Noncontact Crack Detection on Compressor Rotor Blades to Prevent Further Damage after HCF-Failure

M. Zielinski and G. Ziller
Engine Control & Testing Division
MTU Aero Engines GmbH
Dachauer Strasse 665
80995 Munich
Germany

michael.zielinski@muc.mtu.de
gerhard.ziller@muc.mtu.de

ABSTRACT

This paper reports on the status of MTU's BSSM system concerning crack detection on compressor rotor blades. BSSM (Berührungslose Schaufelschwingungsmessung, i.e. Noncontact Blade Vibration Measurement), being under continuous development at MTU and used on all major engine development programs there, primarily serves to measure vibrations. The test data acquired, however, can also be used for crack detection. This paper describes the working principles of several BSSM-based crack detection techniques and illustrates their usefulness by way of concrete experimental applications. Reported also are the limitations and problems still embarrassing crack detection with BSSM.

ABBREVIATIONS

Blisk	Bladed Disk
FW	Full Width
HCF	High Cycle Fatigue
RMS	Root Mean Square
1F	First Flexural Vibration Mode
1T	First Torsional Vibration Mode
2F	Second Flexural Vibration Mode

INTRODUCTION

This paper deals with the noncontacting identification of vibrations and cracks on blisk rotor blades of axial-flow compressors used in turbine engines. In service, these rotor blades are exposed to aerodynamic alternating pressure forces resulting from, for instance, the inlet geometry ahead of the compressor, varying blade tip clearances caused by ovalized casings, or by the stator cascades. These excitation sources are linked with the compressor casing and therefore cause alternating forces with a multiple of rotor speed and in their wake, resonant vibrations when the excitation frequencies intersect blade vibration modes. Given certain conditions, however, also self-excited flutter vibrations may occur that are not directly coupled to rotor speed. This equally applies to vibrations triggered during compressor surge.

Severe vibratory stresses may induce blade cracks and ultimately the loss of blades, which in many cases leads to the total failure of the entire engine (**Figure 1**). To avert such events and the costs and hazards accompanying them, every effort is made in the development of turbine engines to measure rotor blade vibrations at all operating conditions and promptly detect potential cracks. Critical rotor components are subjected in specific high cycle fatigue (HCF) tests to vibratory, centrifugal and temperature stresses that under controlled conditions are boosted until the components under test fail. This is to reliably corroborate theoretical predictions of safe component loads. In production engines, additionally, critical components are checked at regular intervals by visual, ultrasonic or other inspection to detect potential cracks.

Figure 1: Severe damage to an axial-flow compressor following the loss of a blade

To measure vibrations and detect cracks in development engines, and perhaps in future also in production engines, some manufacturers are working on a measurement technique allowing them to measure and analyze the times the blades take to travel between case-mounted probes (**Figure 2**). Primarily meant to measure vibrations, the method bases on the phenomenon that the blade passing times vary in the presence of blade vibrations and do so as a function of vibration frequency and amplitude. Yet the test data still carry information above and beyond those purposes that, for instance, can be used to detect cracks. To what extent they do, will be exemplified below for a number of typical applications. Overall, the method provides significant advantages: it is noncontacting, covers all blades and readily integrates into turbine engines, the instrumentation consisting merely of a few casing-mounted probes.

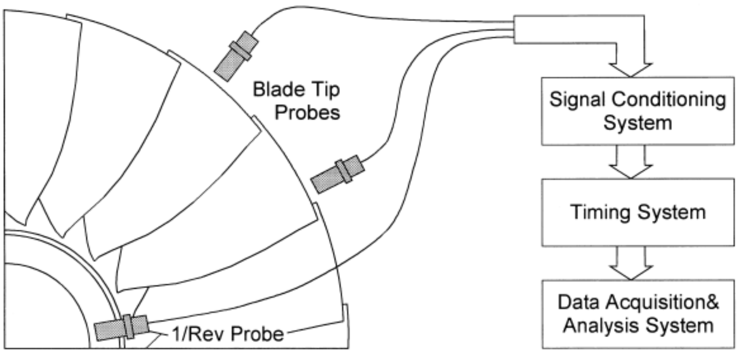

Figure 2: BSSM principle: The blade passing times at the probes vary during vibrations as a function of vibration frequency and amplitude. The signals are correlated with the blades using the once-per-revolution pulse

