INFLUENCE OF LASER SHOCK PROCESSING ON BLADE MODAL STIFFNESS

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Abstract

In this study theoretical measurements and the finite element method are used to analyze the mechanism of blade vibrations using laser shock peening (LSP) for the mode shapes with stretching of the mid-plane, which are commonly encountered in some local zones of the higher-order mode shapes. The effects of LSP in different selected zones on modal stiffness are investigated. It is shown, that interaction between the stretching of the mid-plane of the mode shape and the residual stress induced by LSP causes a change in the modal stiffness, and then the resonance frequency of the targeted mode is changed.

Keywords and phrases: Laser shock peening, vibration, modal stiffness, finite element method (FEM), residual stresses (RS)

AMS subject classification (2010): 76L05, 65M60, 74A10.

1 Introduction

The vibration of the blades of aircraft engines and industrial gas turbines, which are core components working in harsh environments, is difficult to avoid and often causes blade failure and even serious safety accidents. There are two ways to avoid fatigue damage and increase the service life of a vibrating blade. The first one is to improve the blade strength limit and the second one is to control the vibration of the blade.

Blade material research and development can significantly enhance the properties of the blade, such as fatigue resistance, strength, hot corrosion resistance and structural stability [1]. However, new materials are usually developed over a long period of time, and changes in material result in changes in mass and stiffness, which are undesirable for blades that have been designed for styling.

Therefore, surface treatment technologies as post-processing techniques for structural strengthening and without the mass and stiffness changes, have been studied and greatly developed [2]. However, structural strength cannot be infinitely improved, and these methods have not solved the root of the damage caused by vibrations.

Vibration control is a method to solve the damage caused by vibrations from the source, in which the resonant frequency is adjusted to avoid resonance, or damping and suppression are used to control the amplitude of the vibration, thus reducing internal stresses in the blade [3]. Due to the denseness of modes and the complexity of the vibration sources, it is impossible for the blade to completely avoid the resonance [4]. In addition, the usage of a broadband frequency damper will increase the difficulty of the design or decrease efficiency. Therefore, it is necessary and valuable to develop a method for selectively controlling the dangerous vibration mode, which can greatly improve the design flexibility.

The passive control technology achieves the purpose of vibration control through the design of a damping or friction structure for dissipating the vibrational energy. The technique is inherently stable and easy to implement for the blades. Thus, the passive control technology, such as friction dampers, particle damping, air film dampers and viscoelastic dampers, is widely applied to blades [5].

The present work applies laser shock peening (LSP) on a selected small zone in the blade for the selective modal control of the blade vibrations, which belong to the category of passive control methods. As an innovative surface treatment technique, LSP is very effective in improving material properties through leading to residual stresses and grain refinement [6-10]. However, due to the fact that LSP slightly affects the vibrational characteristics of the blade in general, and the significant effect only occurs in some special conditions, few studies or observations have been performed.

On the basis that the mode shapes differ significantly from each other, this study utilizes the interaction between the mode shape and the residual stresses and grain refinement caused by LSP to change the modal stiffness and modal damping of the specific mode, resulting in selective modal control. A diamond-shaped specimen is chosen to simulate the blade as the object of the finite element method (FEM).

2 The Shape and Material of the Sample

The sample, shown in the Fig. 1, was investigated. The specimen uses 2 mm-thick 2024-T4 aluminum with a mass density $\rho = 2780 kg/m^3$, a Poisson's ratio of $\nu = 0,33$ and a Young's modulus of $E = 72400$ MPa.

