[Article] Measuring the Performance of Underplatform Dampers for Turbine Blades by Rotating Laser Doppler Vibrometer

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Highlights

Measuring the performance of underplatform dampers for turbine blades by rotating laser Doppler Vibrometer

Mechanical Systems and Signal Processing ∎ (∎∎∎∎) ∎∎∎–∎∎∎

Q1 S. Zucca [a], D. Di Maio [b], D.J. Ewins [b]

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► Experimental analysis of the effect of underplatform dampers on bladed disk dynamics is presented. ► An improved rotating test rig with non-contact excitation and measurement systems is used. ► The rotating measurements are performed by using a Scanning Laser Doppler Vibrometer. ► The non-linear relationship between the force and the vibration amplitude has been measured. ► The effects of the number of engine order excitation have been systematically measured.

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Measuring the performance of underplatform dampers for turbine blades by rotating laser Doppler Vibrometer

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ABSTRACT

Underplatform friction dampers are commonly used to control the vibration level of turbine blades in order to prevent high-cycle fatigue failures. Experimental validation of highly non-linear response predictions obtained from FEM bladed disk models incorporating underplatform dampers models has proved to be very difficult so as the assessment of the performance of a chosen design. In this paper, the effect of wedge-shaped underplatform dampers on the dynamics of a simple bladed disk under rotating conditions is measured and the effect of the excitation level on the UPDs performances is investigated at different number of the engine order excitation nearby resonance frequencies of the 1st blade bending modes of the system. The measurements are performed with an improved configuration of a rotating test rig, designed with a non-contact magnetic excitation and a non-contact rotating SLDV measurement system.

1. Introduction

High vibratory stresses of turbomachinery components can be major cause of high-cycle fatigue (HCF) and so it is very important to maintain vibration amplitudes within the design parameters. A common method for passively controlling the level of vibrations of the blades is to use friction dampers, which are positioned between the blades. Actually, from the literature, the underplatform dampers (UPDs) is a well-established design method for vibration reduction and is thoroughly studied by researchers. UPDs consist of small metal parts, located under the blade platform, which work by the centrifugal force acting on the damper itself during rotation and thus forcing them against blades' platforms. Over the years, a great amount of research work has been produced on UPDs to develop reliable numerical tools capable of predicting their dynamics and effects on the bladed disks responses. In the early works [1–3] the UPDs were modelled as bodies always in contact with the blade platforms and their kinematics deduced from the kinematics of the blades. In later works [4,5], the role played by damper rotation and the possible partial detachment of the UPDs from the blades have been recognized and these features included in the models. Also the bulk elasticity of the UPDs [5–8] is modelled and the relevance of the static/dynamic coupling of the UPDs contact forces has been investigated [9,10]. However, experimental validation of highly non-linear response predictions obtained from FEM bladed disk models incorporating UDPs models has proved to be very difficult so as the assessment of the performance of a chosen design. In the literature, most of the experimental results available on UPDs refer to static rigs [2,3,6,11,12] consisting of a set of blades clamped into a fixture, with a set of interposed UPDs. An external excitation is applied by means of an electromagnetic shaker and the centrifugal

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force on the dampers is simulated by a pulling force applied by using dead weights connected to the UPDs with steel wires. The presented experimental set-ups have the following shortcomings: (i) the effect of the excitation system on the system dynamics, (ii) the effect of the steel wires on the damper kinematics, (iii) the dynamics characterized by standing mode shapes, which are different from the rotating mode shapes typical of axi-symmetric structures like bladed disks under rotating excitation.

An important step has been made in [13], where the damping of thin-walled underplatform dampers is measured experimentally by using a rotating rig, where the test structure included two pairs of turbine blades mounted symmetrically on the disk and the excitation was produced by using piezoelectric devices connected to the blades. In order to test the effect of underplatform dampers on a full 360° bladed disk, a static rig [14] has been designed where the bladed disk is excited by a rotating travelling wave excitation generated by a set of non-contact electromagnets and measured using a Scanning Laser Doppler Vibrometer (SLDV). Non-contact transducers are suitable for this type of measurements because of their capability to measure the vibration remotely and thereby not affecting the dynamics of the bladed disks, which may be sensitive to mistuning. As introduced earlier, the centrifugal force is simulated by pulling the dampers attached to dead weights by steel wires. A significant improvement of the design of the test rigs developed for the experimental characterization of UPDs for bladed disks is presented in [15,16], where a rotating rig has been designed with a non-contact magnetic excitation and a non-contact rotating SLDV measurement system, based on a mechanical tracking method. This paper aims to present an improved configuration of the test rig developed in [15,17]. In fact, the most recent design configuration of the test rig relies on the tracking methods developed in which simplified sensibly the mechanics of the tracking measurement method.

Most of the studies reported on tracking techniques are based on a single acquisition point and so, either a line or an area had to be scanned, addressing a number of predefined positions sequentially. Basically, it was developed on the concept of a conventional step mode similar to the one performed on non rotating targets. The use of tracking techniques was developed to measure vibrations of rotating bladed discs. Three techniques: (i) Point Tracking, (ii) LineScan Tracking and (iii) AreaScan Tracking, respectively, were devised to perform such measurements and all of them have been packaged as a software platform called Caiser Mymesis [17]. The success of tracking measurements using SLDV methods depends on some fundamental contributions: the scanner is embedded into the laser head and the synchronization of the laser spot with the rotating target by means of tachometers, which output the angular position in the form of electrical signals readable by an electronic card.

In this paper the self-indexing mechanical tracking setup was replaced by exploiting the X–Y scanning mirrors installed inside a laser head of a SLDV system, thus improving the capabilities of the measurement technique. The modified test rig has then been used to measure the effect of wedge-shaped underplatform dampers on the dynamics of a simple bladed disk under rotating conditions. The effect of the excitation level on the UPDs performances has been measured and the effect of the number of engine order is systematically investigated nearby some of the resonance frequencies of the 1st blade bending modes of the system.

2. Experimental setup

Rotating test rigs used to measure the vibration of axi-symmetric structures, such as bladed disks, must be designed such that any experiment can be carried out under controlled-known conditions. A key point in the design of these rigs is the use of non-contact excitation and measurement systems not to introduce in the system additional mistuning, which may affect the dynamic properties of the bladed disk, and additional damping.

2.1. Test rig (vacuum chamber, electric engine and measurement system)

The test rig used for this work was designed and manufactured for studying bladed disks under rotating conditions (with and without UPDs). The rig was designed to operate under vacuum conditions, the rotor being enclosed in a chamber, in order to perform measurements without any additional areo-damping effects. The front side of the chamber is closed by a glass window allowing full operational conditions to a Scanning Laser Doppler Vibrometer (SLDV), used to measure the out-of-plane bladed disk vibrations remotely. The rig was equipped with an electric motor whose maximum achievable speed was 6000 rev/min (Fig. 1).