At MTU, where it is known as BSSM (Berührungslose Schaufelschwingungsmessung, i.e. Noncontact Blade Vibration Measurement), the technique is continuously being improved and routinely used in rig and engine testing. Initially, development relied on optical probes, but in recent years, use has been made almost exclusively of capacitive probes, for the reason that their signals can simultaneously be used also for blade tip clearance measurements. Signal digitalization in conjunction with intelligent trigger algorithms allow highly accurately timed triggering to be achieved despite the wide signal shape. The latest systems, therefore, just consist of capacitive probes, suitable probe driver modules and a personal computer to process and analyze the signals (**Figure 3**).

Figure 3: Latest BSSM system using capacitive probes and software triggering

[1] provides a detailed description of the current state of BSSM development at MTU. It also gives references of the literature conveying an impression of the development pursued by other manufacturers. Described herein now are BSSM-based methods and applications for crack detection on rotor blades.

FLUTTER ANALYSIS WITH CRACK PREDICTION

The first instance illustrates flutter investigations on a single-stage compressor rig (**Figure 4**) where high measured vibration levels indicated the presence of a blade crack. The measurements primarily aimed to safeguard and operationally release the rig for the planned comprehensive aerodynamic investigations without the need for continuous, costly vibration monitoring.

Figure 4: Single-stage compressor rig with blisk rotor

BSSM analyzes flutter using so-called all-blade spectra, where initially the data from all blades are viewed collectively and individual blade results are calculated in a second step. For each probe, the variations in the passing times of successively arriving blades are computed from the test data versus the undisturbed times and Fourier transforms are performed. In these all-blade spectra, blade vibrations appear in the shape of system vibrations characterized by an amplitude and a node diameter and are shifted versus the actual vibration frequency, depending on node diameter and speed (**Figure 5**).

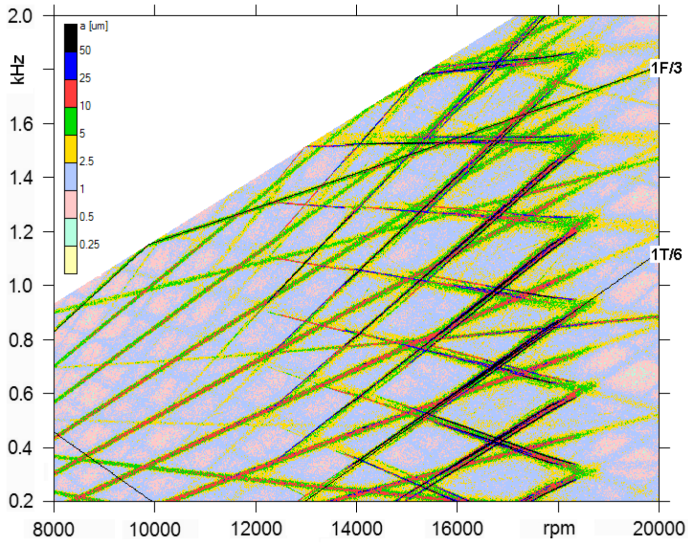

Figure 5: Campbell diagram with all-blade spectra showing flutter vibrations in the 1T mode. The lines of the system modes are shifted versus the real vibration frequency depending on node diameter and speed, and by aliasing effects

