VIBRATION STRENGTH INCREASE OF ROTATING BLADES USING DYNAMIC SPIN RIGS

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Abstract

Specific features of dynamic spin rigs (DSR) are discussed. Some results of gas turbine engine rotor tests at DSR are presented. It is shown that DSR can be effectively used to provide turbomachine vibration strength.

1 Introduction

A major portion of gas-turbine engine (GTE) defects detected in service is due to insufficient vibration strength of parts. Defects connected with high cycle fatigue (HCF) of GTE parts give a significant decrease in economical efficiency of aircraft operation and negatively influence on safety of single-engine aircraft operation [1].

When designing the advanced engines, problem of prevention of parts fracture as a result of HCF becomes much strained because of rise in pressure change in one stage of turbomachine; application of wide chord blades having dense spectrum of natural frequencies; use of assemblies possessing low structural damping (blisks, blings); enhancement of vibrations coupling; application of materials having insufficient plasticity and being defect-sensitive; increase in static stresses and temperature of parts; rise in engine running time without removal from wing. It is well to bear in mind well known difficulties associated with prediction and experimental validation of engine parts vibration strength.

Currently, technologies to prevent HCF fracture of engine parts especially turbomachine blades, are developing intensively [1].

Efficiency of vibration damping, location of fracture origin and fatigue strength of material depend on static stresses that are caused by rotor rotation. Because of this, there is need to carry out investigations of vibration strength of blades in centrifugal-force field. Engine testing consumes considerable costs and time. What is more, in this case it is difficult to make required measurements. These combined points determine actuality to carry out some vibration strength investigations on spin rigs with excitation of rotating blades vibrations.

2 Dynamic spin rigs

Experience shows that spin rigs may be effectively used to provide vibrations strength of rotor parts. In particular, dynamic spin rigs (rigs on which rotating parts vibrations are excited) are widely used for investigation of vibration characteristics of blades and rotors and optimization of vibration damping. Besides, to carry out HCF rotating blades testing it is necessary to provide effective vibrations excitation, high accuracy of adjustment and holding of speed of rotor under investigation (to maintain required resonance regime), possibility to collect and process information about vibration characteristics of object under investigation.

Excitation of rotor parts vibrations performed on dynamic spin rigs (DSR) may be realized by different means, in particular [1-8], with the use of:

- kinematic link between rotor under investigation and electrodynamic vibrator;

- static or rotating (from independent drive) stationary air jets (at rotor rotation in partial vacuumed chamber or in atmospheric conditions);
- turbulators or generators of air impulses (at rotor rotation in partial vacuumed chamber or in atmospheric conditions);
- piezovibrators fixed on parts (in doing so, efficiency of vibrations excitation can be insufficient for fatigue tests);
- liquid (oil) jets or oil mist;
- eddy currents (of permanent magnets);
- vibrations excitation through magnetic bearings of rotor supports.

Investigation of vibration strength of rotating rotor blades are carried out at CIAM with the use of different DSR [5-8]

In Fig.1 is shown developed at CIAM spin rig having horizontal rotational axis on which rotor rotates in atmospheric conditions [5]. Rotor under investigation is driven into rotation by two direct current electric motors operating to one shaft. Heating of blades is realized through air friction of rotating blades. Vibrations excitation is performed with the use of turbulators or air nozzeles (Fig. 2).

Fig. 1. DSR Developed at CIAM Having Horizontal Rotating Axis.

(a) (b)

Fig. 2. Devices for Excitation of Rotating Blades Vibrations Through Turbulators (a) and air nozzle (b).

Number of vibrations exciters (nozzles, turbulators) is defined depending on the selected harmonic. Blade resonance response occurs when the product of rotor speed and number of exciters becomes equal or multiple of natural frequency of vibrations on the selected mode.

With the use of rig having horizontal rotational axis and rolled away armor chamber, assembling of test object becomes significantly simplified. Making of wiring terminals from strain gauges and thermocouples fixed on rotor being investigated in electric driver direction enables object preparation not to be disturbed at opening of rig chamber. The above stated points afford ease of rig handling when investigating vibration characteristics of rotating rotors and optimization of structural vibrations damping. But this rig can not be used to carry out HCF tests.

