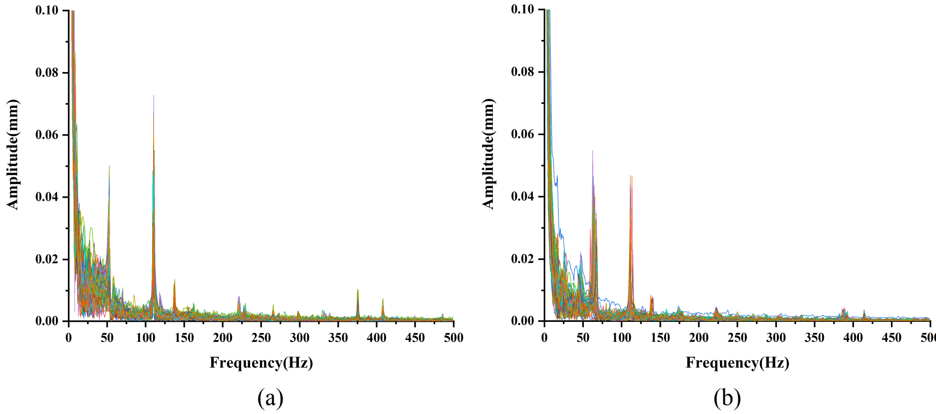
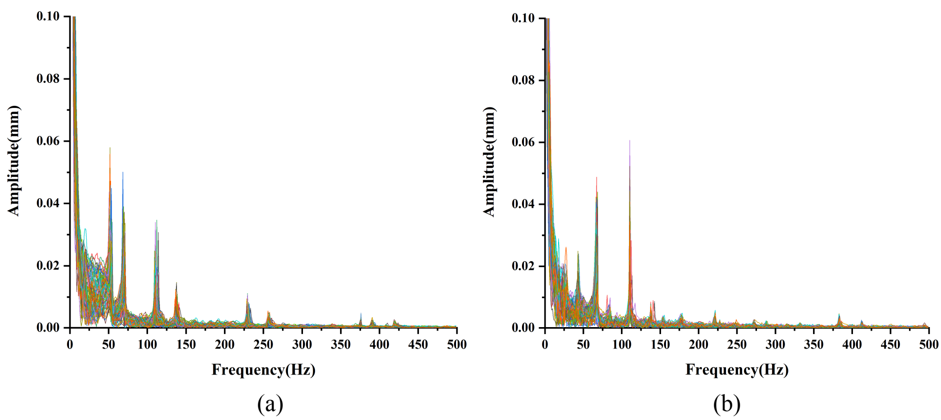


Figure 7 Amplitude-frequency curves on the composite plate originated from no. 48 measurement point: (a) undamaged composite plate and (b) damaged composite plate (see online version for colours)



4.2 Natural frequencies and damping ratios

By analysing the amplitude frequency characteristics of the undamaged and damaged plane plates, its first five orders structural vibration modal parameters can be identified correspondingly. Table 1 shows the comparison of the plate resonant frequency and damping ratio under both the undamaged and damaged conditions. From Table 1, we see that the undamaged plate first order frequency is on the order of 67 Hz, the second order one varies from 70 Hz to 85 Hz, the third order one shifts from 100 Hz to 110 Hz. The next fourth and fifth orders ones are about 140 Hz and 225 Hz, respectively. As for the damaged plate, its first order frequency varies from 60 Hz to 70 Hz. The second order one changes from 90 Hz to 95 Hz. Its third order one is from 100 Hz to 110 Hz. The plate fourth order frequency varies from 130 to 140 Hz and the fifth order one is 200 to 220 Hz. When the composite plate damage takes place, its first order, the third to fifth orders' resonant frequencies deduce. But the presence of the damage on the composite plate reduces mostly the plate structural damping ratio.