The test rig was originally designed such that the tracking of a rotating blade could be performed using a mechanical device capable of addressing the laser spot on a position of the blade tip. This original test rig design was modified to perform a different type of tracking, achieved using the in-built SLDV scanner electronically controlled by a DAQ cards. In the original mechanical tracking method (Fig. 2), a SLDV was used as a single-point transducer, its laser beam was directed along the rotation axis of the rotor shaft, on whose tip was attached a small 45° mirror (rotating with the shaft), and then redirected in a radial direction to encounter an annular, 45° mirror that turned the measurement beam onto a point near the tip of one of the rotor blades. That arrangement allowed the measurement point to rotate with the bladed disk and to measure vibrations at the tip of the given blade in a rotating frame of reference, in a direction parallel of the rotation axis. This type of tracking method is very robust since it is achieved by the rotating mechanical component of the rig itself, but it did not allow to change the measurement point on the blade. The alignment of the SLDV in this method is very critical since misalignment sources could be identified in the indexing mirror, annulus mirror and laser beam-shaft

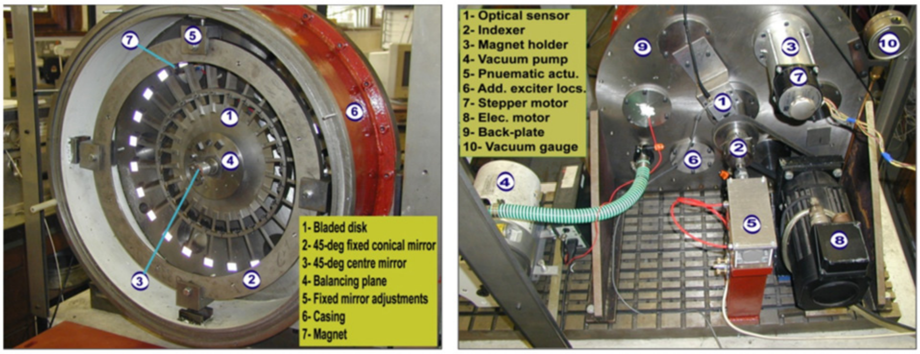

Fig. 1. Front and rear of test rig.

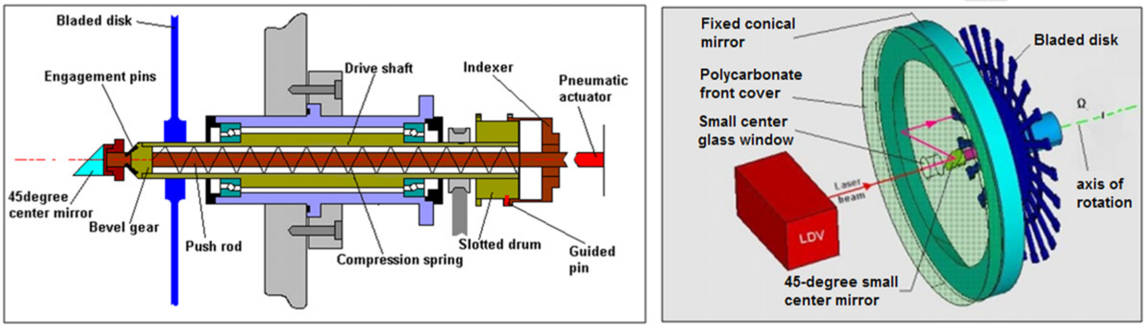

Fig. 2. The index mechanism (left) and the self-tracking system scheme (right).

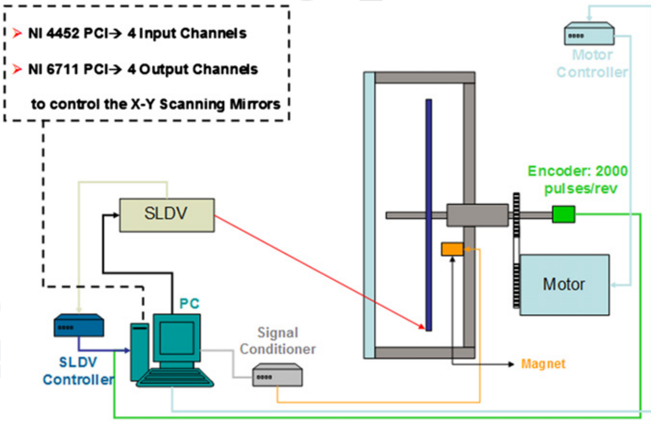

Fig. 3. Test rig configuration used in the present work.

rotational axis. Misalignment is an inherent problem of tracking measurements. The major problem is the pseudo-vibration measured by the laser beam while tracking. This, when the misalignment is severe, can cause the over ranging of the SLDV, In fact, the pseudo-vibration can be synchronous with the EO at which the rotor is spinning and so thereby causing over ranging. As a result, it is very difficult to estimate the actual amplitude of vibration.

The current version of the rotating rig (Fig. 3) uses some of the features already designed but it is mainly based on a new tracking measurement method. It is lighter, in terms of mechanical components, with the SLDV system able to drive the laser spot synchronously with the rotating target.

The laser beam is deflected by two X–Y mirrors which can be controlled either by a DC signals, generating a step movement of the mirrors, or by a continuous waveforms and so generating a continuous movement of the laser beam across the surface. The Point Tracking measurement method, used in the present paper, involves one measurement point at a time. The tracking is achieved by feeding the X–Y scanning mirrors with two waveforms, cosine and sine respectively, so that the laser spot traces out a circle. The amplitudes of those waveforms identify the radial position of the target and the rate at which the waveforms are output must be equal to the rotational speed of the shaft to achieve synchronization.

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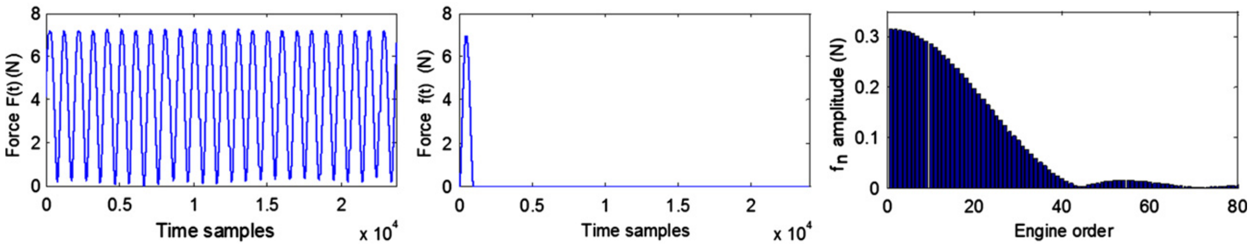

Fig. 4. Magnetic force $F(t)$ acting on the magnet (left), average periodic force $f(t)$ acting on one blade (centre), engine order amplitudes f_n of the force acting on one blade (right).

The two waveforms, in practical cases, have to take into account the possibility of a delay between the output of the sine waves and the actual rotation of the mirrors due to their inertia. Hence the actual true circle path is traced, during the tracking only when the two waveforms take into account both amplitude and phase corrections. Misalignment problems are taken into account and alignment has been performed carefully in order to minimize any undesired pseudo-vibration, as described in details in [18].

