A BENCHMARK FOR TIP TIMING MEASUREMENT OF FORCED RESPONSE IN ROTATING BLADED DISKS

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A BENCHMARK FOR TIP TIMING MEASUREMENT OF FORCED RESPONSE IN ROTATING BLADED DISKS

Giuseppe Battiato*[1], Christian M. Firrone[2],Teresa M. Berruti[2]

[1]Dep. of Mechanical Engineering
Politecnico di Torino
Corso Duca degli Abruzzi, 24
10129 - Turin, Italy
giuseppe.battiato@polito.it

[2]Dep. of Mechanical Engineering
Politecnico di Torino
Corso Duca degli Abruzzi, 24
10129 - Turin, Italy
{christian.firrone,teresa.berruti}@polito.it

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Abstract. *The Blade Tip-Timing is a well known non-contact measurement technique for the identification of the dynamic properties of rotating bladed disks. Even if it is an industry-standard technique its reliability has to be proved for the different operation conditions by comparison with other well established measurement techniques. Typically a strain gauges system in conjunction with radio telemetry is used as reference.*

This paper aims at evaluating the accuracy of a last generation Tip-Timing system on two bladed dummy disks characterized by different geometrical, structural and dynamical properties. Both the disks were tested into a spinning rig where a fixed number of permanent magnets, equally spaced around the casing, excites a synchronous resonance vibration with respect to the rotor speed.

The so called beam shutter method was adopted for the Tip-Timing system. Due to the presence of shrouds a particularly set up of the probes was chosen in order to avoid that the probes look radially inward at the blade tips as in the most common configurations.

The probes are optical laser sensors pointing at leading and trailing edges locations where the blade experiences the greatest magnitude of displacement. The Blade Tip-Timing measured data are post-processed by two different methods, the Single Degree of Freedom Fit (SDOF) and the Circumferential Fourier Fit (CFF). The amplitude and frequency values at resonance obtained by the Tip-Timing system are compared with those obtained by the strain gauge measurements.

1 INTRODUCTION

Vibrations of turbomachinery blades in service could reduce their fatigue life by increasing the risk of crack formation. In this frame the blade health monitoring is an important challenge in order to prevent unexpected blade failures. Traditionally, the rotor blade vibrations have been detected using strain gauges which still represent nowadays the most reliable measurement system. The strain gauges have the disadvantage that, since they need to be stuck on the blade airfoil, they cannot be used in the engine in service. For this reason in the last years for the blades monitoring, the non-contact measurement technique Blade Tip-Timing (BTT) has gained ground.

The BTT technique is based on the analysis of arrival times related to the blade passages under a set of stationary sensors [1, 2]. In the standard cases the BTT sensors are mounted on the casing, oriented radially in order to detect the blade displacements at the tip [3]. One of the main advantages of the BTT is that, besides being non contact and usable on a working engine, all the blades are monitored during the rotation, giving the possibility to identify if one blade vibrates more than the others (for typical mistuning problems).

The last generation industry-standard BTT systems process the arrival times by using indirect [4, 5, 6, 7] or direct [5, 8, 9, 10] identification methods in order to determine the modal parameters characterizing the vibrating blade. A verification on the correctness of the obtained parameters represents a due step in order to consider a BTT system as reliable. Different authors have demonstrated a good correlation between measurements from BTT and strain gauges for both real [3] and controlled [11] operation conditions, in the case where the displacements are measured at the blades tip using the standard radial sensors positioning.

This paper presents results and validation of a new way of using optical sensors in the BTT, the so called *beam shutter* or *beam interrupted* configuration. This configuration is adopted for the identification of the modal parameters of a disk by measuring the displacements at the mid leading or trailing edge of the blades. Instead of the standard configuration for optical probes, where the laser beam is reflected by the measured blades, in the beam shutter configuration the laser beam is interrupted by the blade passage. The technique is here tested on two different dummy disks that experience synchronous vibration patterns when excited by means of a fixed set of permanent magnets. The first dummy disk has the simple geometry of a flat plate and it is characterized by blade bending vibration modes along the axial direction. The second dummy disk was designed to simulate a dynamic behavior closer to a real turbine disk where the blades are connected each other at the tip by an outer ring. The test campaign was performed such that the blade vibrations could be detected simultaneously by using the BTT and strain gauges measurement systems.

