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DYNAMIC ANALYSIS OF FIRST STAGE COMPRESSOR BLADED DISK OF AN AIRCRAFT ENGINE

Key words

Rotor blades, bladed disk, forced vibration.

Summary

There have been a number of SO-3 first stage compressor blade failures. An experiment was carried out on a first stage rotor blade in the compressor of an SO-3 engine at the Air Force Institute of Technology in Warsaw. The natural frequencies of rotor blades were measured. Next, the tip-timing measurement of rotor blade was done in an engine working at various speeds. Various operation conditions were assumed, such as the presence of a foreign object in the engine inlet. Structure and flow calculations were carried out to compare the experimental results. An FEM was used to calculate the natural frequencies of a mistuned bladed disk. The FLUENT code was used to calculate the unsteady forces acting on rotor blades. The mistuned bladed disk was forced to vibrate when rotating at 15000 rpm.

Introduction

Investigations into the effect of blade mistuning on free and forced bladed disk vibrations have been presented by Wagner [1], Dye and Henry [2], Ewins

[3], El-Bayoumy and Srinivasan [4], Rzadkowski [5], Wei and Pierre [6], Castanier and Pierre [7], Lombard et al. [8], He et al. [9], Avalos and Mignolet [10], Ganine et al. [11], Avalos et al. [12], Kaneko et al. [13], Petrov et al.[14], [15].

In most of the above articles, the mistuning was not based on measurements taken from real gas turbines.

In this paper, the free and forced vibrations of real tuned and mistuned bladed disks of the first compressor stage of an ISKRA aircraft SO-3 engine are presented. The numerical results are compared with experimental ones.

1. Experimental and numerical results

Experiments were carried out on a first stage rotor blade in the compressor of an SO-3 engine at the Air Force Institute of Technology in Warsaw to measure the blade amplitude [16].

The rotor blade and disk of the first stage was made of steel 18H2N2. The blade length is L_o =0.1063 [m], the radius of blade attachment in the disk is R_o = 0.2077 [m], the number of rotor blades in the stage is N_2 = 28 and the number of stator blades is N_1 = 34.

FEM was used to calculate the natural frequencies of the blades and bladed disks. Eight-node izoparametric elements were used in ANSYS to model the blades and bladed disks.

Three FEM models of bladed disks were considered. Model A, a tuned bladed disk, where all the blades have a first natural frequency of 346 Hz. Model B, where the natural frequencies of blades change from 334 Hz to 359 Hz and the rate of mistuning is 3.6%. Model C, where the frequencies of blades change from 318 Hz to 385 Hz and the rate of mistuning is 9.53% (Fig. 1) [16].

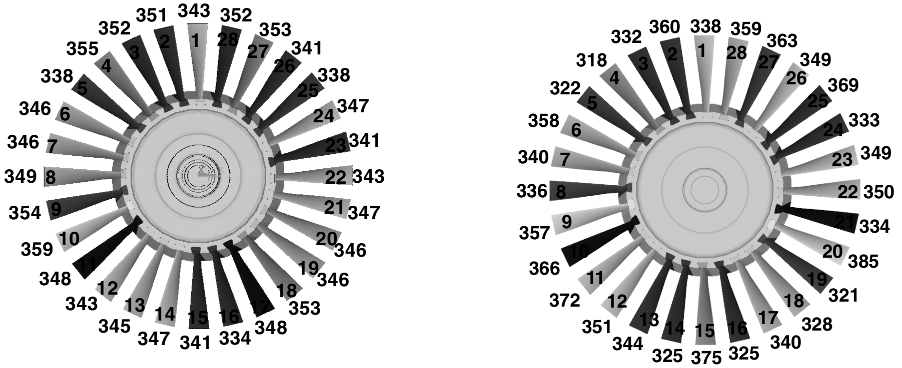


Fig. 1. Blade mistuning pattern in Model B and Model C

The rotor blade mistuning patterns of Model B and Model C are taken from measurements of the first natural frequencies of blades in a real bladed disk in an SO-3 engine first compressor stage [16].

In the experiment, a crack was initiated in the 1st compressor stage with 9.53% blade mistuning [16] by placing rectangular blocks (125x100 mm) on the stator blades, which in real life could be caused by birds engulfed in the engine. In the experiment, the maximal amplitude of blades was observed at different rotation speeds from 15000 to 16000 rpm. Rotation stall was observed at speeds between 7000 and 10000 rpm.