The shaft was equipped at one end with an encoder (2000 Pulses/rev) which serves as time base clock for the SLDV X–Y mirror waveform sampling rate. Thanks to improved electronics, the acquisition and output board cards were more compact and easily mountable into a PC desktop digitally controlled by software only. The output card was an NI 6711 PCI with 4 output channels two of them used for driving the SLDV. That card has the option to use an external time base clock to sample the output waveform, represented by the TTL output signal of the encoder.

2.2. Excitation system

Synchronous vibration of rotating turbine blades are excited by periodic travelling forces mostly due to the unsteady aerodynamic phenomena of the vanes of the previous stage. In order to reproduce in vacuum conditions this kind of excitation with a rotating bladed disc made of metallic materials, it may be common to use fixed permanent or DC magnets to excite the passing blades. Using this type of excitation arrangement, the bladed disc experiences a phased pulse excitation whenever a blade passes an exciter magnet and, in fact, the excitation force fluctuates from near-zero when the magnet is between blades, to a maximum when the magnet is centred on a blade tip. Since the magnet only attracts the blades (and does not repel them), all the force values are positive. Either a single or multiple magnets can be installed behind the rotating blades. The higher the number of the magnets, the higher is the force level provided to the system, but if N equally spaced magnets are used, the harmonic content of the excitation acting on each blade would contain only terms corresponding to the multiples of N.

In this work, in order to split the harmonic content of the excitation over the whole spectrum without filtering out any harmonics, a single permanent magnet is used to excite the rotating bladed disk, as shown in Fig. 3, located near the tip of the passing blade, where the system receptance is maximum for the fundamental bending mode shapes object of the investigation. The magnet is mounted on a threaded bar, in order to control the clearance between the magnet and the blade tip and thereby to adjust the level of force.

In order to extract the amplitude of the generic engine order component, the force signal $F(t)$ measured by the force transducer connected to the permanent magnet is recorded for a single period (Fig. 4), showing a number of peaks equal to the number of blades, then the average peak is computed and used to define the time history $f(t)$ of the average periodic travelling force applied to each blade (Fig. 4). Finally, the engine order components of the force $f(t)$ are computed as

$$f_n = \frac{\chi}{N}\sum_{k=0}^{N-1} f(k\,\Delta t)e^{-(2\pi/N)ikn} \quad \text{with} \quad n=1,\ldots,\frac{N}{2} \tag{1}$$

where Δt is the sampling time and with

$$\chi = \begin{cases} 1 & \text{if} \quad n = N/2 \\ 2 & \text{if} \quad 1 \le n \le (N/2-1) \end{cases}$$

3. Test structure

3.1. Description

The test rig described in the previous section has been used to perform experimental measurements of the blades forced responses levels at different numbers of engine order with underplatform dampers installed.

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The test piece (shown in Fig. 5) used in this work is an integral bladed disk (blisk) with 24 blades [15]. In this way, multiple damping sources are avoided since no friction damping phenomena will be produced at the blade/disk joint. The underplatform dampers are wedge shaped and are placed between blade inserts (Fig. 5), glued under the blade platforms. The 24 blades are staggered at 40° with respect to the axis of rotation, in order to couple the in-plane and the out-of-plane displacements of the blade in bending vibration.

3.2. Modal analysis

In order to identify the modal properties of the blisk, modal analysis has been performed using test data measured under stationary conditions by using an impact testing method. Natural frequencies, damping and mode shapes of the first modal family of the blisk, corresponding to the 1st bending mode of the blades, were obtained. The analysis is focused on mode shapes with a number of nodal diameters between 2 and 12. Mode shapes with ND 0 (umbrella mode) and ND 1 have been discarded, because they involve not only the blisk but also the driving shaft and the supporting bearings. In Fig. 6, the identified natural frequencies are shown, and the effect of mistuning in terms of frequency splitting of the double modes (#1 and #2 for 2–11 ND) can be observed.

The mode shapes are plotted in Fig. 7, while the harmonic coefficients of the mode shapes, computed with a Fourier analysis are shown in Fig. 8. It can be observed that the effect of mistuning in terms of modal distortion is almost negligible

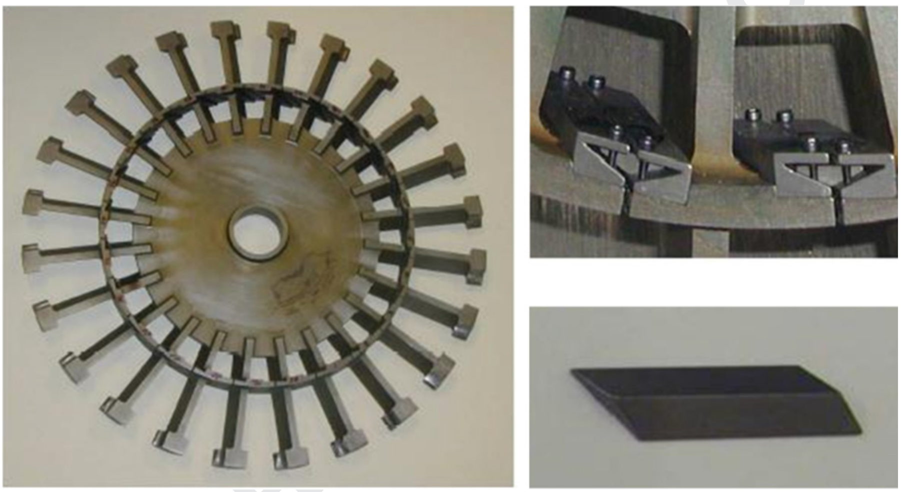


Fig. 5. Test article: the blisk, the blade inserts and the damper.

ND	Frequency (Hz)		
	#1	#2	Split
0	-	-	-
1	-	-	-
2	136.60	136.66	0.06
3	204.55	204.74	0.19
4	247.48	247.52	0.04
5	266.98	267.04	0.06
6	275.92	275.98	0.06
7	280.95	281.10	0.15
8	283.82	283.86	0.04
9	285.36	285.73	0.37
10	286.51	286.54	0.03
11	287.10	287.32	0.22
12	287.93	-	

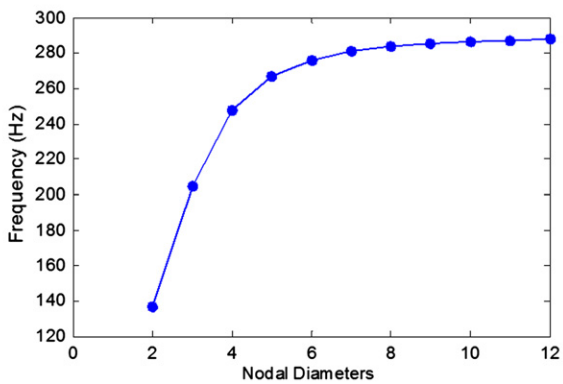


Fig. 6. 1F natural frequencies of the blisk.