Figure 3 illustrates TDI-developed vacuum rig having vertical rotational axis which is also used at CIAM. Excitation of rotating blades vibrations is realized through air impulses (at rotating of object in chamber with partial vacuuming and temperature up to 800^{o}C) or by oil jets (in vacuum at temperature up to 200^{o}C, Fig.4) [3, 4, 6-8].

The analysis performed showed that application of oil jets is the most effective way to excite high frequency vibrations of turbomachine blades on complex modes. When blades vibrations is excited through oil jets, nozzles directed towards blades are arranged in periphery area of rotor being tested (Fig.4).

Fig. 3. DSR Developed at TDI Having Vertical Rotational Axis.

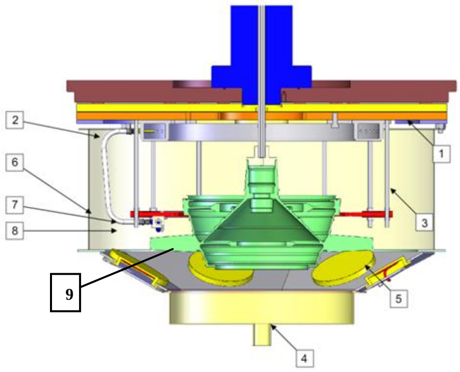


Fig.4 System of Excitation of Vibrations Through Oil Jets: 1-setting plate;2-oil pipe; 3-chamber; 4-oil reservoir; 5-heater; 6-exciter fixing tooling; 7-sets of nozzles; 8-nozzeles; 9-blade being investigated.

During testing liquid is continuously pumped through these nozzles so that the sprayed flow would strike rotating blades in assigned positions. Selecting the number of jets at assigned rotor speed value, it is possible to achieve that exciting force frequency (or multiplicity) will coincide with frequency of mode concerned. With such a coincidence, a blade responds at vibrational stress level that is regulated through liquid flow via nozzles control. In addition, efficiency of vibrations excitation depends on design of nozzles and their position relatively to parts being tested. The described system provides good (sufficient for blade fracturing) vibrations excitation level (even for vibrations on complex high frequency modes), however it is operable only at relatively low temperatures in view of oil ignition hazard. For this reason this system would be appropriate for use when carrying out tests of rotating blades without heating or with heating up to 200^{o}C including blades vibrating on high frequency modes. In case of exciting by liquid jets, the hazard exists of occurring drop wise erosion of blades. Efforts made to excite blade vibrations through oil mist for decrease erosion resulted in considerable reduction in efficiency of excitation. More effective way to excite vibrations is application of thin oil jets. To provide the possibility of exciting vibrations by oil jets, the rig is equipped with special systems of vacuuming of chamber and heating of oil that is used to excite vibrations of blades being investigated.

If it is necessary to carry out tests at high temperature (up to 800^{o}C), it is more preferable to excite vibrations by air impulses. Air impulses generator (Fig.5) provides excitation with the use of aerodynamic forces that are created as a result of transfer of kinetic energy from flow swirlers caused by rotating blades.

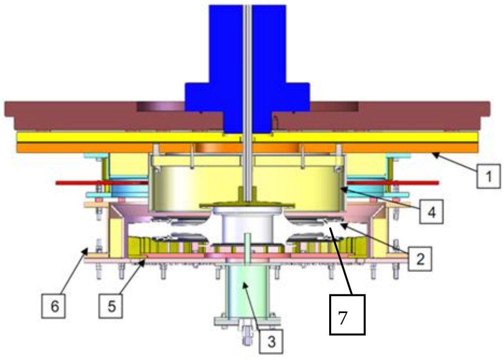


Fig. 5. System of Vibrations Excitement Using Air Impulses Generator: 1-setup plate;
2- chamber; 3-air reservoir; 4-heating device;
5-tooling for exciter fixing; 6-exciters; 7-blade under investigation

Periodical impulse of exciting force is generated when blade passes exciter. Chamber pressure is kept at a level sufficient to make required excitation with the use of precision vacuum monitoring system. Excitation amplitude is controlled through change in chamber pressure. This method provides lower efficiency of blade vibrations excitation as compared with the previous one but it enables the tests to be carried out at blade temperature up to ~800^{o}C. Thus, this system would be appropriate for use when carrying out tests at high temperatures, e.g. for turbine blade tests.