2 THE SPINNING TEST RIG

The dummy disks were tested in a laboratory spinning rig under vacuum conditions [12]. As shown in Figure 1-a the test rig has vertical axis with two cylindrical protection structures (1) and (2) which are mounted coaxial to the rotating shaft which is positioned under the floor. On the top of the shaft a flange allows the disk accommodation (3). The cylinder (1) also supports two static rings (Figure 1-b) aimed at keeping in a fixed position a set of permanent magnets (ring 1), which are used to excite the rotating disk, and the Tip-Timing laser sensors (ring 2).

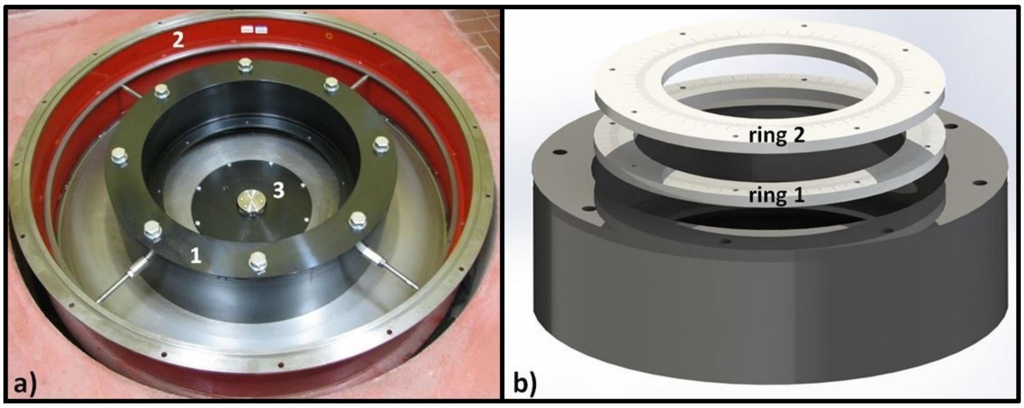

Figure 1: a) The spinning rig, b) static rings for magnets and laser sensors.

2.1 The dummy disks

The dummy disk 1 (Figure 2-a) is an aluminum disk with a simple geometry of a flat plate where each blade has the shape of a cantilever beam. The disk 1 has 12 blades whose length and width are 150 mm and 25 mm respectively. The axial height and the outer diameter of the disk are 5mm and 400 mm. A cylindrical magnet (5 mm diameter, 5 mm height) is glued in a special housing machined at each blade's tip.

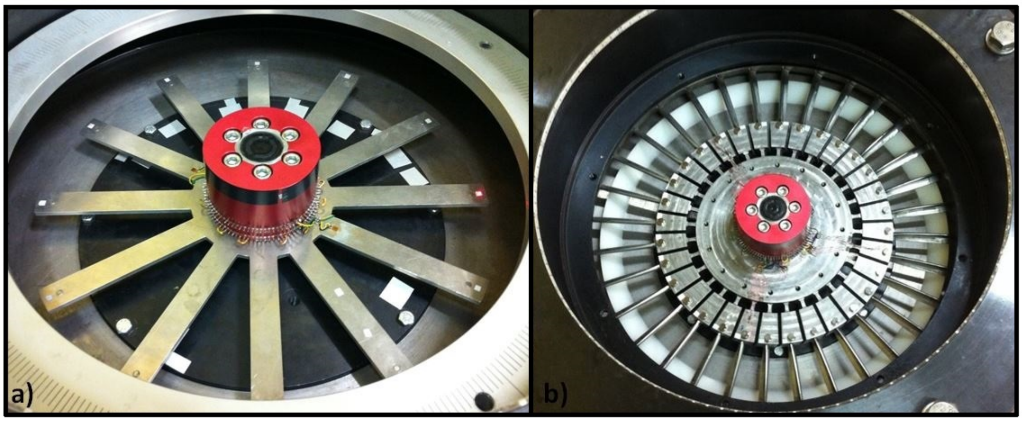

Figure 2: a) The dummy disk 1, b) the dummy disk 2.