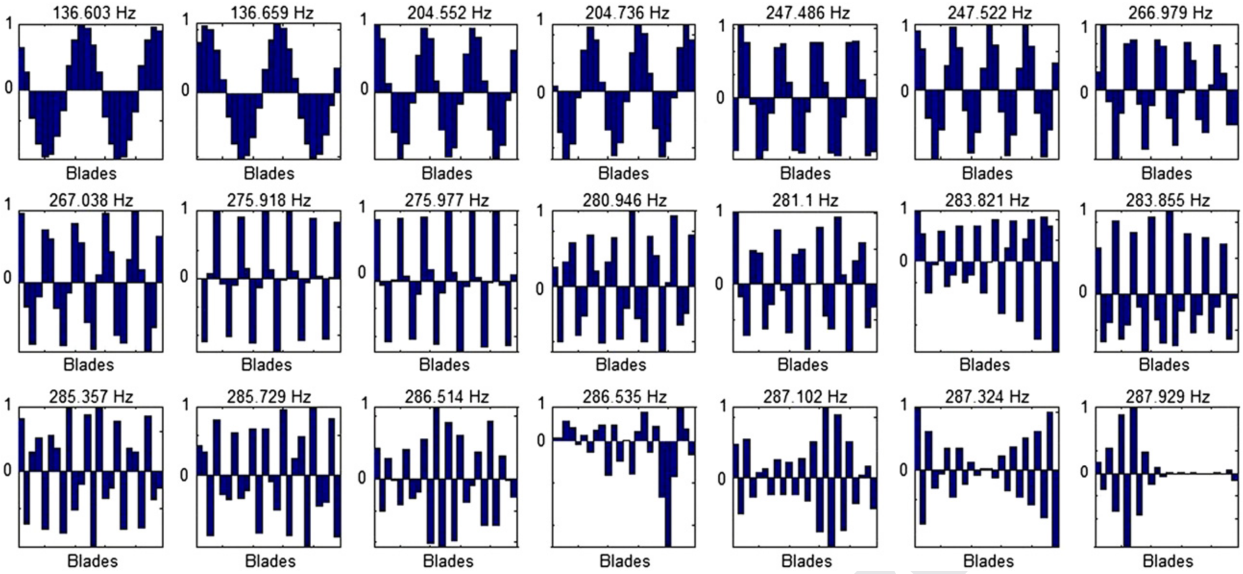


Fig. 7. 1F experimental mode shapes of the blisk.

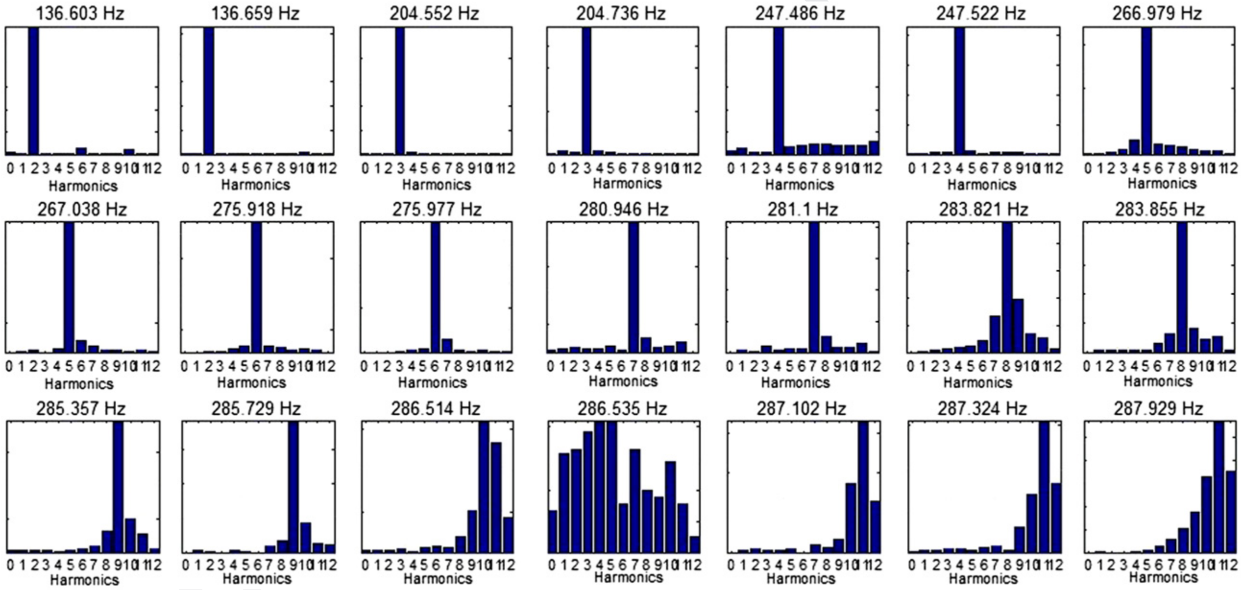


Fig. 8. Fourier analysis of 1F experimental mode shapes of the blisk.

for the first few mode shapes (ND 2–6), while it becomes more and more relevant for the higher modes (ND 7–12), located in a frequency band characterized by a higher modal density.

4. Measurements

4.1. Overall description

The experimental campaign aimed at characterizing the effect of the underplatform dampers on the forced response of the bladed disk, since resonant vibration at steady-state would represent the worst operating conditions for bladed disk, with the possibility of failure from high cycle fatigue. As a consequence, a stepped rotating speed method is selected for the experimental plan. A set of measurement parameters is defined for the experimental measurements:

- A range of rotating speeds of the blisk and speed step resolution are defined; these correspond to one of the resonance crossings on the Campbell diagram of the blisk.
- The phase delays of the X–Y scanning mirrors is determined at the selected speed range, since, if not taken into account for, the correction of the X–Y mirrors signal waveforms can address the laser spot at a different position to the target.
- The acquisition parameters are set (e.g. acquisition time, sampling frequency, etc.).

Two different types of measurements have been performed:

1. Frequency Response Function (FRF) measurement of a blade was carried out in the selected rotational speed range (time length: 3–4 h for all the 24 blades); each blade was measured using the following steps:
   - Steady-state measurements are performed at the blade tip in the given rotating speed range.
   - A data file containing the time signal is saved.
   - The laser beam is rotated of an angle equal to $2\pi/24$ in order to measure the next blade.
2. Response measurement of all the 24 blades at a given rotational speed, referred to as an Operating Deflection Shape (ODS) measurement (time length: 10 min), consisting in the following steps:
   - Steady-state measurement is performed at the blade tip at a fixed rotational speed.
   - A data file containing the time signal is saved.
   - The laser beam is rotated of an angle equal to $2\pi/24$ in order to measure the next blade.

The experiments are performed around the resonance frequencies of the 1st modal family of the blisk, and in particular around the resonances corresponding to mode shapes with 2–4 nodal diameters. These modes have been chosen because they are well isolated resonances, as shown in Fig. 6, and because they are very lightly distorted from inherent mistuning of the blisk as shown in Figs. 7 and 8. The numerical Campbell diagram of the blisk is shown in Fig. 9. This plot has been produced for a rotational speed range between 0 and 4000 rev/min, which is the working range of the SLDV scanner. The crossings of 2–4 ND natural frequencies with the 2–4 engine order (EO) lines are beyond this operating range and so the selected resonances could not be investigated. However, any ND mode shape of a whirling bladed disc can be excited by any engine order EO by the following relationship:

$$EO = n\,N_b \pm ND \tag{2}$$

where n is an integer number, N_b is the number of blades. Hence, 2–4 ND resonances are measured at the rotational speeds crossings the engine order #22, #21 and #20 respectively, which are shown as red points in the Campbell diagram of Fig. 9.