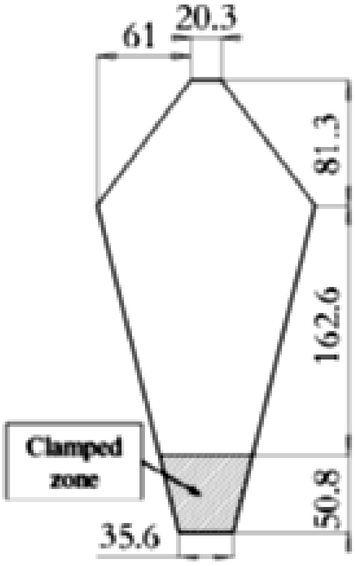

Figure 1: Specimen dimensions (mm)

3 Laser Shock Peening

A pulsed laser, striking on the target surface, with a short pulse width ($\sim 20ns$) and high-power density ($\sim 1,88 GW/cm^2$) was used for the study. The LSP process in detail is described in [11]. The strategy of finite element modeling in order to study the influence of LSP technology on the dynamic characteristics of the blades is presented in Fig. 2 [7].

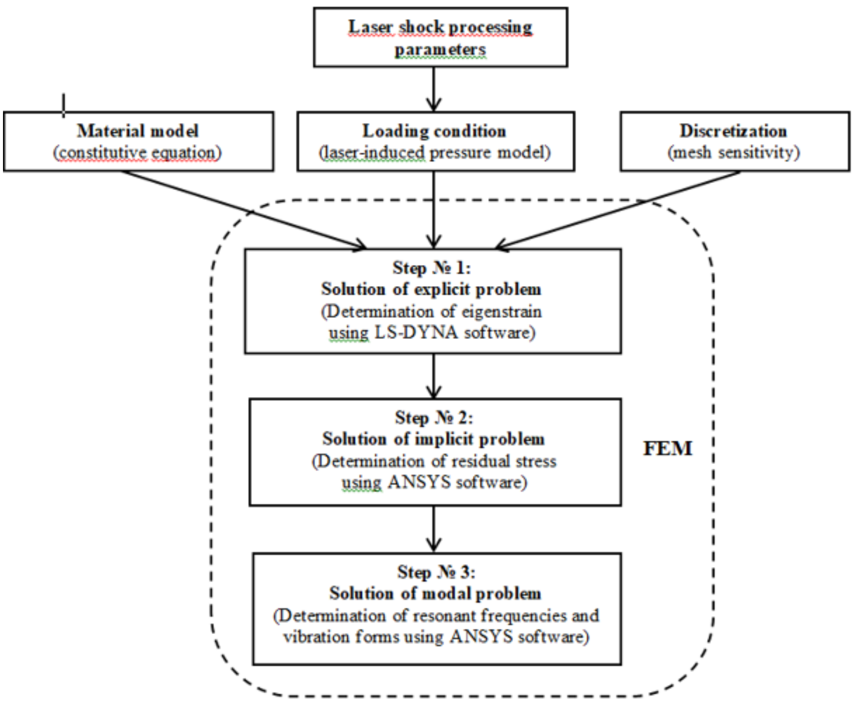

Figure 2: Strategy of finite element modeling (FEM) in order to study the influence of LSP technology on the dynamic characteristics of the blades

4 Results and Discussion. Effect of LSP on Modal Stiffness

The out-of-plane displacements and von Mises stress for the first six modes observed by FEM are shown in Figs. 3 and 4, in which the local zones with stretching of the mid-plane are marked with white dashed boxes. It is noted that only the modes that are excited by the vibration in the direction perpendicular to the specimen are considered here.

This kind of specimen is considered here for the following reasons: (i) in the fourth and sixth modes, the biaxial bending, which means stretching of the mid-plane, occurs in the local zones (shown in Fig. 3 (d) and (f) and Fig. 4(d) and (f)), and in particular, the fourth mode contains only biaxial bending, while does not contains uniaxial bending; (ii) the other modes in the study only contain uniaxial bending without stretching of the mid-plane, which are convenient for comparison; (iii) compared with actual blades, the specimen is more easily excited. It should be noted that stretching of the mid-plane is more commonly encountered in the higher-order modes of the blade compared to the specimen. The main purpose of using this specimen is to study the mechanism of selective modal control.