To make possible maintaining of resonance blade vibrations of required vibration stresses amplitude, the rig provides a means for precision controlling and monitoring of speed used as a driver of air turbine. With the purpose of getting information about vibrational and thermal conditions of rotor under investigation, rotor can be equipped with strain gauges and thermocouples whose signals enter recording equipment through high-speed slip ring. Since in the process of endurance HCF tests, strain gauges fixed on blade being investigated and (or) slip ring may become inoperative, the rig is equipped with two contactless vibration measuring systems (one system uses laser sensors, the other system uses eddy-current sensors). When counting loading cycles, it is possible to exclude cycles in which vibrational stress amplitude was lower than the required value by more than specified magnitude.

3 Test results

Considerable experience has been gained at CIAM on use of spin rigs for investigation of blade vibration strength in centrifugal forces field and optimization of design damping of rotor blades vibrations [5-8].

Blisks of different designs have been investigated [5]. One of blisks (Fig. 6) had blades with guaranteed clearance between periphery platforms. Removable inserted dampers differing in mass (m1=0,36g, m2=0,76g m3=1,2g) were placed under platforms. Rotor with partially inserted dampers (Fig. 7) has been also investigated (adjacent to blade having dampers in two sides, blades having one damper were placed, adjacent to which blades without dampers were placed). Rotor of second design had blades with elongated legs and two rows of lower platforms under which dampers of box design were placed. Mass of each of dampers placed at a less radius (leg length-to-airfoil length ratio ~0,18) is 0,8g, mass of each damper of second row is 1,0g. Rotors having one row of dampers (upper or lower) and two rows of dampers have been tested.

In addition, rotor without inserted dampers assembled with pressed down elastic disk-deflector has been tested.

Different ways of blades vibrations excitation (for the first flexure mod of vibrations) have been used, such as rotation of rotor being investigated in atmosphere and with the use of partial vacuuming (up to residual pressure of 0,1-0,2 atm), excitation of vibrations through bosses-turbulators (with no supply of air into nozzles) and with partial vacuuming (up to residual pressure of ~0,2 atm), supply of air at pressure of 6 atm into nozzles.

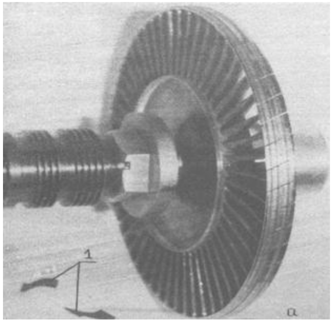


Fig. 6. Cast Turbine Blisk Having Slotted Periphery Platforms (1-damper)

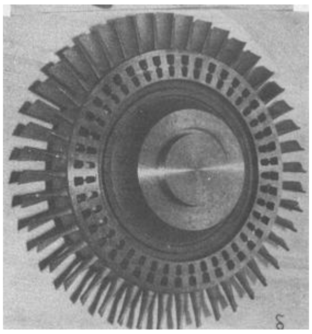


Fig. 7. Cast Blisk Having Two Rows of Lower Platforms

Test results of rotor of first design showed that placing of dampers of 0,36g mass under periphery platforms leads to increase in frequency of mode being investigated by a factor of 1,35, this conforms with results of specimens tests carried out on damper rig. Decrement of vibrations damperless rotor is ~ 0,01, i.e. is in level typical for cast rotors. When placing dampers of mass 0,36g, decrement increases significantly and vibrational stresses level decreases by approximately a factor of 4. Enhancement of damper mass up to 1,2g doesn't result in considerable change of vibration stress level . Rise in stresses when vibrations are excited with the use of partial vacuuming compared to rotate in atmosphere may be related to change in intensity of exciting harmonics when pressure in chamber decreases and (or) change in aerodynamic damping level. When making partial placing of dampers, stress level in damperless blades decreases by approximately, a factor of 2.