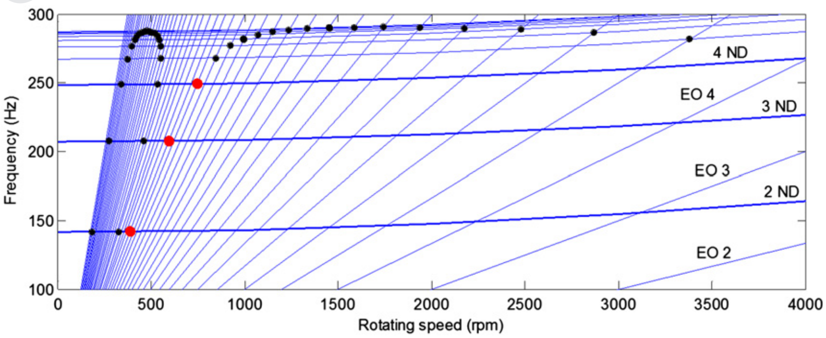
4.2. Linear forced response

A preliminary measurement of the linear response of the bladed disk without underplatform dampers installed has been performed around the 2–4 ND resonances (see Fig. 10) so as to check the results of the modal analysis previously performed, earlier, in static conditions with a hammer test, described in Section 2.2. That was needed so as to define the rotational speed ranges of the blisk for subsequent non-linear measurements.

4.3. Non-linear forced response

4.3.1. Repeatability checks

After the linear forced response measurement, the blisk was removed from the test rig and the underplatform dampers inserted in the blade cavities. In literature is reported that, in experimental measurements performed using underplatform dampers [12,15], the friction effects are subjected to a large variability in time as a consequence of changes occurring on



Q4 **Fig. 9.** Numerical Campbell diagram of the blisk: critical crossings (black and red dots) and investigated resonances (red dots). (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

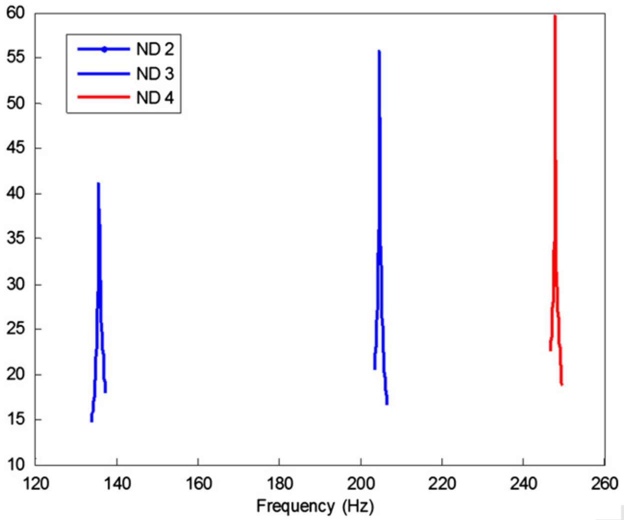
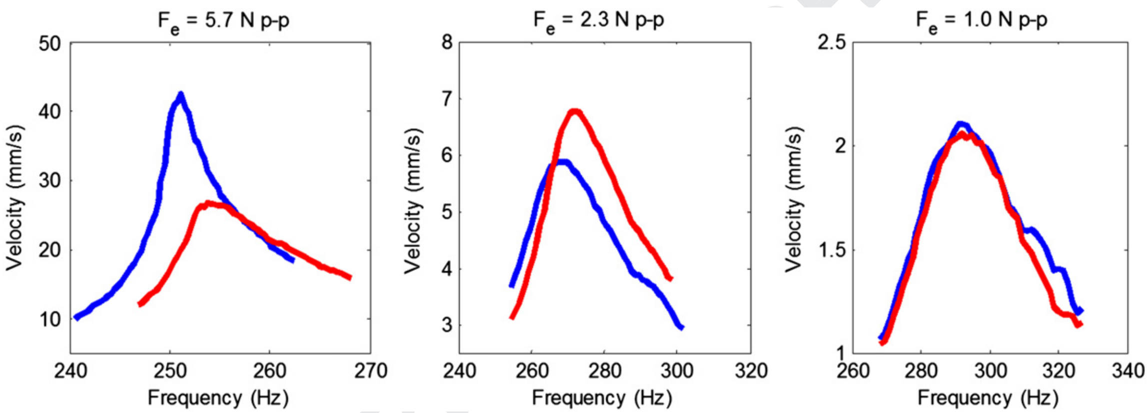


Fig. 10. Linear forced response (mobility (dB)—re 100 mm/sN).



Q5 **Fig. 11.** ND4, EO20, 1st installation—repeatability of the measurement on the 1st blade. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

the contact surfaces (wear, debris, etc.). At present, in order to obtain consistent measurements from the 24 blades, the repeatability of the experiment is checked at each run, by measuring the 1st blade two times: the first time at the beginning of the measurements and the second time at the end of the measurement after the 24th blade.

An initial scoping, after the first installation of the underplatform dampers, was performed by measuring the ND4 mode. All 24 blades were measured at three different levels of magnetic force (5.7 N, 2.3 N and 1.0 N). Fig. 11 shows that the repeatability improves by keeping the bladed disc spinning for a long time and, in fact, that is proved by looking at the blue and the red lines, which refer to the beginning and the end of the test, respectively.

At the end of the 1st experimental campaign, the dampers were removed from the blisk, their surfaces cleaned from the debris accumulated during the tests, and reinstalled in the blade cavities. A second run was performed around the ND4 mode (magnetic force: 6.0 N, 2.4 N, 1.0 N). As shown in Fig. 12, the repeatability has improved with respect to the previous installation, as confirmed also by the results of Fig. 13, referring to the experiment run around the ND2 resonance.

4.3.2. *Effect of mistuning*

As observed in Section 2.2 with the preliminary modal analysis of the non rotating blisk, the effect of mistuning on ND2, ND3 and ND4 mode shapes is mostly visible in terms of natural frequency split and not in terms of modal distortion. Nevertheless, significant scatter of the forced response of the blades is observed experimentally, as shown in Fig. 14, which refers to a ND2 resonance (F_e=3.1 Np–p). A more careful analysis of the response curves shows that their pattern is far from being random. In Fig. 14, for instance, blades located at 90° of angular distance from each other have very similar response curves, as shown in Fig. 15, where the response curves are grouped by four.