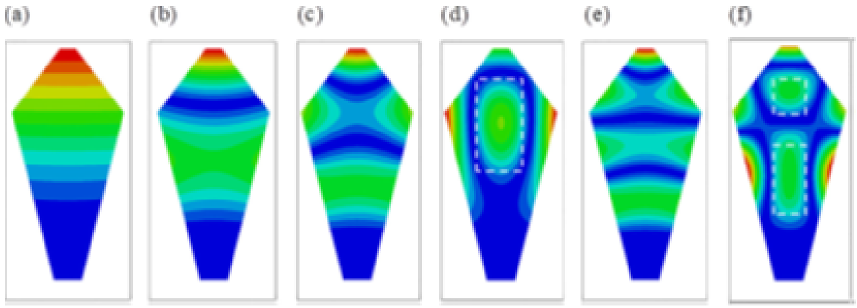

Figure 3: Out-of-plane displacements for first six modes

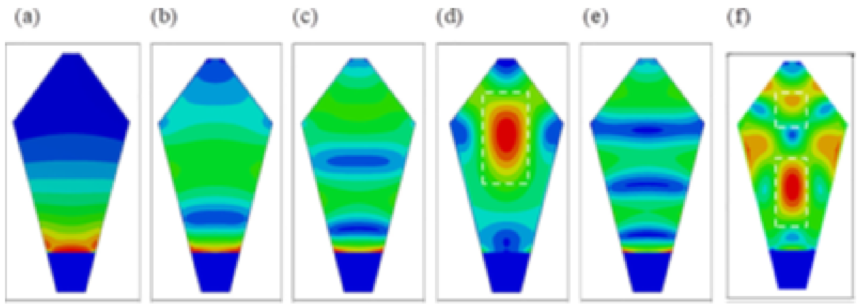

Figure 4: Von Mises stress for first six modes

The mechanism of the effect of LSP on modal stiffness is discussed in this study. The assumption that the change in modal stiffness of the structure vibration is caused by the combination of residual stress and the stretching of the mid-plane is raised.

The effect of LSP on the mechanical properties of a wide variety of materials has been well investigated in the scientific literature, most of which note that LSP has no obvious effect on the elasticity modulus [12]. At the same time, in a large number of studies on fatigue properties based on the vibration method, it is rarely discussed that the resonant frequency of the specimen changes after LSP treatment. This means that LSP does not affect the stiffness of the material, so the stiffness of the structure, especially a particular order mode, has to be changed by another method.

High amplitude residual stress is one of the main mechanisms for LSP to improve material properties [9-10]. The stretching of the mid-plane is often encountered in the higher-order modes of the blade. The interaction between the residual stress and the stretching of the mid-plane causes a change in the modal stiffness. A theoretical analysis is carried out to explain the mechanism and FEM simulations are used to demonstrate the assumption and compare the control results.

A simple geometric nonlinear model, which is a beam with two fixed ends, is shown in Fig. 5(a), in which the tensile and bending elastic forces provide the bending resilience of the beam. The resilience caused by the tensile elastic force is the main reason for the nonlinear vibration, so the resilience model is further simplified, considering only the tensile elasticity shown in Fig. 5(b), in which the stiffness of the springs is k and their length is l. When the point O in the role of external force F, generated displacement x, the force balance in O is shown in Fig. 5(c). We have:

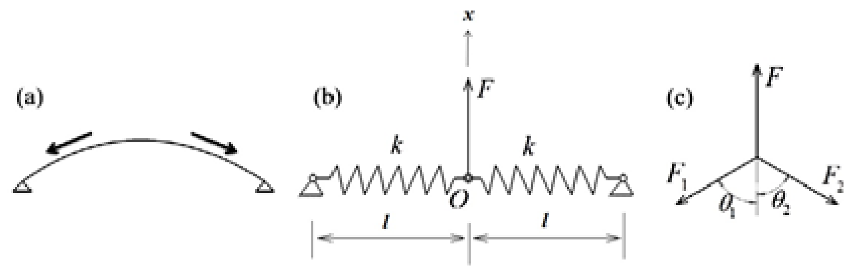

Figure 5: Effect of LSP on resilience

$$\theta_1 = \theta_2 = \arctan \frac{l}{x}, \tag{1}$$

$$F_1 = F_2 = k \left(\frac{x}{\cos\theta_1 - l} \right), \tag{2}$$

$$F = 2F_1 = \cos\theta_1, \tag{3}$$

$$k_0(x) = \frac{dF}{dx}, \tag{4}$$

where $k_0(x)$ is the stiffness of the model. Because the stiffness is nonlinear and varies with the displacement x of point O, it is expressed in the form of a function of x [3].

