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An Experimental Investigation of Dynamic Response of Bladed Disk

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Abstract

This work focuses on controlling of the dynamic behavior of turbomachines which requires an experimental validation phase to ensure their safety. The application of an innovative experimental techniques to measure the modal properties of a rotating bladed integral disk (blisk) is described in this paper. The aim of this analysis is to simulate a correct harmonic component of engine order (EO) that excites the blisk in an engine. The blisk is tested without rotation as static part and an excitation of the impeller is performed by rotating of a jetting air.

This work focuses on the study of the dynamic behavior of a centrifugal compressor impeller in view to preparing of an operational test in an engine requiring precise characterization of modal frequencies and nodal diameters.

Introduction

Turbomachinery designers have a constant preoccupation: improve performance while reducing masses. One main consequence, often underestimated, is the effect on the blade stress design and dynamic behaviour. Static loads due to the rotation speed, the temperature and the steady aerodynamic forces are now so important that the ability of the material to accept extra dynamic stresses is dramatically reduced. Moreover, the intensity of the sources of excitation in turbomachines has increased: smaller axial gaps between stators and rotors, smaller tip gaps inducing rotor/casing contact risks, etc... as was recalled in [1].

Nowadays, the analysis of failure problems under operating condition show that the majority of problems encountered is connected to High Cycle Fatigue (HCF). These problems are often very sudden because a huge number of stress cycles can be performed in a very short time. The margins for dynamic loads being smaller and smaller, the tolerance to vibration have become a key point.

That is the reason why the measuring of the modal parameters of the bladed disk is very useful for predicting their response to aerodynamic forces, for mistuning identification and FE model updating.

Although the measurement of rotating vibrating structures has been addressed by numerous authors but in spite of a new approach was found. Standard procedure is a challenging task especially with hight rotation speed, the surrounding environment, gyroscopic effect, boundary conditions and the possible presence of sensors where the dynamic response of a rotating impeller is measured inside an evacuated chamber [2].

This work focuses on the study of the dynamic behavior of a centrifugal compressor impeller in view to preparing of an operational test in an engine requiring precise characterization of modal frequencies and nodal diameters.

Experimental set up

General description of a test rig:
A general diagram of the test rig is shown in Fig. 1. The system consists of two main parts. The first part is an excitation frame (position 05) and the second is a frame (position 07) for the test piece. These frames are dynamically separated so that the measurement is not affected. The centrifugal compressor impeller is fixed to the frame by a flange. Excitation of the centrifugal compressor impeller is performed by rotating the jetting air (position 02). Specific harmonic excitation is defined by the number of nozzles on the exciter rotor. The excitation rotor is driven by electric motor (position 08). A rotor speed range is from 0 to 11000 rpm.