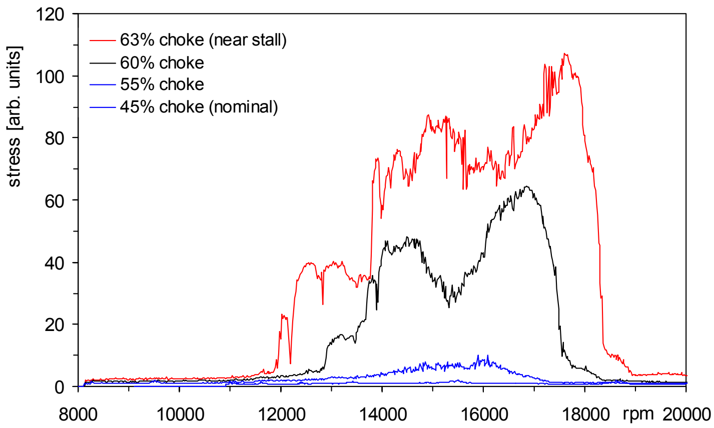

Figure 6: Stress-speed profile for the 1T flutter event for various compressor choke settings. Shown at each speed point is the respective stress peak across all blades

The analysis then proceeds to combine the all-blade spectra of the individual probes. Phasewise, the inputs of the various vibrations differ from one probe to the next as the vibrations continue while the rotor revolves from one probe to the next. These considerations lead to a system of equations permitting the amplitudes and phases of the contributing system vibrations to be computed at each frequency point from the amplitudes of the all-blade spectra. These complex amplitudes are computationally superimposed to yield the individual blade amplitudes. An in-depth description of the procedure will be found in [1].

To be able to evaluate the blade loads resulting from the vibrations, the relationship between the circumferential deflections at the blade tip and the maximum vibratory stress on the airfoil is needed. For the purpose, theoretical modal shape analysis yields, for each vibration mode, the deflection distribution on the airfoil for a given maximum stress value, from which the conversion factors MPa/mm are computed at the axial measurement positions.

The flutter investigations were made with BSSM using five casing-mounted probes, at the 12°, 35°, 133°, 211°, and 240° positions. During startup, the compressor characteristics were completely traversed and the test signals mapped in the process. For a first indication, all-blade spectra were computed online and plotted. It was seen already then that in a certain speed range, high vibration amplitudes occurred in the first torsional (1T) mode of the blades as the compressor was increasingly choked, approaching the stall line (**Figure 5**). When an exact offline analysis was made, albeit somewhat later, it revealed single blade amplitudes that peaked a little above the theoretical fatigue limit anticipated for that case in the Goodman diagram (**Figure 6**).

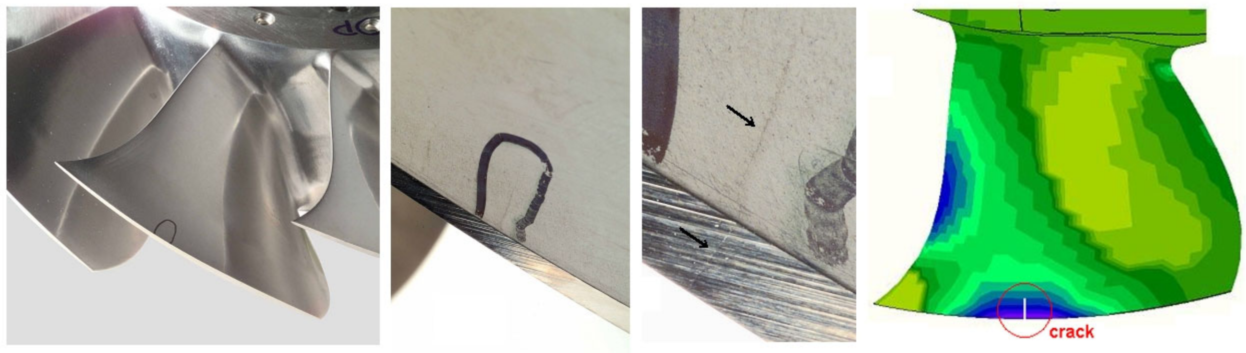

Figure 7: Cracked blade (left) and theoretical 1T stress distribution on the blade (right)

The critical region of the compressor map was then barred. However, by that time the compressor had been operated there for some time already. Estimation suspected that a notable number of cycles had meanwhile been accumulated also at maximum vibratory loads. To be on the safe side, the rotor was therefore crack inspected, when in one blade a crack was detected in exactly the region of the theoretical stress peak of the first torsional mode (**Figure 7**). The position of the crack at the blade tip did not immediately threaten blade separation. Accordingly, the rig was on the one hand released for operation without continuous vibration monitoring, while on the other the barred region was extended a little and the crack put under regular optical inspection.