The dummy disk 2 (Figure 2-b) was designed to be closer in the dynamic behavior to a real turbine disk where the blades are connected each other at the tips by an outer ring as in the case of shrouded blades. It is a single piece made of ferromagnetic steel AISI 460 in order to allow the magnetic interaction between the permanent magnets and the blade airfoils. The dummy disk 2 has 32 real profiled blade whose length and aspect ratio are respectively 100 mm and 7,31. Its outer diameter and axial height are 630 mm and 20 mm.

2.2 The excitation system

The excitation system in the spinning test rig uses cylindrical permanent magnets (diameter 18 mm, height 10 mm, magnetization N52). The magnets are mounted in equally spaced positions on the static ring facing the rotating test article (ring 1 in Figure 1-a). A graduated scale impressed on the ring upper surface is used to fix the magnets at the right angular positions [12].

Different supporting rings with different inner diameter are available in order to guarantee the correct radial positioning of the permanent magnets. Each magnet is glued on the tip of a screw which allows the regulation of the axial gap with respect to the test article blades. Six of the magnets are instrumented with force transducers which measure the force in axial direction during the tests [12].

The number of magnets used in a specific test must be equal to the main engine order (EO) characterizing the excitation force that should be simulated. The main engine order EO^1_m can be defined as the fist not-null harmonic index resulting from the Fourier transform of the excitation force. In general for m equally spaced magnets, the EO pattern exciting the disk is defined as follows:

$$EO^i_m = i \cdot m, \qquad \forall\, i = 1, 2, 3, \ldots \tag{1}$$

A modal shape corresponding to a certain number of nodal diameter (ND) can be excited by different EO according to the following relationship:

$$EO = jN_b \pm ND, \qquad \forall\, j = 0, 1, 2, 3, \ldots \tag{2}$$

Where N_b is the number of blades characterizing the disk. The excitation frequency is related both to the EO and to the rotational speed n by the following equation:

$$f_{exc} = \frac{EO \cdot n}{60} \tag{3}$$

Where n is expressed in rpm and f_{exc} in Hz.

From the Eq. 3 it can be stated that higher EO values can correspond to the same f_{exc} for lower rotational speeds.

2.3 The strain measurement system

The two dummy disks were instrumented by means of strain gauges. For both the disks the identification of the strain gauge positions came out from their FE modal analyses in cyclic symmetry conditions. Areas of high strains and low strain gradients were identified as best locations for strain gauges. The strain gauges signals are acquired through a telemetry system.

For the dummy disk 1 the strain gauges where attached at the two sides of the blade root (Figure 3-a). This position was chosen in order to measure the out-of-plane bending (1F) mode belonging to the first modal family. Each strain gauge is composed by a single grid (1.52 mm $\times$ 3.05 mm, grid resistance 350 Ω). The two grids at the two sides of the blade were connected together as a half bridge. For the dummy disk 2 the strain gauge was attached on the back of the blade airfoil (Figure 3-b). The selected area is not affected by strain gradients for both the flap-restricted (1FR) and torsional (1T) mode shapes belonging to the second and third modal family respectively. The strain gauge is a tee rosette (grid resistance 350 Ω) composed by two separate grids (1.52 mm $\times$ 1.78 mm) with perpendicular axes. The two grids were connected together as a half bridge.

The strain gauges measurement chain was verified by means of a static test on the dummy disk 1. The static test on the instrumented blade consists in measuring the bending strain at the blade root due to calibrated masses positioned on its free end.
Three static tests corresponding to three different dead masses were performed. The measured

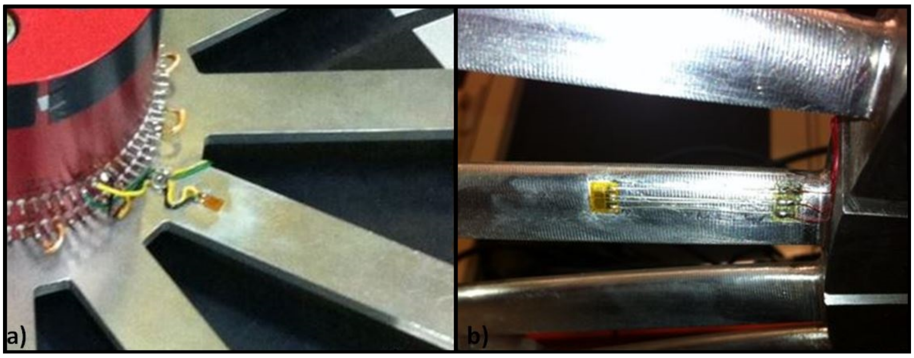

Figure 3: a) Strain gauge at the blade root of the dummy disk 1, b) strain gauge at the back of the airfoil for the dummy disk 2.