Investigation results of rotor with blades having elongated legs and two rows of lower platforms showed that placing of dampers at a lesser radius increased frequency by approximately 3%, at a greater radius – increased frequency by 6-10% and placing of two rows of dampers increased frequency by 10-13%; in doing so, relative frequency enhancement increased with rise of speed in the latter two cases. Decrement of vibrations of damperless rotor was about 0,01. Blade stress level was reduced with placing of dampers at a lesser radius by approximately a factor of 3 and at a greater radius by a factor of 4, 5-6. Two rows of dampers enabled blade stress level to be reduced by more than a factor of 15.

Use of pressed-down elastic disk-deflector that is not in contact with blades but is pressed down to bead on disk web with accepted tightening force (deflector web deflection at radius of 35mm was 0,05mm) caused stress level to lower by a factor of 1,5 at the same vibrations excitement conditions.

Results of investigations have been used for development of damper design methods.

Figures 8,9,10 presented the results of investigations of HPT rotor (with inserted unshrouded blades) vibration damping performed with the use of dampers located under path blade platforms [6].

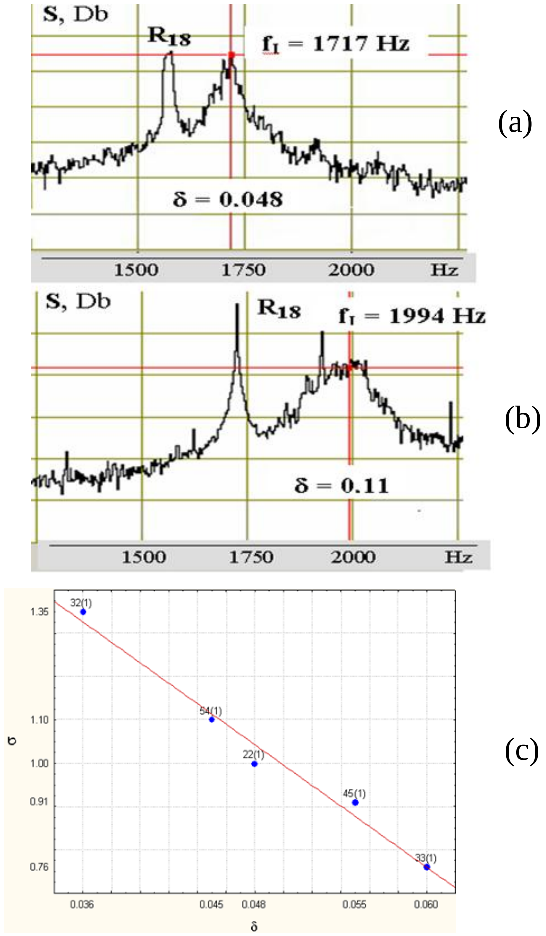


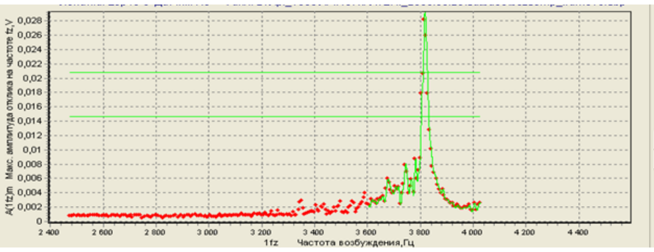
Fig. 8. Results of HPT-inserted Blades Vibration Damping Investigations (Noise Excitement). Vibration spectrums of damperless blade (a) and blade having a damper (b). Interrelation between relative level of resonance stresses and values of generalized vibrations decrement for different blades in rotor (digit indicate blade number and strain gauge code) (c).

The results obtained have been processed with the use of method of determination of generalized vibrations [6] decrements from spectrum at noise excitement (Fig. 8.) and method of plotting at resonance curve on a basis of tracking analysis (Fig. 9.)

The generalized decrements of vibrations for different blades in rotor have a considerable spread and their values correlate well with measured level of alternating stresses at resonance with harmonic of circumferential gas flow distortion. This is illustrated in Fig. 8c.

