Chapter 16
Identification of Linear Time-variant Rotor Components in Resonance Crossing Experimental Validation

Luigi Carassale, Roberto Guida, and Davide Iaffaldano

Abstract The mechanical properties of rotor blades used in turbomachinery are identified exploiting the transient amplification that appears when an integer multiple of the rotor speed matches a resonance frequency of the component. To this purpose, the measured dynamic response is suitably processed and fitted with an analytic model representing a linear time-invariant system. In some cases, however, the variation of the blade stiffness due to the changing rotor speed is large enough to make the mentioned approach inaccurate as the system under study is time variant. A mathematical method to deal with this problem formulated using a perturbation method was recently proposed and validated against numerical simulations.

This paper describes an experimental study carried out to validate the identification technique mentioned above. To this purpose, it has been realized a test rig consisting of a vibrating beam whose natural frequency can be continuously variated through a pneumatic cylinder. The excitation is provided through an electromagnet driven by a linear chirp simulating and engine-order excitation. The response is measured by a laser doppler vibrometer. The system is set in such a way to have two closely-spaced resonances on multiple harmonics of the excitation.

Keywords Resonance crossing · Time variant system · Identification

Introduction

The modal identification of rotor blades in turbomachinery is typically carried out by exploiting the amplification that occurs whenever an integer multiple of the rotor speed crosses a resonance of the component during the run-up or cast-down of the engine. This operation implies three main difficulties that are somehow addressed by the techniques that are presently available.

The first problem regards the fact that tests cannot be repeated several times like it is usually done in experimental modal analysis. It is because the engine startup usually requires a long preparation, the instrumentation installed on rotor components have usually a very short life, the time employed for testing have a very high economic impact for operating engines. Besides, even when it is possible to repeat tests, the results are not fully repeatable as the operating conditions of the component, in particular the temperature, can be hardly controlled. For this reason, identification techniques that perform a local averaging of the data are preferred, e.g. the short-time Fourier transform is used instead of the global Fourier transform.

The second issue that makes the modal identification difficult is related to the unavoidable transient response produced by the resonance crossing, combined to the impossibility of measuring the force acting on the component, which has an aerodynamic origin. The transient nature of the response prevents the use of Operational Modal Analysis (OMA) techniques, at least when transient effects are relevant. This issue has been addressed following two alternative strategies. The first idea consists of fitting the transient response of a mode isolated through band-pass filtering with an analytic (approximated) expression of the transient response of a s-dof system in resonance crossing [1]. The second approach is based on the separation of the synchronous response from the non-synchronous transient component through a time-frequency filter [2]. The synchronous response is then fitted by the Frequency response Function (FRF) of a linear system.

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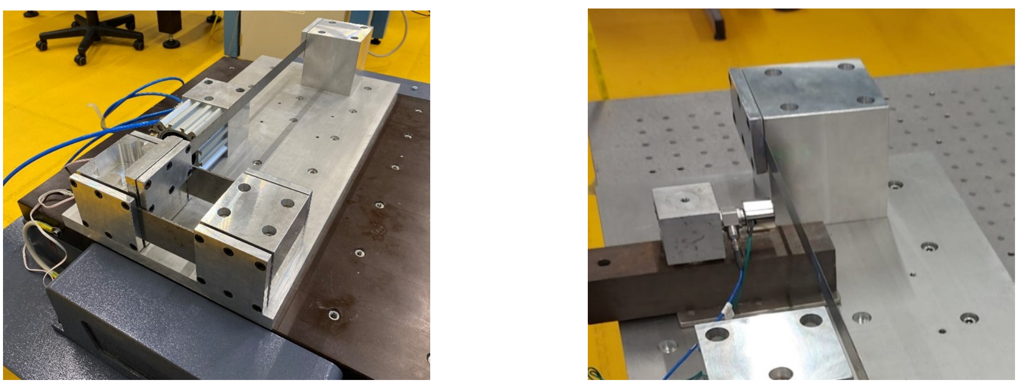
The third difficulty of modal identification appears as the stiffness of rotor blades can be significantly modified by the rotor speed as the centrifugal force has a stiffening effect. For this reason, the natural frequency of the blades changes during the engine run-up (usually increases) and may vary significantly even during the amplification phase produced by the resonance crossing. This effect makes the system under observation as time variant. To address this issue, two alternative strategies have been proposed. The first one consists of modifying the synchronous response extracted by time-frequency filtering exploiting a time-variant phase-shift defined according to an expression dependent on the rate of variation of the natural frequency [3]. The second approach relies on the application of a specifically-designed time-frequency filter realized through a wavelet transform whose mother wavelet is rotated according to the rate of variation of the natural frequency [4].