Mass	*Strain Gauge*	*FE model*
kg	$\mu\epsilon$	$\mu\epsilon$
0,76	130,23	130,19
1	169,30	169,27
1,99	342,32	342,25

Table 1: Experimental and numerical strains for static tests on dummy disk 1.

strains were compared to the corresponding numerical strains determined by means of the static FE analysis.
The results listed in Table 1 show the reliability of the strain gauges which can be used as a reference measurement system for the validation of the BTT technique.

3 THE BLADE TIP TIMING SYSTEM

The Blade Tip Timing technique uses a set of non contact sensors constrained to the casing and facing the rotating disk [1, 2, 3]. This technique is based on the measurement of the Times-of-Arrival (ToA) of each blade under each of the stationary sensor. From the analysis of the ToA the dynamic properties of each blade, i.e. the resonance frequency, the vibration amplitude and the damping, can be determined. The BTT has two main advantages over the strain gauges:

- it is a non-contact measurement system that does not affect the dynamic behavior of the blade;
- it allows the measurement of all the blades of the disk, while the strain gauges are usually attached only to some blades.

The BTT system used in this activity is a last generation system with laser optical probes. For the dummy disks 1 and 2 all the tests were performed by employing 5 optical laser sensors for detecting the blades vibrations. An additional sensor (1/rev sensor) which measures the rotational speed was used a reference for the other sensors.

The BTT software which post-processes the measured blades displacements uses two different approach to extract the vibration data:

- *Single Degree of Freedom Fit* (SDOF): it is based on the analysis of data collected during an rpm sweep. The data coming from only a single sensor can provide the resonance parameters. The curve measured by sensor for one blade has one of the shapes shown in Figure 4. The peak-to-peak values are equal to each other and represent the zero-peak value of the actual response in resonance condition [4].

- *Circumferential Fourier Fit* (CFF): it uses a different mathematical approach to determine the resonance parameter that the SDOF. This method requires 3 or more sensors installed on the same chord-wise position. Assuming a certain response order, corresponding to the selected EO, for each averaged rpm a least mean squares fitting of the data with a sinusoidal wave is performed [5, 8, 9, 10].

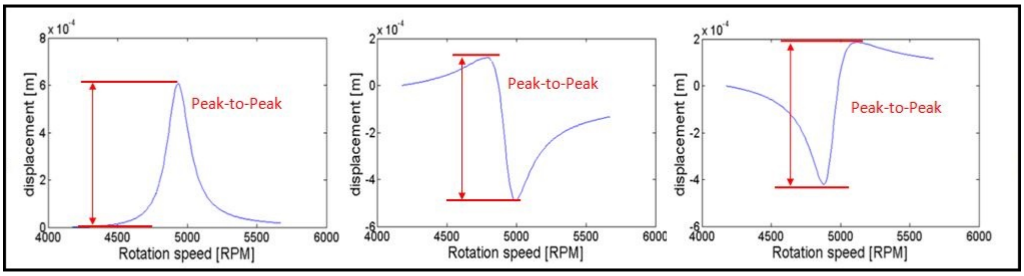

Figure 4: Curves for one blade measured by three sensors placed at 0° 120° and 240° respectively.

3.1 The beam shutter method and sensor positioning

The standard BTT measurement approach is based on sensors which are positioned in radial direction in order to measure the blade displacement at the tip. In this activity a new sensor configuration called *beam shutter* or *beam interrupted* has been tested. The beam shutter configurations employed in the spinning test rig for the dummy disks 1 and 2 are shown in Figure 5.