While this is not a genuine HCF test, this example nevertheless shows how well BSSM, in conjunction with theoretical modal shape analysis, can provide evidence of the vibration load imposed in service on all blades of a rotor stage.

RESONANCE ANALYSIS AND ONLINE CRACK DETECTION

In a second application, BSSM is applied in a genuine HCF test. The aim was to make a blisk rotor resonate, under operating conditions, in a certain vibration mode, using compressor intake gauzes, and boosting the excitation gradually to blade crack. At each load stage, a certain number of vibration cycles were to be run. The experiment was to corroborate theoretical blade stress and life predictions.

The BSSM system's task during the tests was to measure the vibration amplitudes of all blades online to document the integral load on each individual blade and moreover immediately identify blade fatigue so that the rig might be halted before major consequential damage occurred, such as blade rupture.

BSSM crack monitoring was here predicated on the assumption that when a crack occurred, the vibration amplitude of the affected blade was bound to diminish, considering that the crack would lower the natural frequency of the vibration mode and the blade should therefore exit the resonance point.

The plan was to perform the load test with the second flexural (2F) mode of the blades, exciting it through the fourth engine order using gauzes in the compressor intake. This suggested the use of solely three probes spaced 120° apart over the rotor circumference. That enabled the amplitudes of the blades to be computed directly from the blade arrival times at the probes also at constant speed. The procedure is described in [2]. When this method is used, casing distortions or radial rotor displacements relative to the casing may cause the effective probe positions and hence the measured vibration amplitudes to change. But since the vibration amplitudes were expected to be rather large, that presented little if any problem.

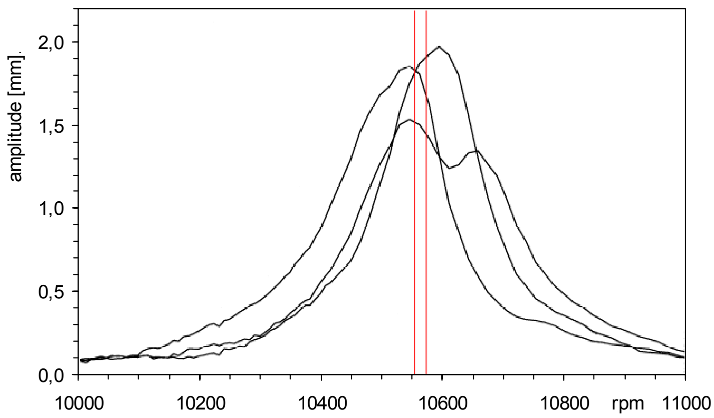

Figure 8: Typical resonance profiles of some blades, vibrating in the 2F mode excited through the fourth engine order. The curves show the manufacture-related scatter of resonant frequencies and the interaction between blades. Plotted also are the two speeds at which the HCF test was conducted

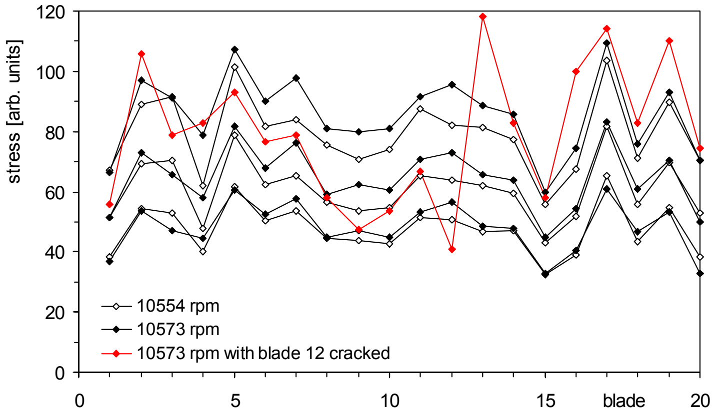

Figure 9: Blade amplitude distribution at the two stationary speed points at different excitation levels before and after crack initiation in a blade