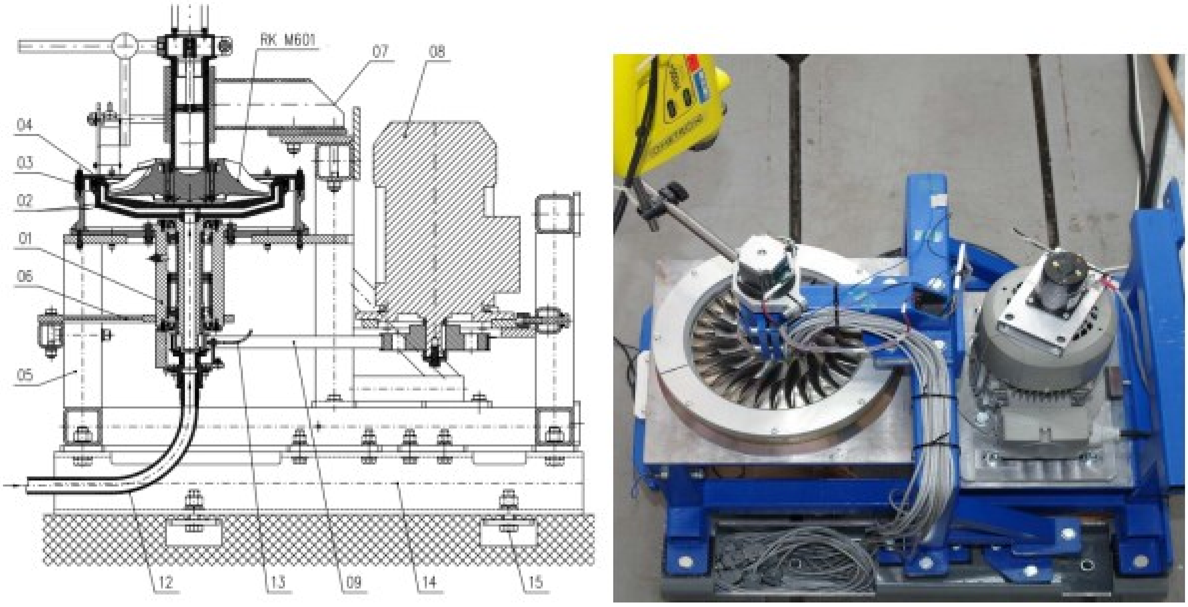

Fig. 1: Test-rig in general view and with labels

Measurements were performed by three means: the first one by accelerometers that were located on disk and on the root of main blades, the second one by strain gauges using an inductive telemetry system; the third by a Laser Doppler Vibrometer (LDV) Ometron VH300+.

The test conditions were monitored using an accelerometer, a thermocouple and a pressure gauge. The tests were performed at ambient temperature.

Definition of harmonic excitation
The sources of excitation in the engine are numerous and most of them are harmonic. Effectively, blades excitations are mainly due to non-uniformity of the upstream pressure field. When rotating, the blades are loaded by a fluctuating pressure field at a frequency connected to the rotation speed. The test rig simulates this aerodynamic dynamic load caused by interaction between rotor blades and diffuser vanes. The sources of excitation can be by the upstream/downstream stator (with *Es* blades) generating excitation frequencies equal to the rotation frequency (ω) times the number of stator blades plus higher harmonic ($Es.\omega$, $2Es.\omega$, $3Es.\omega$, etc…) with decreasing intensity. Turbines can be excited also by the number of injectors (Ei) in the combustion chamber at a frequency equal to $Ei.\ \omega$ and upper harmonics. Moreover, low engine orders are present in the engine, due to static parts non perfect symmetries, such as not perfectly circular casings, not perfectly constant stator vanes or injector spacing, etc…

The rotor of the centrifugal compressor impeller has maximal speed 40 000 rpm. The excitation of the rotor in the test rig has a limited speed, as mentioned above. Therefore, it is important to find another harmonic excitation that excites the same operational shapes in the same frequencies as is the case of the engine.

Bladed disk with Er blades that is excited by Es-th harmonic, can oscillate by modal shape of bladed disk with x nodal diameter (ND). The number of ND is defined by following equation

$$x = |h \cdot Es - k \cdot Er| \tag{1}$$

Where h is harmonic index $\langle 1, 2, 3, \dots \rangle$, k is integer $\langle 0, 1, 2, \dots \rangle$, Es is the number of stationary elements (it is also EO) which define harmonic of speed $h \cdot Es$, Er is the number of rotating blades.

In this specific case, the number of ND is determined the number of rotation blades $Er=30$ and the number of stationary elements (diffusor vanes) $Es=25$. For given number of stationary elements Es is excited five ND ($x_{min}=5$ for harmonic index $h=1$). The maximum emphasis is put on minimal values of x (harmonic index $h=1$). All x values for a particular harmonic index h can be used as harmonic excitations of impeller and all these harmonic excitations will generate operation shape with the same number of ND, which is given by the value x_{min}.

Table 1: Number of ND which is excited by diffusor vanes

k	h	
	1	2
0	25	50
1	5	20
2	35	10

Numerical Analysis of the centrifugal compressor impeller

In order to get an idea about basic vibration characteristics a FE model segment is derived from the ideal design of centrifugal compressor with main blade and splitter blade as is shown in Fig. 2a. A sector of impeller was used to launch FE calculations under cyclic symmetry boundary conditions as is shown in Fig. 2a. The sector of impeller includes three structures (disk, main blade and splitter blade) and represents one fifteenth of the whole.