To determine logarithmic vibrations decrements at resonance regimes, method of resonance curve has been used. However, difficulties emerged during tests that were related with provision of conditions for plotting a resonance curve. To overcome these difficulties we used method of tracking analysis for strain gauging results to be proceed. Processing of signal being recorded in the process of experiment has been performed with the use of specialized software complex that is developed at CIAM and intended for processing of signal from vibration sensors (or strain gauges) that were recorded by digital recorder during tests in synchronous tracking analysis regime.

Figure 9 illustrates amplitude - frequency characteristics of damperless blade and of blade having a damper (for the flexural mode of vibration). It is seen that, with placing a damper, generalized blade vibrations decrement increases by nearly a factor of 2 and frequency shifts by 277Hz in increasing side.



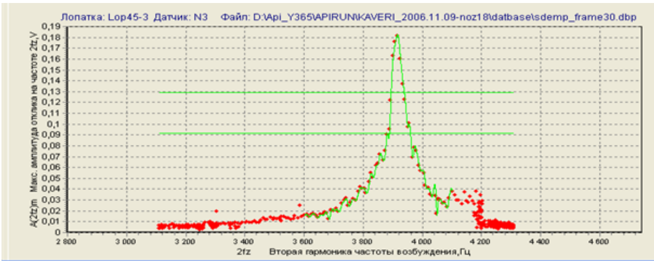
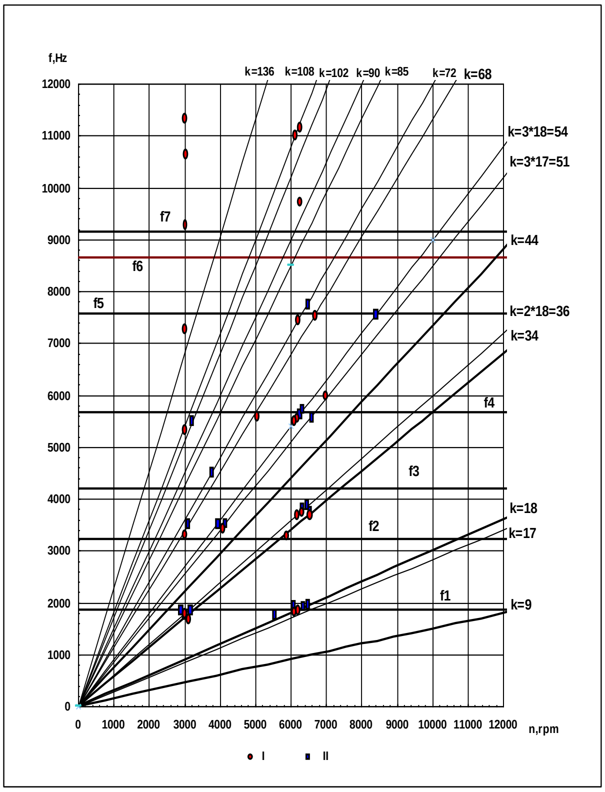


Fig. 9 Amplitude Frequency Diagrams of Inserted HPT Blades During Without Dampers (a) and With Dampers (b)

Logarithmic vibrations decrements for the flexural modes of vibrations have considerable spread in different blades. With presence of damper insertion, range of logarithmic dampers of vibrations on the same modes clearly shifts in increasing side. Increase in minimum values of logarithmic vibrations decrements is rather important considering that just they determine maximum value of resonance stresses.



Vibr. mode	Without damper			With damper		
	f,Hz	k	d,%	f,Hz	k	d,%
1	1700-1850	18,34, 36	2.0-6.6	1700-2000	18,34, 36,44	5.6-12.6
2	3400-3800	34,36,51,68	1.1-2.5	3300-3900	36,51, 54,68	3.0-5.9
3	5400-5500	51,54,68,108	1.3-3.2	5500-5600	51,54, 102	1.5-3.9
f - frequency, k – harmonic number, d - decrement						

Fig. 10. Campbell Diagram and Results of Determination of Logarithmic Decrements of Inserted HPT Blades Vibrations.

Performance of tests and special additional calculations showed that rotor design being investigated had insufficient clearance between path platforms of adjacent blades. Campbell diagram is illustrated in Fig.1. It was one of reasons of low efficiency of dampers for the tensional mode of blades vibrations.

Figures 11 to 14 illustrated some results of investigations of vibration characteristics of compressor blisk [7, 8].