Early into the experiment, the resonance was completely traversed once and the measured resonance curves were used to select two suitable stationary speed points for the HCF test (**Figure 8**). The test then gave entire satisfaction. After various excitation stages with the respective number of vibration cycles had been traversed and a high excitation level reached, the amplitude distribution of the blades, so far stable, abruptly changed significantly (**Figure 9**). The rig was immediately halted and the rotor subjected to thorough crack inspection. In the process, a fatigue crack was revealed in one blade at precisely the theoretically anticipated point of the stress peak (**Figure 10**).

In accordance with the Goodman diagram generated for this blisk, with due consideration paid the material data, material treatment, surface finish, etc., the crack occurred at the time expected, i.e. after a corresponding amount of load and number of cycles. As previously with the first example, theory and measurement matched closely. It should here be emphasized that the change involved not only the amplitude of the cracked blade but the whole amplitude distribution (**Figure 9**). This is explained by the strong interaction between the blades. The blisk must be viewed as a collective; if one of its blades experiences change, that change will affect the entire system.

The experiment shows that given the proper environment, HCF tests can be conducted with BSSM under real conditions, i.e. centrifugal load, until crack initiation occurs, and can be terminated in a controlled manner.

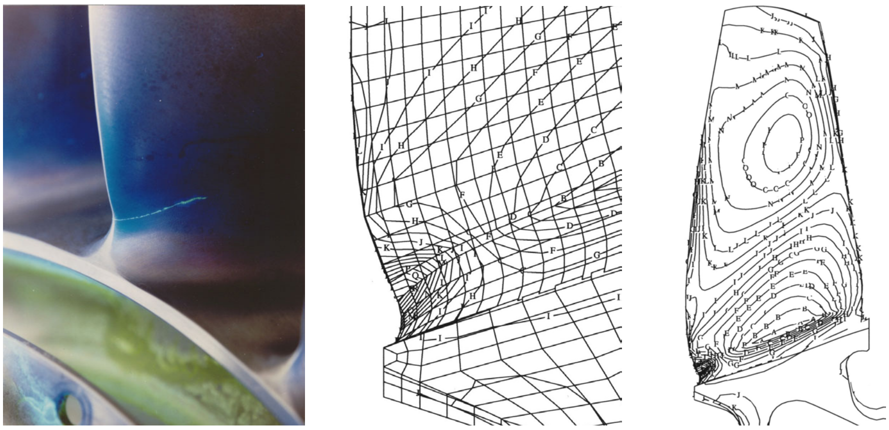

Figure 10: Cracked blade (left) and theoretical 2F stress distribution on the blade (right)

FURTHER CRACK DETECTION METHODS

Described immediately below are two further crack detection methods that are integrated in the BSSM system and successfully used at MTU on compressor rigs and development engines. The former of these bases, like the one just described above, on the phenomenon that resonance positions will shift in the presence of a crack. This method in principle works satisfactorily with only two probes used to record resonance curves. The characteristic shape of these curves (**Figure 11**) results because the time the blades travel between the two probes during a resonance pass will change through the change in vibration amplitude and the phasewise displacement between the exciting force and vibration movement systematically with the speed [1]. The curves very accurately reflect the position of a resonance for each blade.

At the beginning of a test campaign, a suitable resonance is traversed with the undamaged rotor and the position is accurately recorded for the individual blades. Subsequently, further runs are regularly recorded at the same conditions and the deviation of the resonance positions relative to those from the reference run is determined (**Figure 11**). To balance temperature-related displacements, the distributions are first shifted to the same mean value. Two different indicators suitable for identifying cracks are used: the scatter about the mean value (RMS) and the overall width (FW) of the differential distribution across all blades. So long as no cracks occur and the resonance positions of the various blades remain stable relative to each other, both indicators yield values around zero. Changes in the resonance position of one or more blades, however, produce values other than zero. If use is made of several probes, more resonance curves can be formed, adding to the accuracy and reliability of the method.

