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RESEARCH ARTICLE

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Vibration analysis of blade-disc coupled structure of compressor

Abstract While a 3D assembly model of blade-disc structure was established, a finite element model for calculating the vibration characteristics during blade-disc coupling was built by taking into consideration the coupling action of contact stress between the blade and the disc. The vibration characteristics of the blade-disc coupling structure was calculated and analyzed using cycle analysis method with the aid of ANSYS software. The modeling experiment shows that this method is feasible for analyzing the rabbet assembly structure.

Keywords aerospace propulsion system, blade, disc, rabbet assembly, coupling vibration, finite element analysis

1 Introduction

Since the aero-engine tends to have high rotating speed and less weight, the blade-disc structure is becoming thinner and lighter, which brings about severe oscillation of the blade and the disc. Moreover, the dynamic characteristics of the blade-disc will be influenced by the oscillation. For a long time, the design and analysis of the dynamics of the blade and the disc separately, or the blade-disc structure as an entire part, have been performed [1], while the assembling and mating relationships between the two parts have been disregarded. As a result, there exist distortions on the predicted frequencies of the structure, which has resulted in unreasonable selection of design parameters. Therefore, the coupling vibration characteristics of the blade-disc structure have to be studied. In this paper, the validity of the finite element (FE) method for the modal analysis of the rabbet assembly structure considering contact effects was verified by an experiment. Then based on wave transmitting technology, the natural vibration characteristics of some blade-disc coupling assembly structure were analyzed using a FE program with cycle symmetry method.

2 Verification of FE method for analyzing rabbet assembly structure

The vibration of the rabbet assembly structure has always been a pivotal issue during the performance analysis of the compressor. This paper proposed a FE method considering the contact effects between blades and the disc for analyzing the vibration of the rabbet assembly structure. Meanwhile, an oscillating experiment for the rabbet assembly structure using hammer bump method was conducted to verify the simulation results [2]. A comparison of the results obtained from the simulation with those from the experiment shows that the FE method based on the contact pre-stress was feasible.

2.1 FE analysis of rabbet assembly structure

The FE model of the rabbet assembly structure built with the help of ANSYS software was shown in Fig. 1. In this model, element Solid45 was adopted for all 3D entities. Besides, the 3D, 8-node and surface-surface contact element Conta174 along with the 3D target element Targe170 were employed on the contact surfaces between part 1 and part 2. It was assumed that the contact behavior was an aluminum-aluminum one. To make sure that the FE model approximates the real situation, a uniform distributed pressure with the value of 1000 Pa was applied to the bottom of part 1, which simulated the centrifugal force caused by the rotation. After analyzing the FE model in ANSYS, the 1–4 order natural frequencies of the rabbet assembly structure were obtained.

2.2 Modal experiment of rabbet assembly structure

To realize a static measurement in place of that under a rotating state, the centrifugal force caused by the rotation

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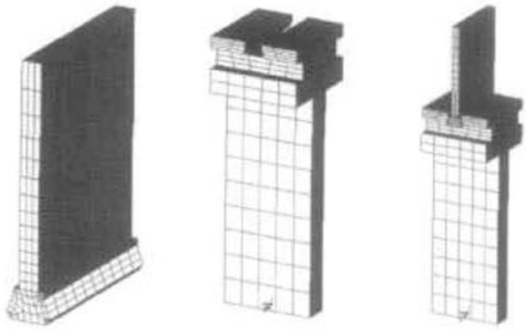

Fig. 1 Part 1, part 2 and dovetail joint assembly

was simulated by a thrust force acting on the rabbet of workpiece 1 from the bolts fixed on workpiece 2. The forces between the blade rabbet and the dovetail slot in the disc under different rotating speeds were then acquired by changing the preload of the bolts. The first four order mode results obtained from the experiment were compared with those from the FE analysis, as shown in Table 1, where, f_{FE} is the FE modal analysis frequency result, f_e the experimental modal analysis frequency results, ε the relative errors.