This paper describes the laboratory validation of these techniques by means of a test rig in which the natural frequency of a beam can be modified in a continuous manner. The laboratory setup enables the comparison of the modal parameters identified in transient and time variant condition with the parameters identified by the traditional experimental modal analysis.

Test Rig

The test rig is composed of a thin still beam whose natural frequencies are modified by changing its axial load. One extremity of the beam is fixed to the frame, while the other end is fixed to an anchor block that only allows the axial component of the displacement. The axial load of the beam is controlled by a pneumatic cylinder (Fig. 1a). The excitation is produced through an electromagnet and measured by a piezoelectric load cell (Fig. 1b). The moving end of the beam is connected to the frame through a pair of elastic elements in such a way to be relatively flexible in axial direction of the beam and very stiff in all the other directions (Fig. 2).



(a) (b)

Fig. 1 Test rig (a), electromagnet used for excitation (b).

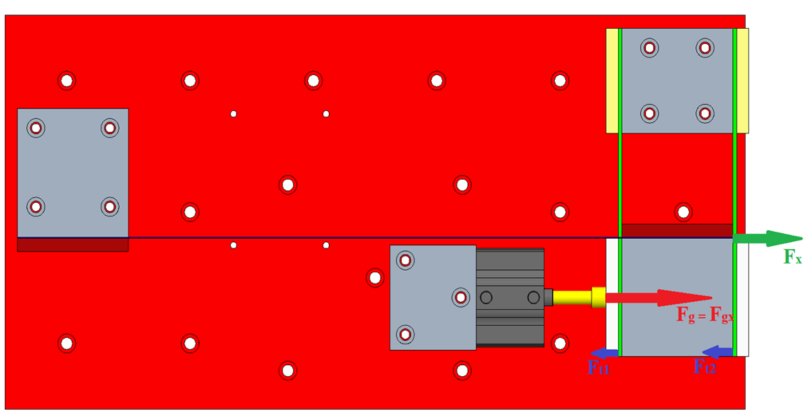


Fig. 2 Working schema of the test rig.

Modal Identification Of The Beam In Stationary Condition

The modal properties of the beam have been identified using the magnet force as input and the velocity measured in a series of points of the beam through a laser doppler vibrometer as output. The test has been repeated for several values of the axial force, produces by the pneumatic cylinder and measured by a strain gauge mounted on the beam. The results discussed herein refer to the second bending mode of the beam (Fig. 3).