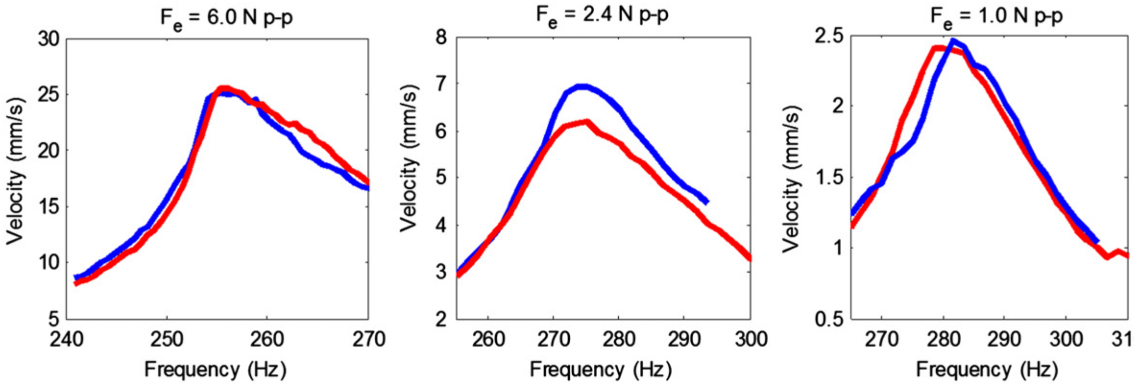


Fig. 12. ND4, EO20, 2nd installation—repeatability of the measurement on the 1st blade.

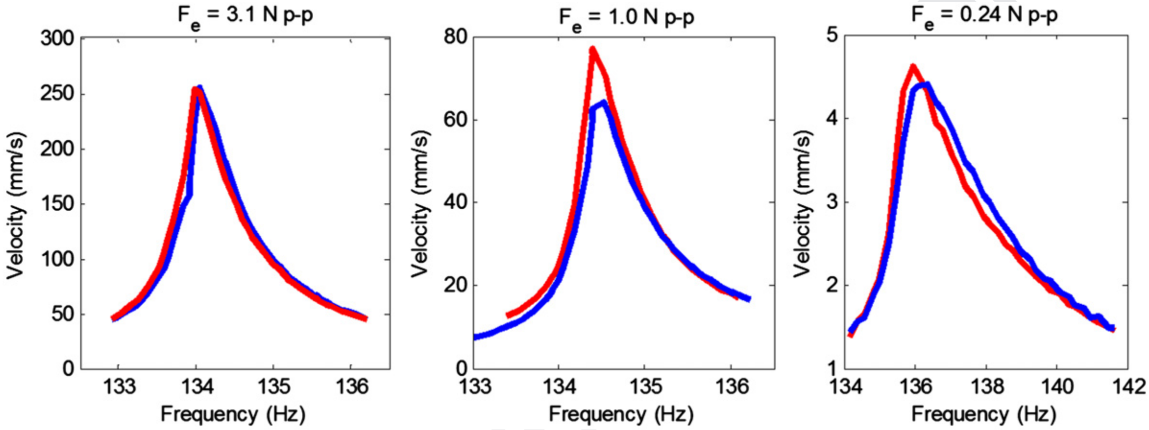


Fig. 13. ND2, EO22, 2nd installation—repeatability of the measurement on the 1st blade.

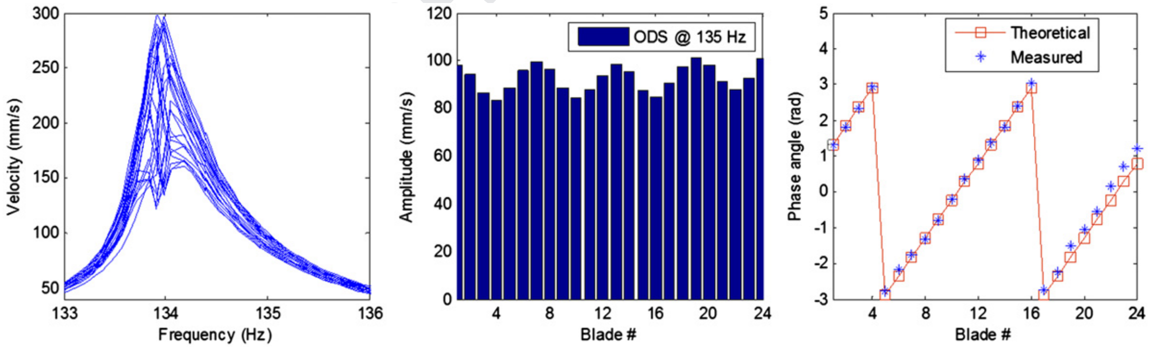


Fig. 14. Response of 24 blades at ND2 resonance with F_e=3.1 N p–p (left); ODS measured at 135 Hz: amplitude (centre) and phase (right).

In order to check whether the scatter of the blade response could be attributed to the frequency split induced by the geometrical mistuning of the blisk, the following analysis has been performed. A linear system made of 24 blades is studied numerically. Consistently with the results of the modal analysis of Section 2.2 (Figs. 7 and 8), it is assumed that the geometrical mistuning does not affect the mode shapes of the system and therefore the dynamic behaviour of the system around any resonance whose mode shapes has ND nodal diameter is modelled by means of two harmonic mode shapes defined as

$$\Phi_1 = \cos\left(\frac{2\pi\, ND\, n}{N_b}\right) \quad \Phi_2 = \sin\left(\frac{2\pi\, ND\, n}{N_b}\right) \text{ with } n = 1\ldots N_b \tag{3}$$

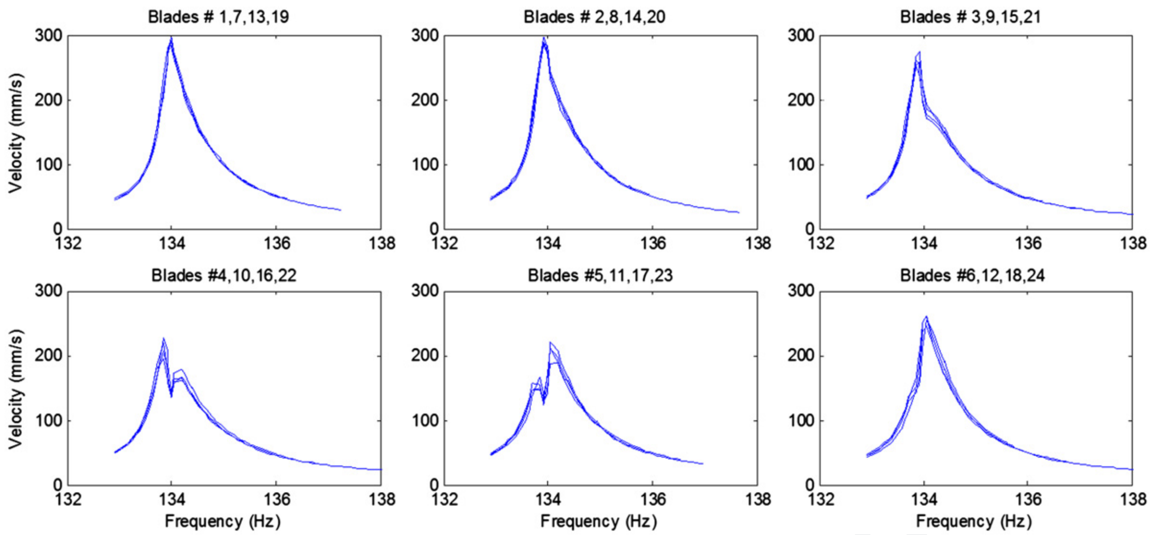


Fig. 15. Measured response curves of groups of 4 blades at ND2 resonance.