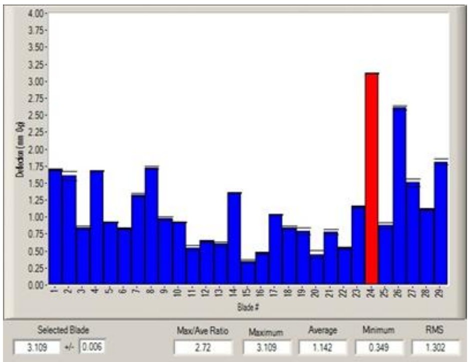


Fig. 11. Maximum Vibrations Amplitude of Compressor Blisk Blades.

This figure presents values of maximum vibrations amplitude for every blade of blisk. Black level mark maximum and minimum values of amplitude for every blade at resonance speed, posts give average blade amplitude value. Red color marks a selected blade (blade having maximum vibrations amplitude).

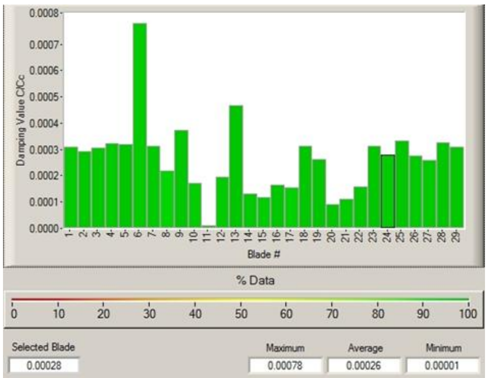


Fig. 12. Damping Coefficient for Compressor Blisk Blades.

Figure 12 gives values of damping coefficient for every blade. Damping analysis is performed through application of non linear least squares method for single degree of freedom model to amplitude data.

Figure 13 gives amplitude value for every blade depending on speed of rotation.

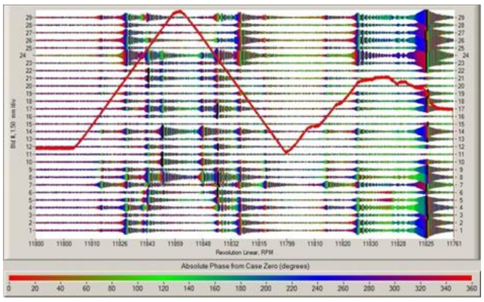


Fig. 13. Dependence of Blisk Blades Vibration Amplitudes on Speed of Rotation

Red plot indicates change in speed. Change in colour of vibrations amplitude curve indicates change of absolute phase relative to upper dead point and may be used for as an indicator of exciter orientation relative to casing.

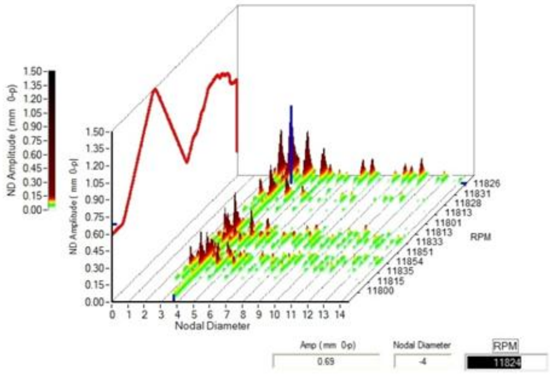


Fig. 14. Modal Diameter Waterfall Plot for Blisk Blades Vibrations.

Figure 14 illustrates nodal diameter waterfall plot that represents composition of vibration response on nodal diameters. Nodal diameters from 0 up to number of blades divided by 2 are plotted on X axis, intervals of analysis scope are plotted on Y, axis deflection magnitudes for amplitudes on nodal diameters are plotted on Z axis. Red plot indicates change in speed.

At the present time, compressor blisk blades are being tested for high-cycle fatigue. Vibrational stress level of blades whose vibrations are excited through oil jets is sufficient to get HCF fracture, vibrational stress stability allows HCF tests to be carried out to plot Goodmen diagram.

4 Conclusion

Dynamic spin rigs can be effectively used to provide turbomachine vibration strength.

5 Acknowledgement

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