Table 1 Comparison of natural frequencies between simulation and experiment

mode order	f_{FE}/Hz	f_e/Hz	ε/%
1	692.81	578.453	16.500
2	1146.00	1130.827	1.324
3	1168.80	1164.700	0.035
4	2466.50	2401.280	2.600

It can be seen from Table 1 that there were small errors of the natural frequencies between the simulation and the experiment, with 4.1–65 Hz absolute errors and 0.04%–2.6% relative errors, except the one of the first order. It could be concluded that the FE method for the vibration analysis of the blade-disc assembly structure brought forward above was feasible. The factors leading to a higher error of the first order mode result include the errors caused by the sensitivity of sensors, errors occurred during the modal fitting, errors from the preload of bolts, and machining errors of workpieces.

3 Vibration of blade-disc coupling structure

To take the influence of the geometry of the blade on the mode of the coupling structure into account, an accurate geometry model was built with Unigraphics software and imported into ANSYS for FE analysis. The pre-stress caused by contact effects was calculated and the vibration characteristics under the action of this pre-stress were computed. The flow is shown in Fig. 2.

For a cycle symmetry disc with N blades, according to the theory of wave transmitting, one blade and its corresponding

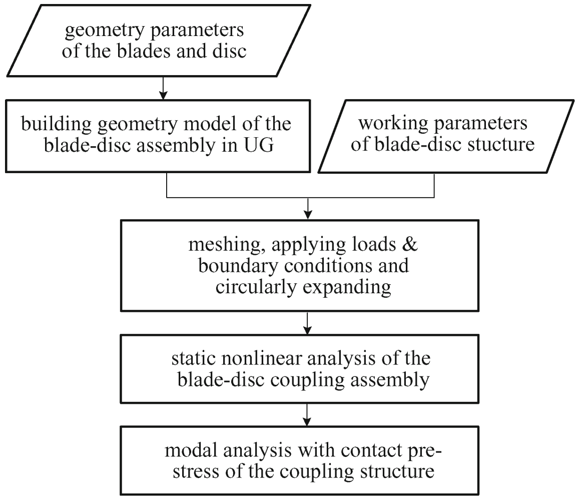

Fig. 2 Flow chart of modal analysis for blade-disc coupling structure

$1/N$ disc sectors can be used as the calculating region (basic sector). By calculating the mode of the FE model of this basic sector and introducing complex constraints to recon in the influence of other parts of the structure on the entire model, the Hermit eigenvalues were deduced. Then the vibration characteristics of the whole blade-disc structure were obtained [3,4]. Since there were some fixing angles between the blade and the disc, it was impossible to form a sector region along two radial directions. Planes that have certain angles with the disc axis were used as wave transmitting interfaces so as to produce a basic sector of the blade-disc assembly model, which was then imported into ANSYS and the mesh. As the dovetail rabbet was symmetric, the pressures acting on the two contact surfaces should have the same values in opposite directions. However, in reality, there are stress concentration and dynamic stresses. Therefore, a contact should be added between the rabbet and the slot when meshing the model in ANSYS. A basic sector of FE model was shown in Fig. 3. The FE model of the whole blade-disc structure was created as shown in Fig. 4 by circular extension of the basic sector.

Boundary conditions: freedoms were coupled between the back of the blade rabbet and the rabbet slot of the disc to simulate the axial constraint on which the snap ring acted on the blade. The location where the disc center hole and the shaft were coupled was completely fixed. Loads: the centrifugal force caused by rotation and the contact stress between the rabbet and the slot because of the centrifugal force. The maximum rotating speed of the disc was 27000 r/min.