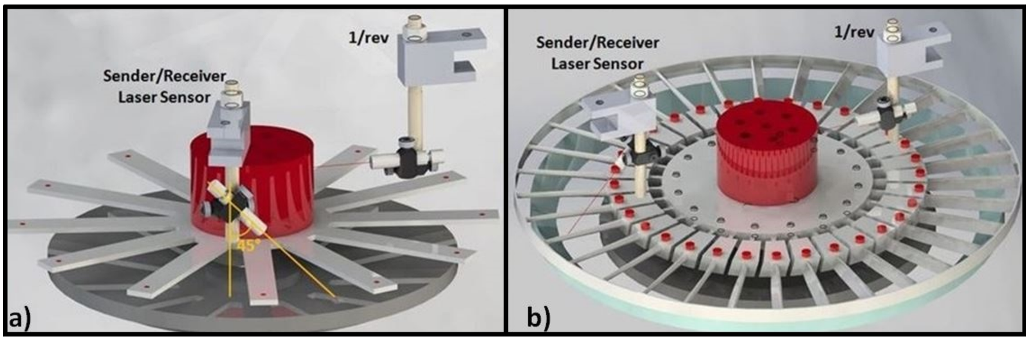

Figure 5: Beam shutter configuration implemented for the dummy disk 1 a) and the dummy disk 2 b).

A set of sender/receiver optical laser sensors with lenses was used for both the disks. Each sensor is mounted above the disk and produces a laser beam which is collimated on a reflective tape stuck on a stationary surface placed under the disk. When the disk rotates the passing blade acts as a shutter, blocking the returning light towards the sender/receiver sensor. The system is set up in order to detect the ToA of the leading or the trailing edge of the blade.

The sensors were installed at the same distance from the disk center. The relative circumferential position of the sensors was chosen in both cases by means of an optimization tool which allows to avoid the aliasing effect in the measurement of the traveling wave characterizing the vibrating disk. The optimization tool requires as input the number of sensors, the engine order of the travelling wave of and the number of blades.

4 THE RESULTS COMPARISON METHOD

The vibration parameters detected by the BTT system have to be compared to the same parameters coming from the strain gauges connected to the telemetry system. Simultaneous measurements with BTT and strain gauges were performed for a certain set of vibrating modes characterizing the dummy disks.

The comparison in terms of resonance frequencies is straightforward and requires a fast data processing. The comparison in terms of vibration amplitudes is a more demanding task. It requires a finite element (FE) modal analysis of the disks, since the strain gauges system measures strains (ϵ_{SG}), while the BTT measures displacements (u_{BTT}).

By the FE model the parameter $K_{mod} = u_{mod}/\epsilon_{mod}$ can be calculated, where:

- u_{mod} is the modal displacement of the node corresponding to the laser position on the blade in the same direction of the displacement detected by the BTT;

- ϵ_{mod} is the modal strain in the area corresponding to the strain gauges position in the same direction of the strain detected by the strain gauges.

The same parameters can be defined for the physical measured quantities u (displacement of the blade at the BTT laser position) and ϵ (strain at the strain gauges position) $K_{phy} = u/\epsilon$. Since the two disks can be considered linear and their responses give well separated modes, the following relationship should be satisfied:

$$K_{phy} = K_{mod} \quad \Rightarrow \quad \frac{u}{\epsilon} = K_{mod} \tag{4}$$

The displacement of the blade corresponding to the strain measurement from Eq. 4 can be determined as:

$$u_{SG} = K_{mod} \cdot \epsilon_{SG} \tag{5}$$

Since the FE model was tuned previously by the strain gage measurements (see Table 1) the obtained parameter u_{SG} is considered as reference for the displacement value measured by the BTT system u_{BTT}.

5 RESULTS ON DUMMY DISK 1

In order to predict the natural frequencies and the modal shapes a FE dynamic calculation in cyclic symmetry conditions was performed. In Figure 6-a the natural frequencies of the disk are plotted against the nodal diameters. The modal family plotted in Figure 6-a is the family $1F_{oop}$ where each blade vibrates according to a bending out-of-plane mode shape for each ND. The calculated numerical Campbell diagram is shown in Figure 6-b. From this plot the rotational speeds at which the resonances occur can be estimated. The modes corresponding to 5 and 6 ND were chosen to be tested. The mode at ND6 is excited with E0 6, while, in order to avoid high operation speed, it was chosen to excite the mode at ND5 not by EO 5 but by EO 7 according to Eq. 2.