The method will work also with small resonance amplitudes, considering that potential offsets caused by casing distortions or radial displacements of the rotor position make little difference in the identification of resonance positions. For safe crack detection using this method, a frequency shift of at least 0.5% should result (cf. **Figure 11**). A first theoretical investigation into the subject showed that at least in the fundamental modes and with cracks in the lower airfoil region, frequency shifts on the order of several percentage points may readily occur. A disadvantage is, however, that the resonance must be traversed every time, so that the method will not work at constant speed. That, of course, is a handicap in HCF tests as described above.

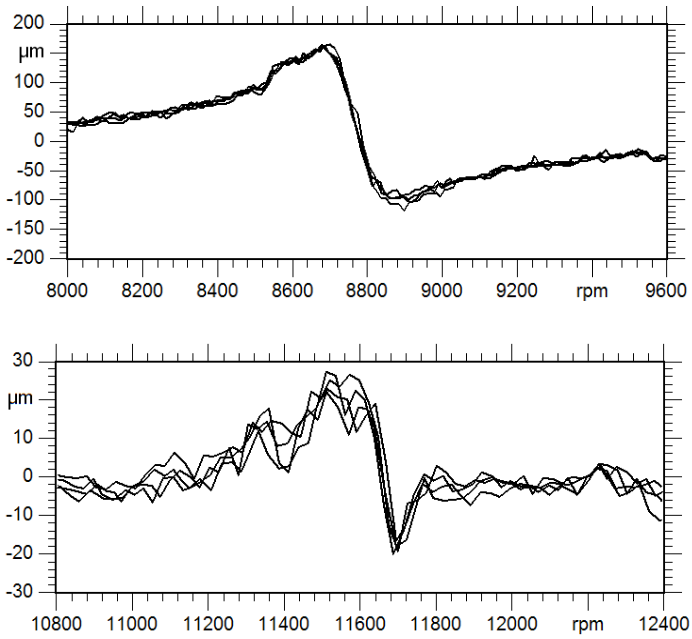

Figure 11: Crack detection by resonance postion analysis: Resonance curves of a single blade from four runs, the figures revealing the high reproducibility of the curves with undamaged blades, here within ± 0.11% (top) and ± 0.17% (bottom). Cracks will lead to shifts in resonance positions

For crack detection, the latter method exploits the phenomenon that blades having suffered an incipient crack will under aero-loads and centrifugal force deform differently than in the undamaged condition. To measure these changes, the time data of the blades travelling past the probes are used to compute the relative positions of the blade tips on the extended rotor section. This is again done against a reference run with undamaged blades. As the reference data is being recorded, the speed may optionally remain constant or else vary within a certain range. Subsequently, further runs are recorded at the same conditions and the blade positions compared with those of the referenc run (**Figure 12**). Blade cracks manifest themselves through deviations from the reference positions. As in the resonance position analysis, this is indicated with the aid of the two indicators RMS and FW of the differential distributions. The more probes are used, the higher the resolution of the method will be; in principle, however, a single probe will be sufficient.

A disadvantage embarrassing the method is that initially, no differentiation can be made between cracking and bending, as by foreign object impact or other forces, e.g. during heavy surge events (**Figure 12**). But so long as no deformation is indicated, a critical crack should not be present, either. Immediately the blade position analysis comes up with an indication, the more elaborate resonance position analysis can be made, which will respond hardly if at all to minor blade bending. This is why at MTU, both methods are applied in parallel.