Additionally, the modes can be distinguished using the blade percentage of the total blisk strain energy as a reliable indicator Table 2. Hence, several blade-modes mode families can be easily identified. These mode families occur as nearly horizontal lines. In contrast to this, disk-modes and coupled modes show more rising frequencies with an increasing number of nodal diameters caused by increasing disk stiffness, as mentioned above can be found in [3,4].

Table 2: Distinguish of blisk modes

	Disk mode	Coupled mode	Blade mode
Blade percentage of total blisk strain energy	<50%	50%÷75%	>75%

Based on this model an eigenvalue analysis with the effect of rotational speed included is carried out, firstly to derive the nodal diameter plot Fig. 2b and with that to get an overview about relevant blade mode families and disk dominated modes as well. A first overview of the possible resonances in operating range and a suitable point for measurement of the response by strain gauge or LDV were obtained based on this analysis. The strain gauges are the most common technique used to measure the dynamic behaviour. The effectiveness of these techniques relies on the pertinent placement of the strain gauges or the laser target point.

The numerical analysis shows a lot of structural vibration modes in engine operating range. Some of them are close to operating modes of the engine. The operating modes of the engine are 60% speed (Idle), 94% speed (Continuous mode) and 100% speed (Starting mode). The engine operating range for the 25 EO is represented by a red line in Fig. 2b. The critical point is between the continuous mode and the starting mode of the engine because there is structural vibration mode which is excited by 25 EO.