2. Free vibration

The natural frequencies of the bladed disks of models A, B, and C were calculated using the ANSYS code (

Table 1).

The mistuning was modelled by changing the Young modulus of the blade. The natural frequency of a single blade in Model A equalled 346 Hz; therefore, the natural frequencies of the bladed disk were less than 346 Hz in the first series.

The modes of the bladed disk were classified by using an analogy with axi-symmetric modes, which were mainly characterised by nodal lines lying along the diameters of the structure and having a constant angular spacing. They were either zero (k = 0), one (k = l), two (k = 2), or more (k > 2) nodal diameter bending or torsion modes. Series 1 was associated with the first natural frequency of the single cantilever blade. Series 2 was associated with the second natural frequency of the single cantilever blade; and so on, k is the number of nodal diameters.

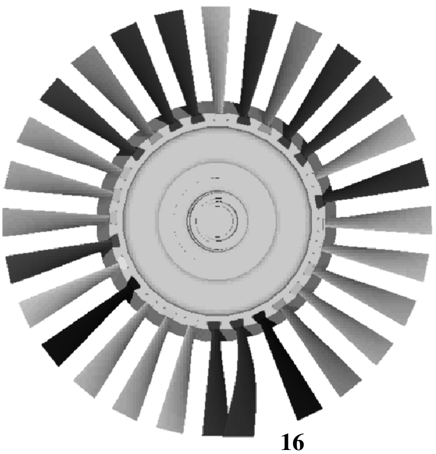
Table 1. Natural frequencies for three models of bladed disks for non rotating systems

Model A 0% of mistuning		Model B 3.6% of mistuning		Model C 9.5% of mistuning	
[Hz]		[Hz]		[Hz]	
0 RPM	15000 RPM	0 RPM	15000 RPM	0 RPM	15000 RPM
342.20	497.00	331.94	497.23	316.10	486.62
342.20	497.02	335.62	499.46	319.12	488.64
342.71	497.02	335.85	499.82	323.11	489.56
342.71	498.47	338.76	501.82	323.65	491.15
343.00	498.47	339.03	501.92	326.11	492.07
344.33	500.24	339.18	502.41	330.05	493.59
344.33	500.24	340.84	503.29	330.84	496.16
344.62	500.49	340.94	503.57	332.14	496.53
344.62	500.49	341.18	503.79	333.78	497.72
344.72	500.56	342.93	504.78		498.60
344.72	500.56	343.22	504.95		500.10

For the tuned bladed disk, double modes of vibration appeared in cases with up to 14 nodal diameters for 28 rotor blades from different series.

It is known that mistuning causes the splitting of the doubled bending frequencies of the tuned system. For example, the 342.20 Hz mode of Model A splits into 331.94 Hz and 335.62 in Model B and 316.10 Hz and 319.12 Hz in Model C. But mode shapes are difficult to associate with the nodal diameters, because of the considerable rigidity of the disk.

Fig. 2 presents the 331.94 Hz mode shape (Model B). In this mode, only one blade (number 16, frequency 334 Hz Fig. 1) is vibrating.



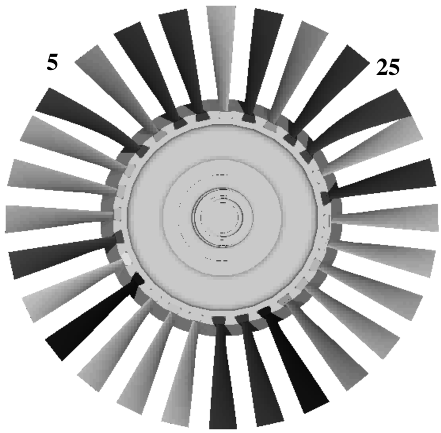


Fig. 2. Mode shape at 331.94 Hz in Model B, mode shape at 335.62 Hz in Model B

In case of the 335.62 Hz mode (Model B), only two blades vibrate (5 and 25, Fig. 1, natural frequency 338 Hz). Both these blades vibrate at 335.62 Hz, but blade Number 5 vibrates with greater amplitude. In the 335.85 Hz mode, only two blades vibrate, but this time blade 25 (with natural frequencies 338 Hz) is the one that vibrates with greater amplitude. The similar behaviour is seen for the modes in Model C. In the case of the 316.10 Hz mode and the 319.12 Hz mode only one blade vibrates. In the 316.10 Hz mode, only blade 4 vibrates; whereas, in the 319.12 Hz mode, only blade 19 vibrates.