4 Results and discussion

4.1 Frequency results

The first five order mode results of the structure under several different rotating speeds were calculated. The

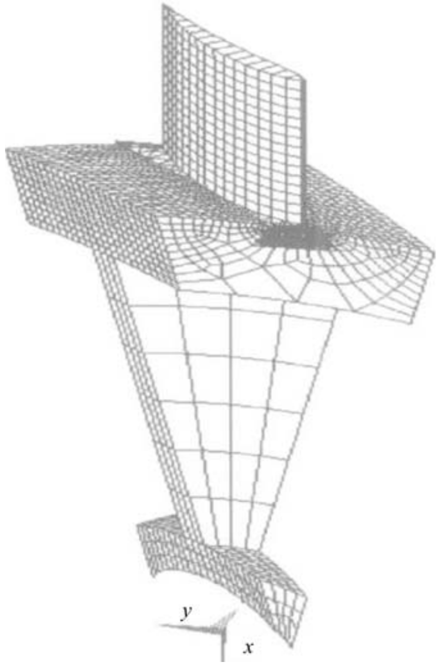

Fig. 3 FE model of basic sector

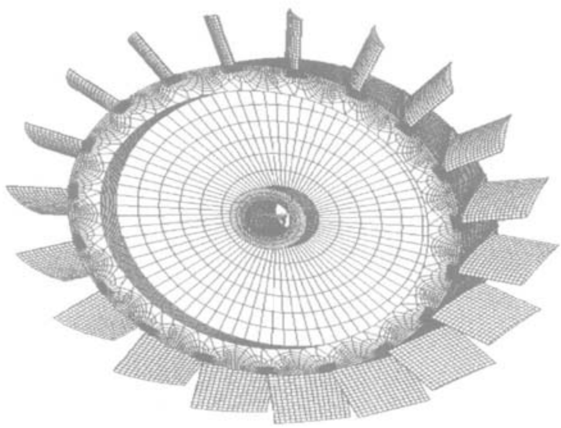

Fig. 4 FE model of whole blade-disc structure

frequency results at 1–3 and 9 pitch diameter locations are listed in Table 2.

As shown in Table 2, the blade-disc coupling structure oscillated linearly relative to pitch diameter numbers. The contact stress between the blades and the disc increased as the rotating speed ascended. Hence, the contact stiffness effect that took place between the blades and the disc intensified. As a result, the natural frequencies of the coupling structure increased along with the increase in rotating speed.

4.2 Mode shapes

Mode shapes of the blade-disc coupling vibration under non-rotating state are shown in Table 3.

Comparing the mode of a cantilever blade with that of the blade-disc coupling structure, it was found that the mode shapes of the blade in the blade-disc coupling structure were ahead of those of a cantilever blade when oscillating. That is, when the frequency of the coupling vibration was between the n-1 and n order natural frequencies of a cantilever blade, its mode shape was similar to that of the n order mode.

4.3 Coupling vibration analysis

The resonance of the vibration of the blade-disc coupling structure occurs when: the natural frequencies are equal to the exciting frequencies; the harmonic number of the exciting force k is identical with the pitch diameter number m. In the actual course of work, the first order vibration of the blade-disc coupling structure was the most prominent. Thus the resonance plot of the first order within the working speeds was provided in Fig. 5, f_n is the natural frequency. As shown in the plot, from the minimum rotating speed (n) to the maximum, the first order frequency curves of the 1–3 pitch diameters were

Table 2 Natural frequencies of blade-disc coupling structure Hz

pitch diameter number	rotating speed/r·min^{-1}	mode level				
		1	2	3	4	5
1	0	340.23	1320.3	2319.2	2659.9	3069.7
	11000	390.27	1386.2	2322.2	2790.3	3102.2
	19000	474	1506.9	2325.3	3006.3	3177.8
	27000	579.07	1669.2	2331.7	3194.7	3366.3
2	0	966.49	1440.5	3050.4	3143.8	3824.5
	11000	1009.9	1503.9	3090.2	3320.5	3828.5
	19000	1090.2	1623.6	3150.5	3645.2	3838.4
	27000	1201.2	1792.6	3236.9	3840	4074.3
3	0	1271.2	2556.2	3094.1	3969.9	5334.1
	11000	1335.6	2588.3	3123.3	4195.4	5348.6
	19000	1454.6	2651.1	3181.7	4598	5378.5
	27000	1618.7	2744.9	3269	5112.8	5437
9	0	1301.5	3061.8	6967.3	7036.9	9667.1
	11000	1368.1	3090	7008.7	7075.1	9717.3
	19000	1491.8	3146.7	7045.2	7196.9	9818.9
	27000	1663.7	3231.4	7084.2	7395.8	9969.7