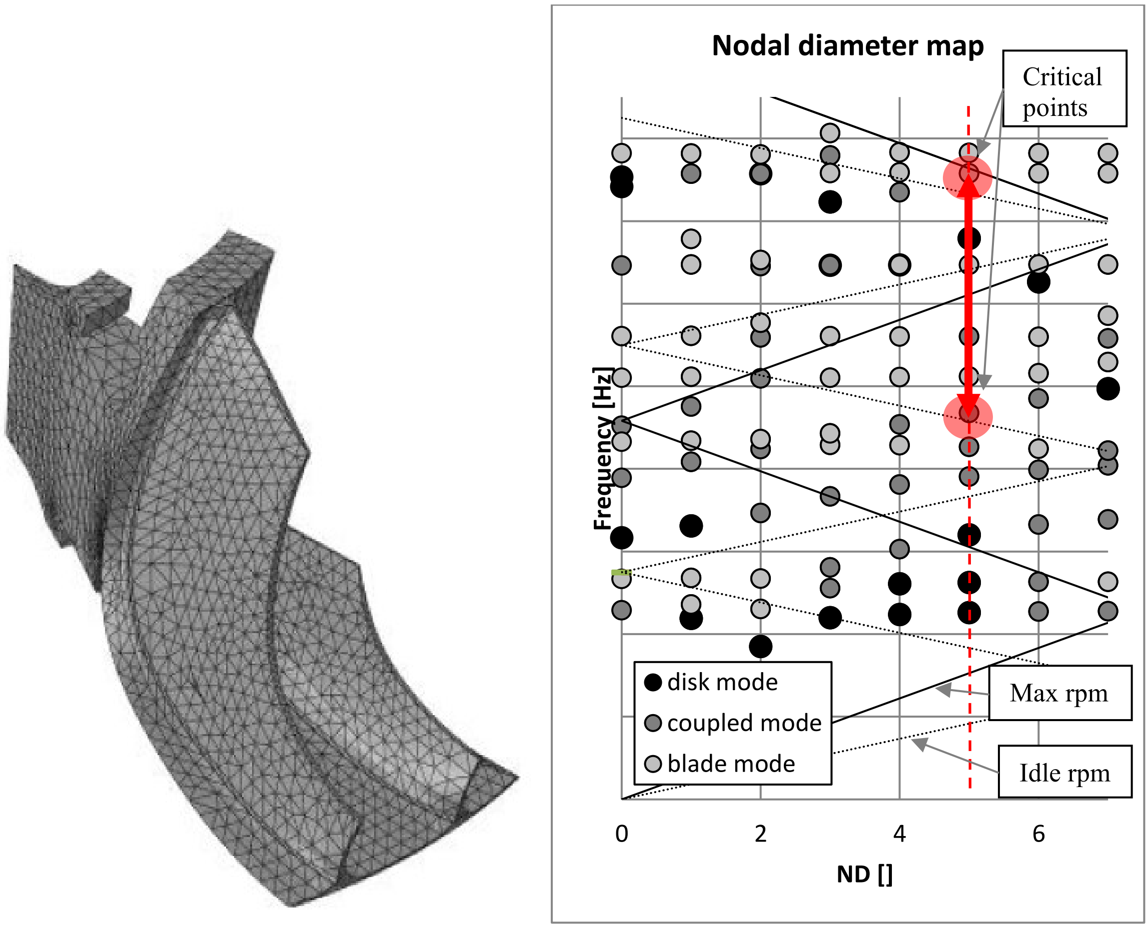

Fig. 2: a) FE model segment of centrifugal compressor and b) SAFE diagram

Experimental method

The impeller was fixed to the frame, as already mentioned. The impeller did not rotate and therefore it was not loaded by static force such as centrifugal force. The measurement was performed using two approaches.

- The first approach determine operational modal shapes. The speed of the excitation of rotor was set up outside any resonance. The impeller was essentially excited by wide-band noise via air turbulence. Responses levels of chosen points were measured by LDV OMETRON VH300+. The chosen points for identification of operational modes are shown in Fig. 3. The Auto and Cross-correlation functions were processed to prepare an input to the modal extraction algorithm in order to analyse natural frequency, damping ratio, and modes shapes.
- The second approach defined the resonance regions in the operating range. The excitation rotor was continually accelerated up to 8600 rpm, which is approximately 108% of maximum operating speed of the compressor rotor. Response levels of chosen points on blades were measured by strain gauges and accelerometers. Harmonic (125th

EO) extraction was performed in off-line mode by digital tracking filter with a bandwidth 0.25h and time step 0.05s.

Excitation of the impeller was performed by rotating of a jetting air. Specific harmonic excitation is defined by the number of nozzles on the exciter rotor. A definition of the number of nozzles was described in the previous chapter. The impeller was excited by 125 nozzles. The mass flow of the jetting air was 172 kg/h.

The responses were measured by LMS SCADAS III frontend with 32 input channels and 2 tacho inputs. The dynamic signal measurement resolution is 16 or 24 bits. Frequency bandwidth was set to 25600 Hz, frequency resolution 2.5 Hz.

Experimental results

Operation modal analysis

Modal identification was performed to verify the excitation method. The identification of modes by operational modal analysis was performed outside any region of resonance (0.81 norm frequency). Fig. 3 shows the selected points on disk, main blade and splitter for the measurement of the mode shapes. The measurement did not include all main blades, splitters and all point on disk. The cause is a frame which holds the impeller. On-line data reduction has been used for modal data measurement. Cross and Auto Spectra were measured for each measured point on the structure, linear averaging of 10 spectra was used. A miniature accelerometer (mass 0.2 g) placed on the disc served as a reference signal measuring in axis of rotation. Operational disk modal shape has expected the number ND, which was excited, as is shown in Fig. 4.