3. Forced vibration

In order to compare the experimental results with numerical results, forced vibration analyses were carried out on tuned and mistuned bladed disks. The analyses were carried out with a foreign object placed on the stator blades and also without the foreign object.

The unsteady forces acting on the rotor blades were calculated for the 3D non-viscous flow of ideal gas (15000 rpm) through the stator-rotor-stator stage using the FLUENT code [17].

A 3D model of the first stage of an SO-3 jet engine compressor is shown in Fig. 3. The model, created using the Gambit program, consists of 44 blades in the Inlet Stator Cascade, 28 blades in the Rotor Cascade (only one of which is seen in the picture) and 34 blades in the Stator Cascade of the first stage. The reference rotor blade in Fig. 3 is divided into 10 cross-sections.

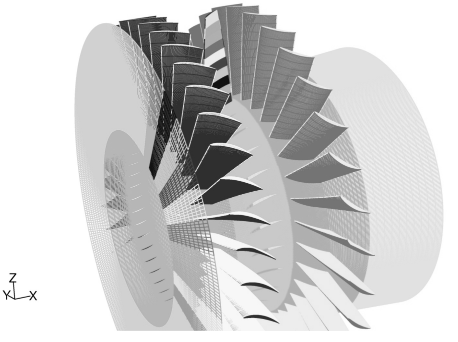


Fig. 3. CFD model of an SO-3 engine first stage compressor

In order to model the block (125x100 mm) in the inlet of the engine, 1/4 of the area was divided into 11 segments. During the simulations, two operating states were analysed. The first was the nominal state (fully-opened inlet), the second was the state in which four of these segments were blocked.

Fig. 3 presents a comparison of results for an unblocked and four blocked inlet segments. It is clearly visible that the blocked inlet segment has a strong local influence on the amplitude.

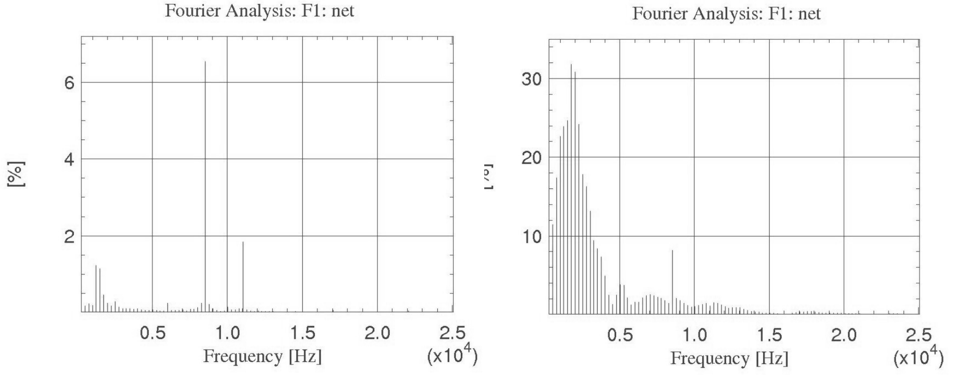


Fig. 3. The unsteady axial force harmonics for nominal state and for four-blocked inlet segments (comparison of blocked and unblocked inlet)

The four-blocked inlet segments caused low-frequency harmonics of a higher amplitude (30% of the steady part) than the unblocked with just 1.3% of the steady part (Fig. 4).

Based on CFD calculations, 2EO excitation was applied to the blades in all the bladed disk models (A, B, and C) in axial and circumferential directions.

The calculations were carried out at 14000 to 15400 rpm for the 0.2 Hz step, with modal damping at 0.005. The maximum value of blade stress on the leading edge in the first cross-section for a tuned bladed disk was 260 MPa for the four-blocked segment.

The maximal bladed stress increase for the mistuned bladed disk (3.6% in Model B) was 36% in relation to the tuned disk (Fig. 5).