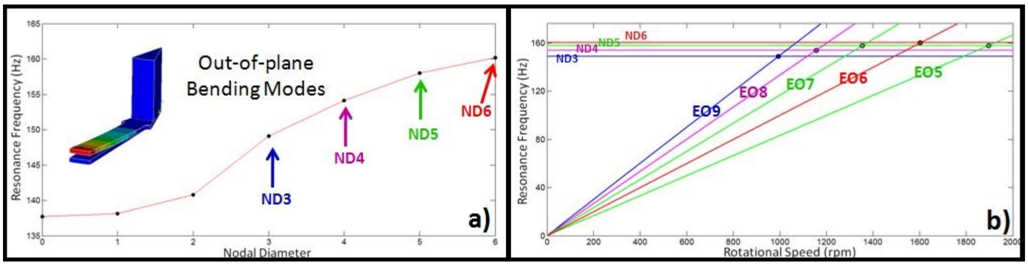

Figure 6: a) numerical ND-resonance frequency diagram for the dummy disk 1, b) numerical Campbell diagram for the dummy disk 1.

5.1 Displacement measurement with the Tip-Timing

The vibration modes of the dummy disk 1 are out-of-plane (axial direction). In order to obtain a configuration where the optical beam is interrupted by the blade during an out of plane vibration, the sensors must be positioned with an angle with respect to the disk plane as shown in Figure 5-a. Each sensor was then tilted by an angle which is $45°$ with respect the plane containing the undeformed disk. This particular angle of the sensors allows the BTT system to see an axial displacement as it was tangential as shown by the scheme of the blade interrupting the laser beam in Figure 7-a. The detected ToA corresponds to the apparent tangential displacement which is equal to the axial displacement.

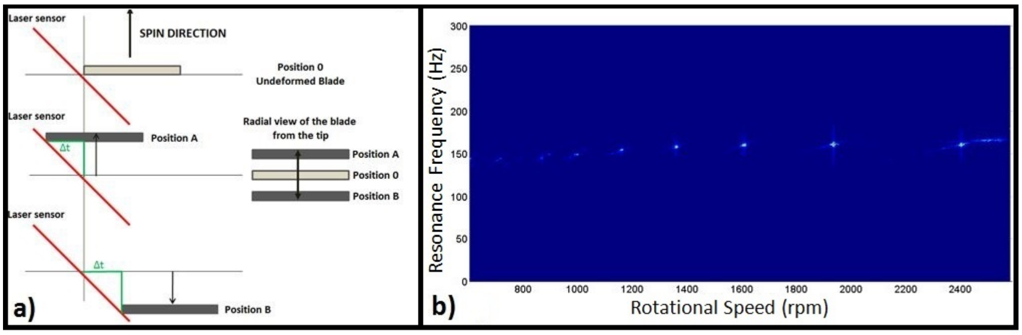

Figure 7: a) Beam shutter working principle adopted for detecting out-of-plane vibration mode for the dummy disk 1, b) experimental Campbell diagram for the dummy disk 1.

5.2 Preliminary test

A preliminary rotating test was performed to identify the real resonance frequencies. A single magnet was mounted on the test rig so that the disk could be excited simultaneously by means of an infinite number of harmonic excitations. The test was planned in order to investigate the speed range 600 rpm - 2600 rpm with an acceleration of 1.6 s/Hz.

From the strain gauges time signal the *experimental* Campbell diagram (Figure 7-b) was processed using Matlab. The natural frequencies for ND 5 and 6 which were determined by reading the y-axis values of the white points are listed in Table 2.

EO	*ND*	n_{res}	f_{res}
-	-	rpm	Hz
7	5	1356	158,2
6	6	1591	159,1

Table 2: Preliminary test results obtained for a 10 mm gap between the fixed magnet and the rotating blade.

5.3 Test campaign

The test campaign was performed for the selected nodal diameters which can be excited by the EO determined from Eq. 2 for j = 1. A single permanent magnet was adopted for the excitation. A gap of 7 mm between the permanent magnet and those glued on the blades was set for all the tests. Strain and displacement were acquired simultaneously for 2 speed ranges including the values of n_{res} at which the resonances occur.