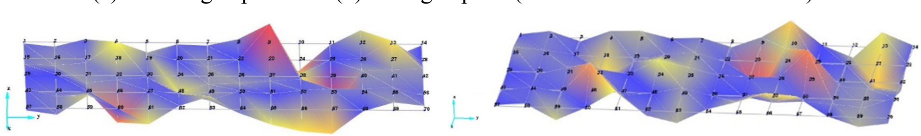
Table 1 Comparison of natural frequencies and damping ratio of the undamaged and damaged composite plate

Measurement points	*Undamaged structure*		*Damaged structure*	
	Frequency/Hz	*Damping ratio*	*Frequency/Hz*	*Damping ratio*
No. 31	67.56	0.07%	62.82	0.20%
	85.83	0.23%	88.16	0.30%
	109.76	0.15%	101.21	0.30%
	136.87	0.18%	129.72	0.43%
	225.46	0.47%	218.35	0.54%
No. 38	66.34	0.14%	62.53	0.06%
	76.24	0.09%	95.86	0.21%
	110.17	0.12%	109.49	0.23%
	138.29	0.24%	146.57	0.19%
	219.46	0.07%	219.01	0.40%
No. 48	67.01	0.24%	68.83	0.48%
	72.22	0.15%	90.47	0.23%
	98.19	0.21%	111.76	0.18%
	139.76	0.01%	128.94	0.26%
	222.47	0.30%	206.58	0.39%

4.3 Vibration mode

Figure 8 shows the first order vibration mode of both the undamaged and damaged composite plates. The red zone represents more important amplitude value and the blue zone signifies that the vibration amplitude is zero. From Figure 8, we see that most zones on the plane plate vibrate synchronously with one bending features. The plate both ends do not participate in the vibration phenomenon. The most important amplitude of the plate vibration occurs at the proximity of its edge. In addition, the presence of the plate damage, as shown in Figure 8(b), reduces its amplitude evidently.

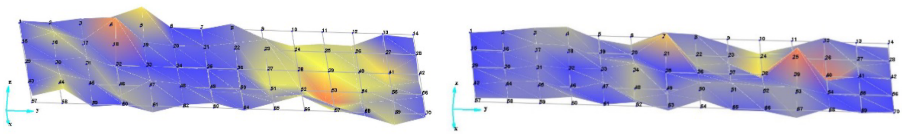
Figure 8 The first order vibration mode of the undamaged and damaged composite plates: (a) undamaged plate and (b) damaged plate (see online version for colours)



(a) (b)

Figure 9 depicts the second order vibration modal of the undamaged and damaged composite plates. The colour zone presents the amplitude value as similar as shown in Figure 8. We can observe from Figure 9 that the vibration takes place mainly in both the plane plate left and right half part, but they vibrate conversely with two order bending characteristics. There is one vibration pitch line at the middle of the plate. The maximum vibration amplitude value occurs at the one and three quarters along the longitudinal direction of the plane plate. The plate damage also reduces the amplitudes.

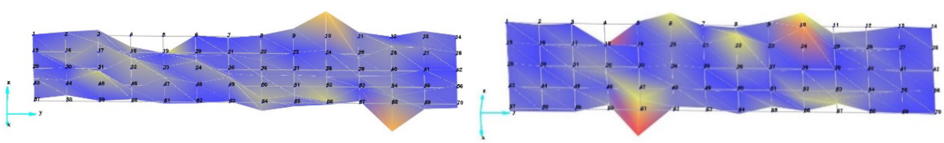
Figure 9 The second order vibration mode of the undamaged and damaged composite plates: (a) undamaged plate and and (b) damaged plate (see online version for colours)



(a) (b)

Figure 10 expresses the third order vibration mode of both the undamaged and damaged composite plates. We can obtain from Figure 10 that the vibration takes place mainly in three parts on the plane plate with three bending features. Most zones in the composite plane plate one-third and three third vibrates in similar direction, but do conversely in the two third plate. Hence, the maximum vibration value occurs nearly at the middle of each parts. The presence of the damage in the plate also its vibration amplitudes.