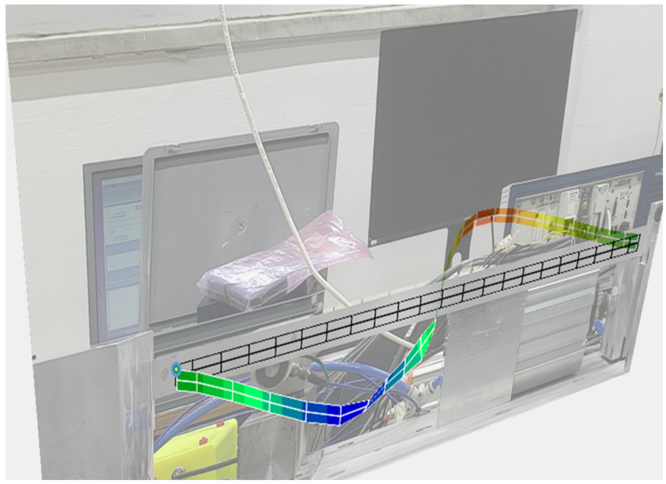
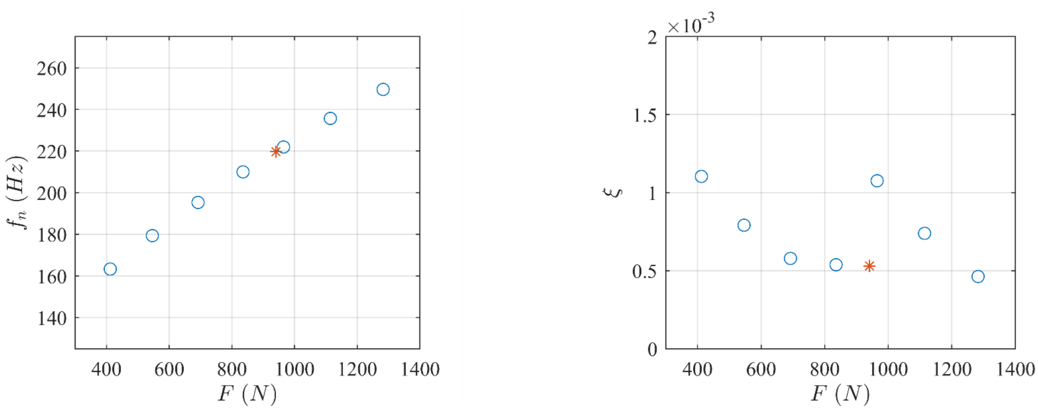


Fig. 3 Mode 2 of the beam used for comparison.

Figure 4 shows the natural frequency and damping ratio of mode 2 as a function of the axial force F (circles). The natural frequency increases linearly as the force increases. The estimated damping initially decreases, but for high force levels have an irregular behavior that still needs to be explained. One hypothesis is related to the difficulty of keeping the pressure in the cylinder constant for the whole experiment. The slight variation of the natural frequency turns into an increment resonance peak bandwidth and, as a consequence, of estimated damping. A refinement of the test rig to ensure a more stable axial force is currently under realization.



(a) (b)

Fig. 4 FRF of the beam in the frequency range of mode 1 for different values of the pressure.

Modal Identification Of The Beam In Transient And Time Variant Condition

For this second test, the beam has been excited through the magnet drive by a linear chirp to simulate the synchronous force acting on rotor blades during the engine cast down. During the reduction of the excitation frequency, the air in the pneumatic cylinder has been progressively released to produce a reduction of the natural frequency. Figure 5 shows a spectrogram of the vibration velocity of the beam measured during the test. It can be noted that the first and the second harmonics of the magnetic force cross mode 1 and mode 2 of the beam almost simultaneously.

The resonance crossing of mode 2 has been analyzed using the method described in [3] leading to the results shown in Fig. 4 (star). It can be observed that the results obtained in transient and time variant condition are very close to what it was obtained by the experimental modal analysis.

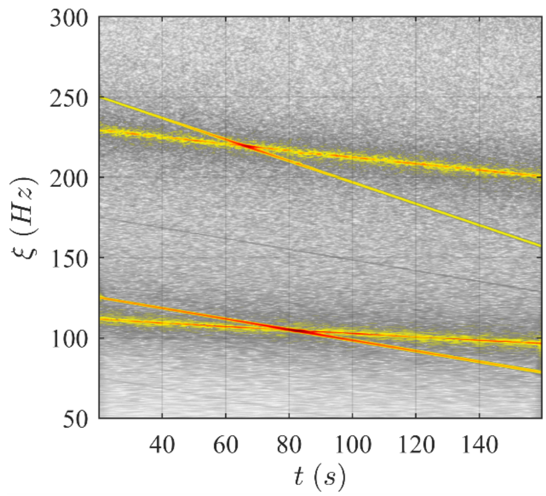


Fig. 5 FRF of the beam in the frequency range of mode 1 for different values of the pressure.

Conclusion

This paper describes a test rig designed to reproduce the resonance crossing of systems whose natural frequency changes progressively with time. The preliminary results pointed out some issues with the stability of the working pressure that affected some results (in particular damping for high pressure levels). However, the results confirmed that it is possible to identify correctly the modal property of a system that is observed during a resonance crossing with its natural frequency changing according to a linear low.

References

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