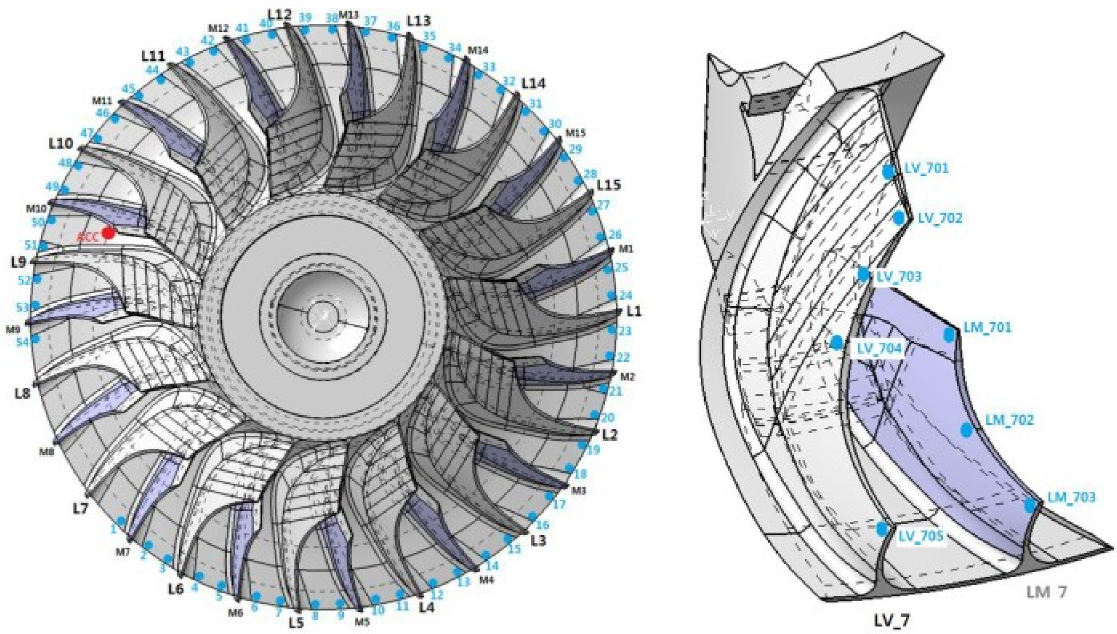

Fig. 3: Points for identification of operational modal shapes

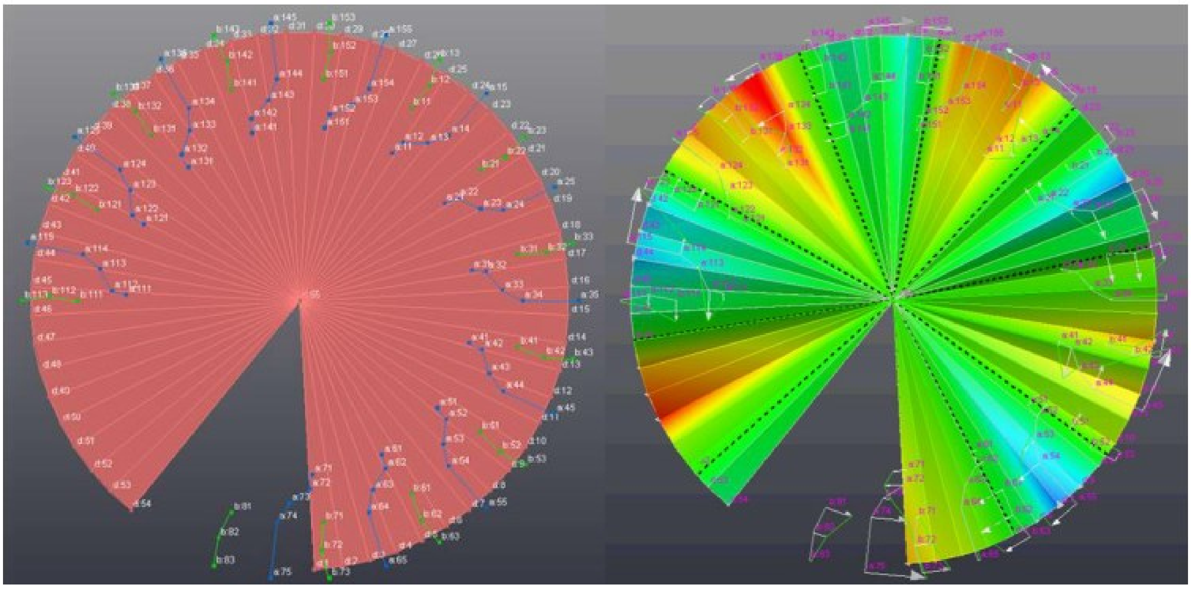

Fig. 4: Model for visualization of the shapes

The first two operational modal shapes have expected the number ND, which was excited, as is shown in Fig. 4. On the other hand the higher operational modal shapes have twice the number of ND. The cause of this phenomenon is the opposite phase of an oscillation of the main blades and the splitters. The mistuning of the blades of the impeller worsens the identification of the mode shapes. Some modes cannot be identified due to the localization of the mode, so-called “blade modes”.

Resonance regions

An example of frequency response measured using strain gauges is shown in Fig. 6 for a particular gauge on all blades at theirs trailing edges as is shown in Fig. 5. This was obtained using a slow acceleration during a run-up test in order to improve the quality of the low signal of strain gauges.