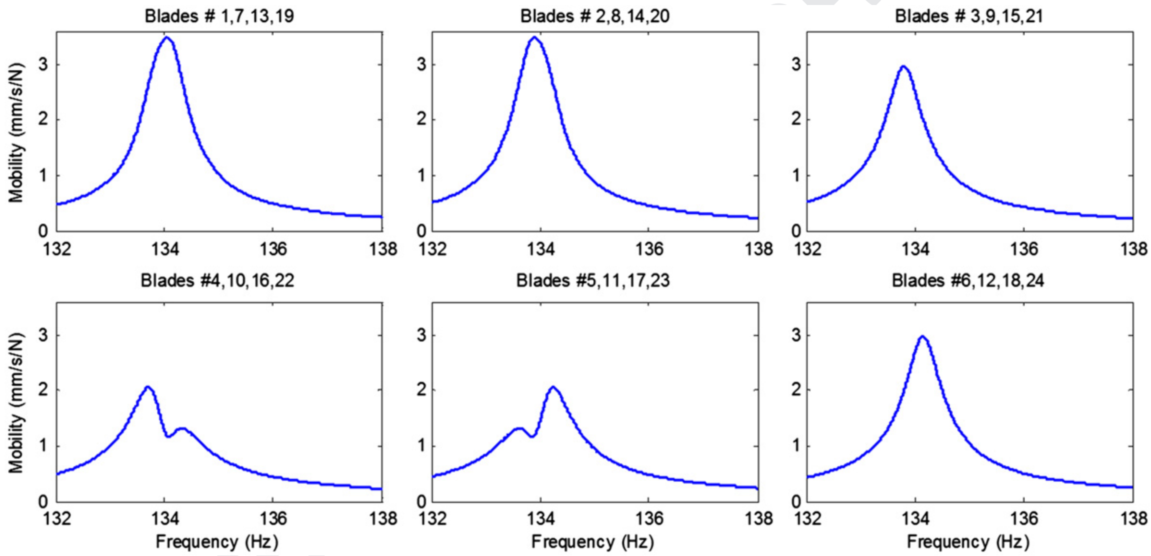


Fig. 16. Forced response computed at ND=2 with $\zeta=2.5\text{E}-3$.

The mistuning affects the system in terms of frequency split and therefore in the model the experimental natural frequencies ω_1 and ω_2 are used. Then the forced response of the blisk around the ND resonance is computed as

$$X(\omega)=[\Phi_1 \Phi_2]\begin{bmatrix} \frac{1}{\omega_1^2-\omega^2+i2\zeta\omega_1\omega} & 0 \\ 0 & \frac{1}{\omega_2^2-\omega^2+i2\zeta\omega_2\omega} \end{bmatrix}\begin{bmatrix} \Phi_1^T \\ \Phi_2^T \end{bmatrix} F_{eo} \tag{4}$$

with a modal damping $\zeta=2.5\text{e}-3$, and an engine order excitation F_{eo} of unit amplitude defined as

$$F_{eo}=\cos\left(\frac{2\pi\, ND\, n}{N_b}\right)+i\sin\left(\frac{2\pi\, ND\, n}{N_b}\right) \quad \text{with} \quad n=1\ldots N_b \tag{5}$$

Results are shown in Fig. 16, where the mobility of the blades are shown for the same groups of blades of Fig. 15. By comparing the experimental results of Fig. 15 with the numerical results of Fig. 16, a very good agreement between the response patterns is observed, confirming that the scatter in the response curves measured during the tests is due to natural frequency split induced by the geometrical mistuning of the blisk.

4.3.3. *Effect of the external excitation*

The installation of underplatform dampers on the blisk produces non-linear responses. The friction forces occurring at the blade/damper contacts depend on the displacements of the contact points and therefore on the vibration amplitudes of the blades. The effectiveness of underplatform dampers in dissipating energy depends strongly on the level of the excitation force acting on the blades. So, that level depends on the air gap between the rotating blisk and the permanent magnet fixed on the test rig. The relationship between the air gap and the excitation force, measured at 60 rev/min, is shown in Fig. 17. Clearly, a fixed magnet passed by rotating blades introduces safety risks because of possible and, thereby catastrophic, impact with the exciter. The maximum excitation force, for a given permanent magnet, is limited by the minimum and safe air gap between the blade and the magnet and so it was decided to keep that to a minimum of 3–4 mm. The larger distance is determined by the signal-to-noise ratio (SNR) set as a measurement parameter.

The mobility of a selected blade measured around the ND3 resonance is plotted in Fig. 18. The measurements were obtained for different levels of excitation force (from 0.75 N to 6.0 N peak-to-peak) and the non-linear behaviour of the system is clearly visible in terms of both vibration amplitude and resonance frequency, showing that the underplatform dampers introduce both damping and stiffness to the blisk. The mobility becomes lower as the external excitation decreases, while a softening effect due to increase of the excitation level is also observed, since the resonance frequency of the system decreases as the external excitation grows larger. This behaviour can be physically explained referring to the amount of slip at the damper/blades contact surfaces. As the excitation level increases, the amount of slip becomes larger. As a consequence, the damper becomes a weaker constraint to the blade vibration and the resonance frequency decreases. At the same time, as the amount of slip increases, the energy dissipated by friction increases proportionally, and so does the damping, at least in the range of excitation levels used in the experiments.

According to theoretical models [5,10] and experimental tests in static rigs [3,6] a further reduction of the external excitation would lead to a minimum value of the mobility up to an increase when the dampers would tend to the full sticking conditions. For the presented tests it was not possible to apply a level of magnetic force lower than 0.75 N because that would reduce too much the vibration level measurable using the SLDV and so compromising the SNR quality.

When the maximum response of the blades is plotted versus the excitation level, as shown in the right hand plot in Fig. 18, a piece-wise linear curve is obtained. The peak velocities measured on the blades show a moderate increment between 0.75 and 3 N whereas a large increase (almost 400%) from 3 N to 6 N. This plot demonstrates that the first part of the curve, between 0.75 N and 3 N, represents a range of performing underplatform damper and thereby highlighting the robust design of them.

In terms of design rules and design criteria for UPDs, it means that despite the uncertainties usually related to the unsteady aerodynamic forces acting on bladed disks, a robust design of UPDs is possible because the damper performs well, and therefore the blisk maximum response increases slowly, in a given range of excitation levels.

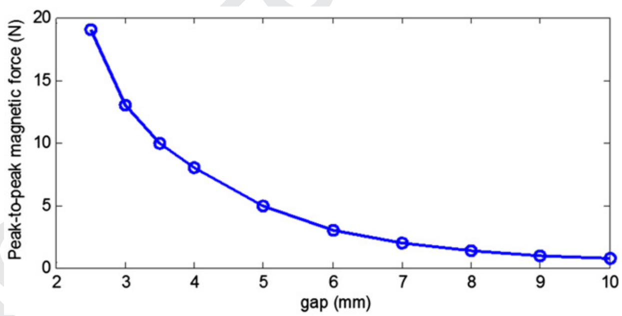


Fig. 17. Relationship between air gap and magnetic force.