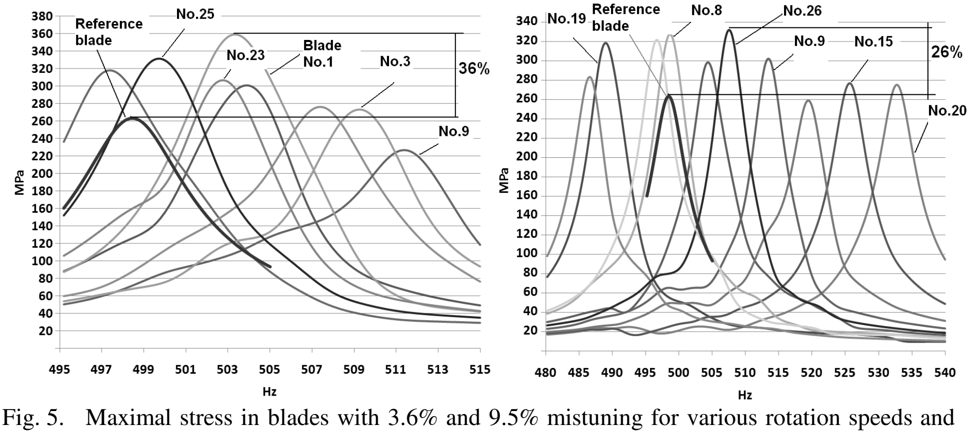


Fig. 5. Maximal stress in blades with 3.6% and 9.5% mistuning for various rotation speeds and for tuned bladed disk (reference blade)

Mistuning causes additional resonances associated with the vibration of individual blades. Thus, in Fig. 5, we can see the resonance peaks for individual blades. The blade number appears above the resonance curve. In the tuned bladed disk, we see only one resonance peak for all blades (reference blade).

The maximal values of stress appeared in blade 1 with a natural frequency of 343 H, and 503.57 Hz for excitation. The maximum amplitude was approximately 360 MPa.

Close to 503 Hz, other blades on this disk with similar frequencies (12, 15, 22, 23) were also deformed. For excitation frequency 499.6 Hz, the maximal amplitude appeared on blade 25 (Fig. 5). Generally, each mistuned blade on the disk experienced its maximum at a different time during run-up.

The maximal bladed stress increase for the mistuned bladed disk (9.5% in Model C) was almost 26% in relation to the tuned disk (Fig. 5, reference blade). As in the previous case, mistuning causes additional resonances associated with the vibration of individual blades. In this case, the situation was similar.

The maximal values of stress appeared in blade 26 with a natural frequency 349 Hz for 507.88 Hz excitation. The maximum amplitude was approximately 330 MPa. Other blades with similar frequencies also become deformed on this disk.

Conclusions

In this paper, the free and forced vibration of real tuned and mistuned bladed disks of an SO-3 engine first compressor stage were presented. The FLUENT code was used to calculate the unsteady forces acting on the rotor blades.

There were three bladed-disk models. In Model B (3.6% mistuning), the maximal blade stress increased by 36% in relation to the tuned bladed disk (Model A); whereas, in Model C (9.5% mistuning), maximal blade stress increased by 26%. In Model C, the maximal stress values were similar to the experiment for four blocked segments in the inlet one sector.

Nevertheless, the numerical models confirmed the experimental results in that the maximal stress in blades did not occur simultaneously.

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Analiza dynamiczna ułopatkowanej tarczy pierwszego stopnia sprężarki silnika lotniczego

Słowa kluczowe

Łopatka wirnikowa, ułopatkowana tarcza, drgania wymuszone.

Streszczenie

Odnotowano wiele uszkodzeń łopatek wirnikowych pierwszego stopnia sprężarki silnika SO-3. W Instytucie Technicznym Wojsk Lotniczych w Warszawie przeprowadzono cykl eksperymentów na łopatkach wirnikowych pierwszego stopnia sprężarki silnika SO-3. Zmierzono częstotliwości drgań własnych łopatek wirnikowych. Wykonano pomiary przemieszczeń łopatek wirnikowych metodą tip-timing dla różnych prędkości obrotowych wirnika. W trakcie badań uwzględniono także obecność obcego ciała we wlocie silnika. Przeprowadzono obliczenia numeryczne przepływu w pierwszym stopniu sprężarki i drgającej ułopatkowanej tarczy, w celu porównania z wynikami otrzymanymi z badań eksperymentalnych. Metoda elementów skończonych zastosowana została do analizy drgań wymuszonych ułopatkowanej tarczy z łopatkami różniącymi się wymiarami geometrycznymi, dla nominalnej prędkości obrotowej n = 15 000 obr./min. Program FLUENT zastosowany został do obliczenia niestacjonarnych sił działających na łopatki wirnikowe.