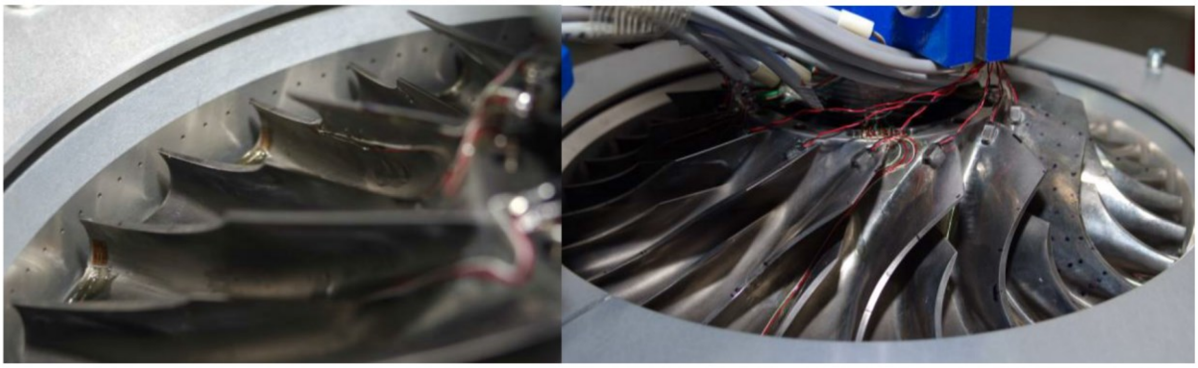

Fig. 5: Placement of the strain gauges on the trailing edge of the impeller blades and accelerometers on root of leading edges

As all the blades were measured simultaneously by their specific strain gauges, the operation deflection shape (in term of strain) with their nodal diameter numbers could be determined automatically as described in the previous chapter. Then, a nodal diameter number (ND) could be attributed to each of the peaks on the response spectrum Fig. 6.

Fig. 7 shows the magnitude of response spectrums from measurement by a mini accelerometers PCB 352A73. The excitation of the centrifugal compressor impeller is the same as was with the strain gauges. The high sensitivity of the accelerometers provides better signal quality. This method was limited by the number of accelerometers. The measurement had to be repeated twice. The accelerometers were placed at the root of the leading edge of the blades as is shown in Fig. 5. A small frequency shift is noticeable. This phenomenon is due to the added mass of accelerometers.