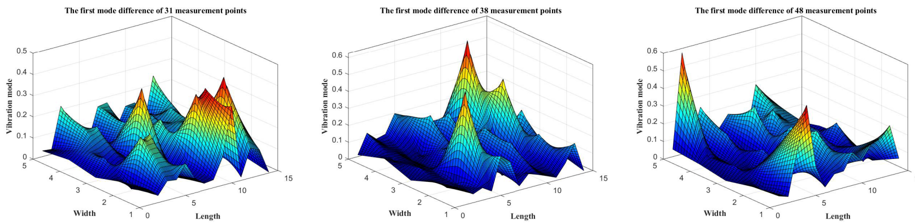
Figure 10 The third order vibration mode of the undamaged and damaged composite plates: (a) undamaged plate (b) damaged plate (see online version for colours)



(a) (b)

In addition, we notice that the plane plate first order frequency value is nearly equivalent in Table 1 when the laser spots were placed in No. 31, No. 38 and No. 48 measurement points. Therefore, we choose the first order vibration mode of the plate. In order to identify the damage location on the plate, we provide that one damage coefficient. It may be defined as one difference value of the vibration amplitudes between the undamaged and damaged plane plates. All the data on the first order vibration shape of the plate were extracted from the its experimental modal data (Altunisik-Kang and Ren, 2018; Altunisik et al., 2017; Altunk et al., 2019). Figure 11 denote the comparison the vibration mode differences between the undamaged and damaged plates in three measurement points. The red zone and blue one corresponds the amplitude maximum and minimum values, respectively. Figure 11(a) and (b) shows that the damage position is nearly located at both the No. 31 and No. 38 measurement points. Figure 11(c) reveals only that the plate damages at No. 31 measurement point. Figure 11(c) indicates that the most important amplitude takes only place at the left side corner of the plane plate, thus it further shows that the damage coefficient does only identify the damage presence at the vane boundary, and its exact location need be further investigated by the vibration-based damage recognition method of such composite vane.

Figure 11 Comparison of first-order mode difference between undamaged plate and damaged plate at different measuring points: (a) measuring point 31; (b) Measuring point 38 and (c) Measuring point 48 (see online version for colours)



(a) (b) (c)

5 Conclusion

The present paper took a composite plane plate as example within both the undamaged and damaged conditions. The plate modal experiments were carried out by classical non-contact measurement technology and devices such as impact hammer, laser vibrometer and dynamic acquisition and analysing systems. The plate structural modal parameters such as vibration amplitude, frequencies and damping ratio together with vibration mode were identified. Meanwhile one damaged coefficient was constructed to describe its effects on the vibration shapes of the plane plate. The main conclusion is listed as follows

1 One modal experiment system on one composite plane plate was designed when the laser spots were placed on three measurement points. The plate structural modal parameters were obtained accordingly. We find that the presence of the damage attenuates to a certain extent the plate vibration amplitude due to the deformation of the plate material near the damage. Several orders frequencies of the plate may not be identified only by the amplitude-frequency curves.

2 For the undamaged plate, its first three orders natural frequencies are 67 Hz, 77 Hz and 105 Hz, respectively. The fourth and fifth orders ones correspond 138 Hz and 220 Hz. For the damaged plate, its first order frequency varies from 62 Hz to 68 Hz. The second order one increases up to about 92 Hz. The third order frequency seems to be unchanged, but the fourth and fifth orders ones mostly reduce. The plate resonant frequencies slow down and the structural damping ratio increases when the damage was presented on the plate.

3 The plate damage condition does not change its vibration shape features. The first three orders modes are classical one, two and three bending vibration characteristics. The plate damage may attenuate its vibration amplitude, but it is not enough to distinguish the damage condition of the plate and identify the damage position of the damaged plate only by the vibration shape changes.

4 A damage coefficient was constructed by the differences between vibration amplitudes for both the undamaged and damaged plates. We find that the damaged coefficient may be used to identify the damage location in mostly cases, but some

spurious peaks at the boundary appear, which may be caused by the boundary singularity caused by the fixture. Further research is still needed in follow-up experiments.

Acknowledgements

This paper is financially supported by the Funding for graduate research innovation projects of Civil Aviation University of China (Grant No. 2021YJS041) in the Laboratory of Aero-engine Modal Analysis.

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