Table 3 Mode shapes of blade-disc coupling structure

pitch diameter number	mode order		
	1	2	3
0	umbrella vibration of the disc	torsional vibration of the disc	bending vibration of the blades
1	disc vibration mainly	bending vibration of the blades	umbrella vibration of the disc and bending vibration of the blades
2	disc vibration mainly	bending vibration of the blades mainly	torsional vibration of the blades mainly
3	bending vibration of the blades	torsional vibration of the blades mainly	torsional vibration of the blades
5–9	the first order vibration of the blades	the first order torsional of the blades	combined vibration of the blades mainly

far from the exciting frequency curve. While those of the 5–9 pitch diameter had intersections with the exciting frequency curve. Therefore, the first order vibration occurred easily at the 5–9 pitch diameters.

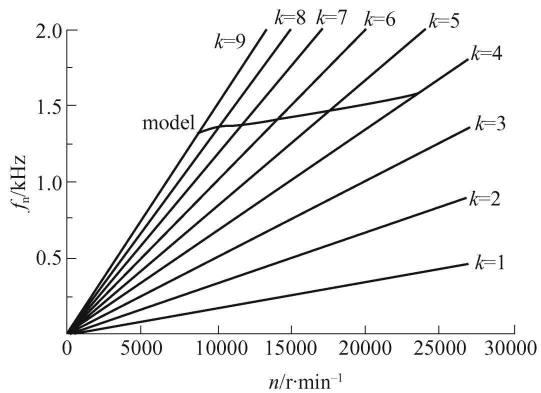

Fig. 5 Coupled vibration of blade-disc structure

5 Conclusions

1) The vibration characteristics of the blade-disc coupling structure were obtained by using cycle symmetry analysis method, based on wave transmitting technology. The results will be useful for the prediction and analysis of the practical vibration of the blade-disc coupling structure.

2) The results of the coupling vibration of the blade-disc under the action of contact pre-stresses, which were caused by the centrifugal force between the blade and the disc because of the rotation, were more reasonable. The results show that, under the action of the contact prestress, the stiffness and the natural frequency of the blade-disc structure increased as the rotating speed ascended gradually. The influence weakened when the rotating speed reached a certain extent.

3) Because of the coupling between the blade and the disc, the high-order natural frequencies of the blade-disc structure were close, and the mode shapes were more complicated. The mode shapes of the blade were ahead of those resulting from the vibration of a single blade.

References

1. Zhou Chuanyue, Zou Jingxiang, Wen Xueyou, et al. Calculation of the blade-disc coupled vibration characteristics of a gas turbine. Acta Aeronautica Et Astronautica Sinica, 2000, 21(6): 545–547 (in Chinese)
2. Zhao Meng, Zhang Yidu, Ma Liangwen, et al. Research on modal analysis of assembly structures based on MSC/Nastran. Journal of Vibration and Shock, 2005, 24(1): 28–30 (in Chinese)
3. Thomas D L. Dynamics of rotationally periodic structures. International Journal of Numerical Methods in Engineering, 1979, 14(1): 81–102
4. Zhang Jin, Chen Xiangjun, Wang Wenliang. Cyclo-symmetric modal synthesis of rotating bladed disc system. Acta Mechanical Solid Sinica, 1992, 13(3): 225–234 (in Chinese)