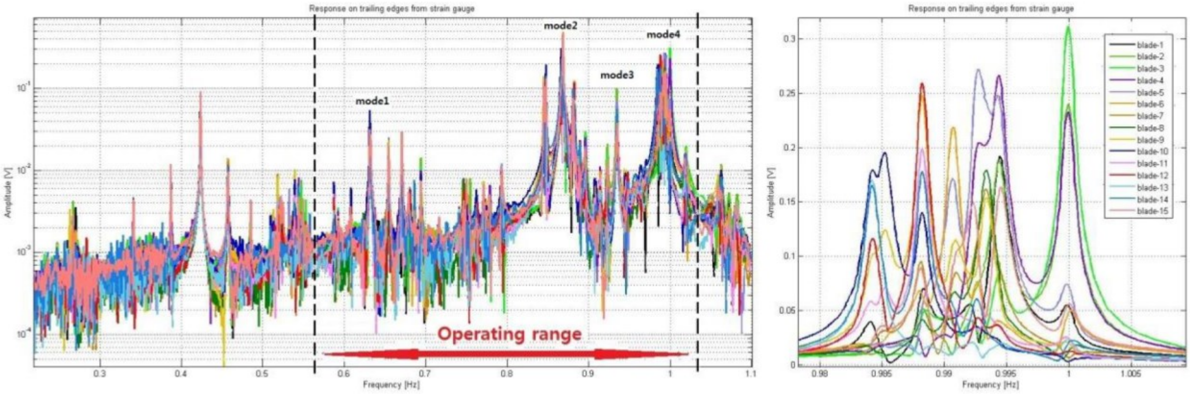

Fig. 6: Spectra measured with strain gauges at trailing edges

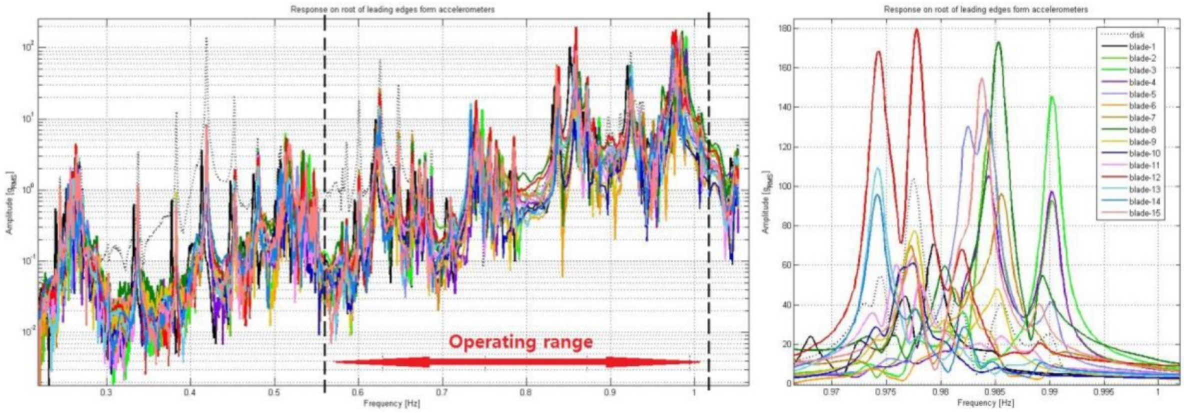

Fig. 7: Spectra measured with accelerometers at the root of leading edges

The mistuning of the system is well obvious in the resonance spectra. Generally, the mistuning of a bladed disk assembly results in several fundamental effects such as splitting of double mode and mode localization [4]. A splitting of double modes is more pronounced in the bladed modes. Disk modes and coupled modes are not susceptible to the mistuning effect as is the case with blade modes [5].

The mistuning is caused by the inaccuracy in the manufacturing. The vibration modes of blade integrated disks are very sensitive to individual blade mistuning. The structure has very low structural damping. The mistuning effect leads to the formation of local zoning of the vibration modes (mode localization) [6]. These local zones have a significantly higher dynamic response and reduce of reserve for the high cyclic fatigue (HCF) of the system.

The instrumentation of strain gauge is an exacting task and in terms of time (tens of hours). It is necessary to perform FEM analysis at first to identify suitable points. On the other hand it is not difficult to install accelerometers. Acceleration sensors have good sensitivity and resolution. The accelerometer is suitable for initial measurement of dynamic behaviours, but the accelerometers cause shifts of natural frequencies though their weight is small (0.2g) and they are placed in the roots of the leading edges of the blades.

Conclusions

This article shows a simple way to experimental validation of the dynamic properties of the bladed integral disk. This process is shown on the centrifugal compressor impeller. The centrifugal compressor impeller was analysed using strain gauges, the accelerometers and the LDV system in order to predict its dynamic behaviours and the influence of certain parameters. A simple test rig was introduced that simulates a specific dynamic load of the impeller (interaction between rotor blades and diffuser vanes) and the definition of the excitation vector. A methodology dealt with searching the correct placement of the strain gauges and advantages and drawbacks of each system were mentioned. The advantages and drawbacks of strain gauges and accelerometers were evaluated, based on the experiment on this specific application.

The measurement of the operational modal shapes was performed by LDV and LMS SCADAS III. The measurements have shown that it is possible to visualize the modal shape oscillation of the particular excited resonance. There was demonstrated that the mistuning of the blades has a useful property. Modal shapes are fixed to the geometry of the impeller and the shapes of oscillation do not change their position. Blades response for a particular resonance is always the same.

The test rig was designed primarily to search out the resonances of the blisks (bladed integral disks). Another function should be a reliable identification of the mode localization, the corresponding blades, and the maximum amplitude magnification. These are not only the prerequisite for dimensioning of aeroengine compressors but also for the optimum selection of blades to be instrumented with strain gauge, in particular, in case of lowdamped blisk rotors. Determination of suitable blades is important because the number of measurement channels is limited. A bad choice of the blades gives a misleading information, as is evident from the amplitude-frequency responses in Fig. 6 and Fig. 7. In general blade vibration modes govern the maximum operational lifetime of compressor rotors caused by the high strain levels of the blades.

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