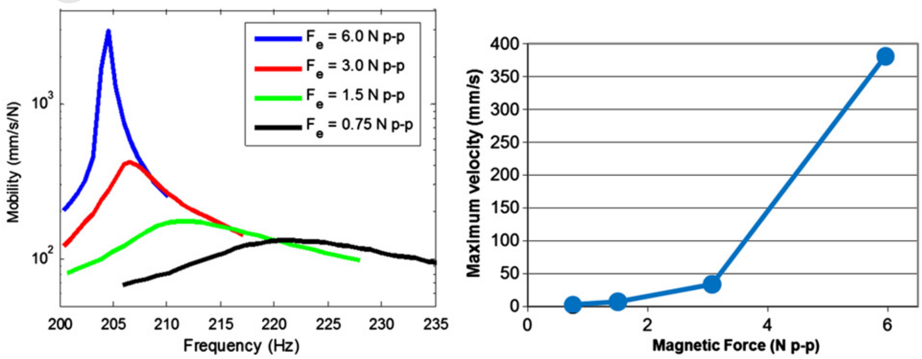


Fig. 18. ND3: effect of excitation level on the mobility (left) and on the maximum response (right).

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4.3.4. *Effect of the inter-blade phase angle*

This work presents a study of forced response measurements of a rotating blisk subjected to travelling wave excitation and this produces a system responding with a rotating wave. The time delay between two consecutive blades depends: (i) on the number of the blades, N_b, (ii) on the rotation speed and (iii) on the number of nodal diameters of the mode shape for an excited resonance. Different time delays imply different kinematics of the blade platforms in contact with any damper, and therefore different periodic contact forces are generated at the blade/damper interfaces; eventually, underplatform dampers can perform differently from the original design specifications. In order to investigate this effect, the forced response of the blisk at the ND2 and ND4 resonances has been also measured and these results were compared with those ones obtained from the ND3 mode. All the experiments refer to the 2nd installation of the dampers, which produced good repeatability results. The excitation force is applied on the rotating blisk by using a permanent magnet and so it is not concentrated into a single harmonics, but distributed over the whole spectrum (Fig. 4). Hence, the amplitudes of the engine orders exciting the blisk are different form one another for a given level of magnetic force. The comparison of the responses measured for different nodal diameters needs to be done by extracting from the time history of the force measurement as described in Section 2.2, and used in the plots. The maximum blade tip velocity measured using the tracking measurement method is shown in Fig. 19 for ND2–ND4 mode shapes versus the amplitude of the engine order components exciting the blisk. The curves referring to ND2 and ND3 resonances have the same shape as shown in Fig. 18. The quasi horizontal part of the curve, corresponding to a robust design of the underplatform dampers, is shorter for ND2 than for ND3 mode shape. In case of ND4 resonance, the curve is very flat over the whole range of excitations, and so the larger is the number of nodal diameters the wider is the range corresponding to a robust design of the underplatform dampers.

Fig. 20 shows the effect of the underplatform dampers when the resonance frequency shifts are plotted versus the engine order amplitudes. Theoretically for an infinite value of the engine order amplitude, all the three curves tend towards the asymptotic value corresponding to the resonance frequencies of the linear blisk without underplatform dampers. The frequency shift induced by the damper to the ND4 mode shapes is larger (27 Hz) than the shift occurring at the ND3 (17 Hz) and ND2 (5 Hz) mode shapes.

The experimental evidence collected and shown in Figs. 15 and 16 is consistent with the experimental tests available in the literature referring to static rigs [2,3,6]. In those cases, the in-phase and out-of-phase bending modes of two cantilevered blades with interposed an underplatform damper were measured. Those results showed that for the in-phase modes (corresponding to a 0 ND mode shape on a rotating blisk), the underplatform dampers are poorly effective in dissipating energy by friction and the frequency shift induced by the damper was negligible. In contrast, for the out-of-phase vibration (corresponding to $ND = N_b/2$ mode shapes on a rotating blisk), underplatform dampers are effective in a very wide range of excitation forces, introducing also a relevant frequency shift. The reason of the different effect of the underplatform dampers on the dynamics of the system for different values of nodal diameters (i.e. different contact kinematics) has been correctly explained [3] numerically predicted [4–7] and experimentally proved [12] to be the different amount of damper rotation. At low nodal diameters, the relative displacement of the blade platforms produces a relevant damper rotation but its displacement in the radial direction is negligible and therefore a limited amount of slip

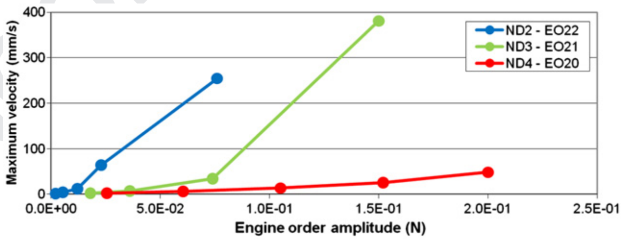

Fig. 19. Maximum velocity at the blade tip vs EO amplitude for different ND#.

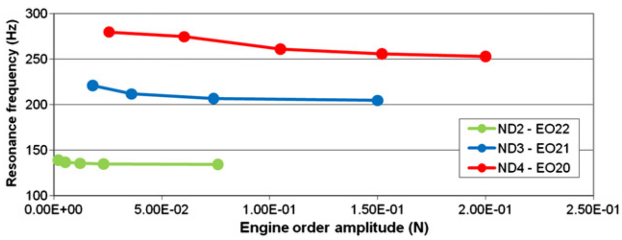

Fig. 20. Resonance frequency vs EO amplitude for different ND#.

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occurs; on the contrary at high nodal diameters the damper rotation is negligible, the damper mostly moves in the radial direction and a larger amount of slip is generated.

5. Conclusions

This paper presents an experimental study of the effect of underplatform dampers on the dynamics of bladed disks. The work has been produced using an experimental rotating test rig equipped with non-contact excitation and measurement systems, this being important for preventing additional and unpredictable mistuning cause by the transducers. The measurements were performed by using tracking techniques developed by using a Scanning Laser Doppler Vibrometer. This new technique overcame some limitations of a previously developed technique, which was based on the self-tracking mechanical method. The Point track method worked by moving the laser beam synchronously with the target. This was achieved thanks to a tachometer, installed on the shaft, controlling the scanning mirrors of the SLDV laser head. The use of tracking Scanning LDV methods for measuring vibrations of rotating structure such as bladed discs is of invaluable importance to succeed in such experimental studies.

The effects of the inherited geometrical mistuning of the blisk have been measured and reproduced with a simplified linear model. The effects of underplatform dampers on the first bending vibration mode of the blades have been investigated for different levels of the excitation force and for different engine order values. The non-linear relationship between the force and the vibration amplitude has been observed and measured experimentally so as to identify a range of excitation forces within which a robust design of the damper worked efficiently. The effects of the number of engine order excitation on the damper performances has been systematically measured. These showed that dampers are effective in reducing vibrations of blades in a range of excitation force levels, which increases as much as the number of nodal diameter associated to the excited mode shapes becomes larger.

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