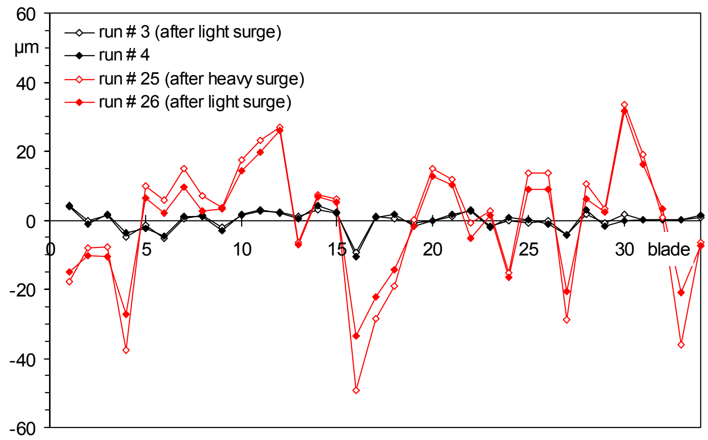

Figure 12: Crack detection by blade position analysis: Blade positions are plotted against a reference run with undamaged blades. Cracks (or bending as in this case) will lead to shifts in blade positions

OUTLOOK

The preceding chapters described the current state of development of the BSSM crack detection approach at MTU. It has been shown that the blades could readily be crack inspected during test operations. Given suitable constraints, also HCF tests can be performed under centrifugal load and with the vibratory load carried to blade crack initiation, under controlled conditions and without consequential damage. A few casing-mounted probes is all that is required by way of instrumentation. That keeps the financial outlay for this measuring technique affordable. As an example for publications of other manufactorers, aiming at crack detection with BSSM, i.e. tip-timing techniques take [3] through [5].

All the progress aside, further development work is still needed. At MTU, therefore, a technology program is underway that aims to further enhance crack detection accuracy and simplify the handling of the software modules. Problems are still lurking in the detection of cracks caused by higher vibration modes, since these exhibit only small deflections, even at elevated stresses, that conceivable are impossible to capture. In such cases, bending in the centrifugal force field will likely be inconsequential, so that crack detection may be impossible using the methods here described. This emphasizes the need for theoretical investigations to shed light on the question of which modes may cause cracks in which places and what impact those cracks will in turn have on the natural frequencies of all modes, and so for various lengths and depths of crack.

Crack detection, moreover, appears to be difficult also on rotors featuring individually mounted blades, where data is needed about how strongly the resonance positions or blade positions will scatter as a result of the seating of the blades in the disk, the fit slightly varying from one acceleration to the next. This, again, calls for theoretical analyses.

For some time now, investigations have also been underway to study options for using the methods here described as monitoring systems for automated crack detection on production engines. That will still require substantial software and hardware-related effort: the hardware should preferably be affordable, light-weight, readily integrable and reliable. The software, too, should allow automated and perfectly reliable operation of the system to be achieved. The bottom line, of course, is whether the airlines see a demand for such systems at all.

REFERENCES

[1] Zielinski, M. and Ziller, G. (2005), "Noncontact Blade Vibration Mesurement System for Aero Engine Application", 17th International Symposium on Airbreathing Engines, September 4-9, 2005, Munich, Germany, Paper No. ISABE-2005-1220

[2] Zielinski, M. and Ziller, G. (2000), "Noncontact Vibration Measurements on Compressor Rotor Blades", Measurement Science Technology, Volume 11, Number 7, July 2000, pp 847-856

[3] Von Flotow, A., Mercadal, M. (2002), "Turbine Rotor Health Management with Blade Tip Sensors: From Laboratory Tool to Fielded System", Proceedings of the International Instrumentation Symposium, Vol. 48, 2002, pp 277-286

[4] Mercadal, M., Von Flotow, A., Tappert, P. (2001), "Damage Identification by NSMS Blade Resonance Tracking in Mistuned Rotors", IEEE Aerospace Conference Proceedings, Vol. 7, 2001, pp 3263-3277

[5] Von Flotow, A., Mercadal, M., Tappert, P. (2000), "Health Monitoring and Prognostics of Blades and Disks with Blade Tip Sensors", IEEE Aerospace Conference Proceedings, Vol. 6, 2000, pp 433-440