By substituting Eqs. (1)-(3) into Eq. (4), one obtains

$$k_0(x) = 2k - \frac{2kl}{\sqrt{l^2 + x^2}} + \frac{2klx^2}{(l^2 + x^2)^{3/2}}. \tag{5}$$

Expanding $k_0(x)$ into a Taylor series and keeping the first three stages, one obtains

$$k_0(x) = 6\frac{kx^2}{l^2}, \tag{6}$$

where the stiffness $k_0(x)$ only includes a nonlinear term.

The presence of residual stresses in the structure is equivalent to the springs in the model in Fig. 5(b) with preload. The preload in the springs is set to the compressive force F', and following the same method, the $k_0^{\prime(x)}$ under preload can be obtained:

$$k_0^{\prime(x)} = 2k - 2\frac{kl + F'}{\sqrt{l^2 + x^2}} + 2\frac{kl + F'}{(l^2 + x^2)^{3/2}}. \tag{7}$$

Expanding $k_0^{\prime(x)}$ into a Taylor series and keeping the first three stages, one obtains

$$k_0^{\prime(x)} = -2\frac{F'}{l} + 6\frac{kl + F'}{l^3}x^2, \tag{8}$$

where the stiffness $k_0^{\prime(x)}$ consist of a linear term $\left(-2\frac{F'}{l}\right)$ and a nonlinear term $\left(6\frac{kl+F'}{l^3}x^2\right)$.

Comparison of Eq. (6) with Eq. (8) shows that the preload can affect the stiffness in both the linear and nonlinear terms.

Next, FEM simulations (using LS-DYNA and ANSYS packets, Fig. 2) are carried out on the effect of residual stress on the stiffness of the structure. First, the resonance frequencies of a beam model with two ends fixed is calculated and the results with and without geometric nonlinearity, which means the stretching of the mid-plane is considered and not considered, are compared, which is used to demonstrate the theoretical analysis.

In the case of the geometric nonlinearity on or off of ANSYS, the resonant frequencies of the first five modes under different residual stresses are given in Table 1. From the FEM results, we can conclude that the stiffness of the structure is only affected by residual stress with the geometric nonlinearity on, and the degree of influence is different for different mode shapes, as the result of the different degrees of stretching of the mid-plane appearing in those modes, which demonstrate that the combination of the residual and geometrical nonlinearity cause the resonance frequency change of some modes of the structure.

Table 1. Resonant frequencies (Hz) with the geometric nonlinearity on or off

Mode	Geometric nonlinearity on			Geometric nonlinearity off	
	0 MPa	200 MPa	-200Mpa	200 MPa	-200 MPa
1-st	1249,8	1344,2	1146,5	1249,8	1249,8
2-nd	3101,2	3135,5	3066,5	3101,2	3101,2
3-rd	3380	3510,4	3243,7	3380	3380
4-th	4888,2	4901,8	4874,6	4888,2	4888,2
5-th	6435,9	6501,2	6329,4	6435,9	6435,9

Then, the residual stress is applied to the model of the specimen in three options, shown in Fig. 6, in which the shaded zones with dimensions are the zones with residual stress. The zone with residual stress in option (a) corresponds to the zone with stretching of the mid-plane of the fourth mode, and the zone with residual stress in option (b) corresponds to the zone with stretching of the mid-plane of the sixth mode. Without loss of generality, the zone with residual stress in option (c) is selected at the root of the specimen according to the traditional LSP scheme. The maximum in-plane (abs) principal stress is shown in Fig. 7.

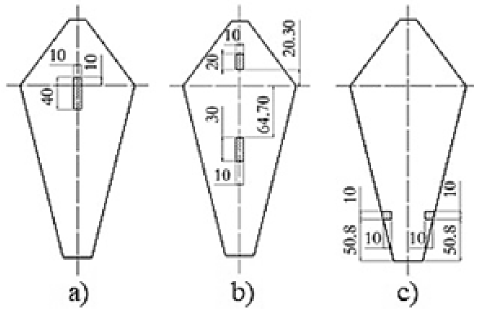

Figure 6: The treated zones of options (a), (b) and (c)

The resonant frequencies (Hz) of the untreated specimen and the specimen with the residual stress are shown in Table 2. The variation of the resonant frequency of each mode is shown in Fig. 8, where only the resonance frequency of the fourth mode of option (a) and the resonance frequency of the sixth mode of option (b) are significantly affected. This demonstrates that the interaction between the stretching of the mid-plane of the mode shape and the residual stress could be used to change the resonance frequency of the targeted mode.

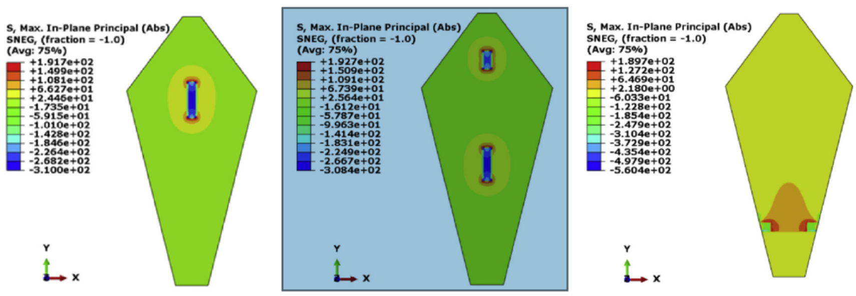

Figure 7: FEM results for residual stresses of specimen in options (a), (b) and (c)

Table 2. Resonant frequencies (Hz) of the untreated specimen and the specimens with the residual stress

Options	Mode					
	1-st	2-nd	3-rd	4-th	5-th	6-th
Untreated	28,3	190,3	504,36	857,06	978,67	1501,6
a	28,34	194,75	525,78	782,91	995,98	1511,4
b	28,45	194,96	509,65	850,66	996,19	1446,4
C	27,83	190,24	504,07	854,98	977,41	1498,8

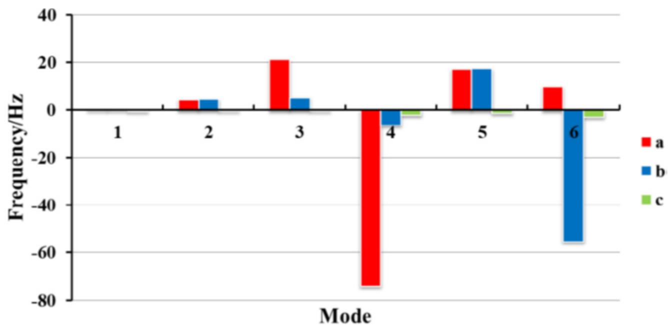

Figure 8: Variation of the resonant frequency of each mode

5 Conclusions

This work explored the selective modal control of blade vibrations for the mode shapes with stretching of the mid-plane by local LSP. The following conclusions can be drawn from the theory and simulation results:

On the basis that the mode shapes differ significantly from each other, the modes with stretching of the mid-plane could be selected for control by LSP in the local zones. The interaction between the stretching of the mid-plane of the mode shape and the residual stress induced by LSP causes a change in the modal stiffness, and then the resonance frequency of the targeted mode is changed.

The results have demonstrated that the use of LSP could selectively control the modes with stretching of the mid-plane from changing the modal stiffness and the vibration nonlinearity.

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