The main tests parameters are listed in Table 3. The test campaign was repeated 3 times.

EO	*ND*	n-range
-	-	rpm
7	5	1300 - 1400
6	6	1550 - 1650

Table 3: Test campaign for the dummy disk 1.

For each test a linear speed sweep with acceleration of 3.2 s/Hz was performed.

5.4 The test results

The strain gauges and BTT data were processed for each of the studied modes. For the strain gauges data processing the same procedure employed for the preliminary test was used. Both the SDOF identification and the CFF methods were adopted for BTT data processing. Since the two identification method gave very close results, only the results produced by the SDOF are here reported. The measured displacement amplitude values in resonance condition and the correspondence resonance frequencies were averaged over three acquisitions. The measured values showed high repeatability (standard deviation $\bar{\sigma} < 0.13\%$).

In Table 4 the displacement amplitude values and the natural frequencies detected both by stran gauges (SG) and by the BTT system are listed. The parameters e_f, e_u are the percentage differences between the resonance frequencies and vibration amplitudes values detected by the strain gauges and BTT, that is:

$$\begin{aligned} e_f &= \frac{\bar{f}_{SG} - \bar{f}_{BTT}}{\bar{f}_{SG}} \cdot 100 \\ e_u &= \frac{\bar{u}_{SG} - \bar{u}_{BTT}}{\bar{u}_{SG}} \cdot 100 \end{aligned} \tag{6}$$

From Table 4 it can be noted that the difference in terms of resonance frequency is negligible (e_f is lower than 0.5%). The difference in terms of resonance amplitude is also very low, the paramer e_u is less than 2%.

ND	$\bar{f}_{SG}$	$\bar{f}_{BTT}$	e_f	$\bar{u}_{SG}$	$\bar{u}_{BTT}$	e_u
-	Hz	Hz	$\%$	μm	μm	$\%$
5	158,0	158,0	0	1662,23	1631,48	1,88
6	159,3	160,2	0,44	2298,75	2275,60	1,02

Table 4: BTT - strain gauges comparison for the dummy disk 1.

6 RESULTS ON DUMMY DISK 2

The dummy disk 2 shows a dynamic more similar to real turbine disk. Measurements on this disk allow testing the capability of the BTT system to give accurate results even on a disk with a more complex dynamics. The FE dynamic calculation of the dummy disk 2 showed that the first three modal families are the flap (1F), flap restricted (1FR) and torsional (1T) modes. The plot of the calculated frequencies-nodal diameters for these three modal families is shown in Figure 8-a. In the selected position chosen for the strain gauges (middle on the blade airfoils, Figure 3-b), the FE model showed that the highest strains are given by the 1FR and 1T modes. It was then decided to excite in the tests these two modes for a certain nodal diameter in order to compare BTT and strain gauges measurements.

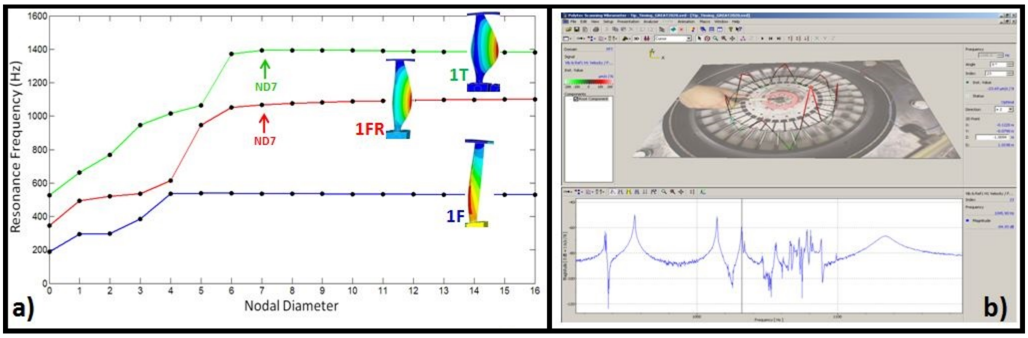

Figure 8: a) numerical ND-resonance frequency diagram for the dummy disk 2, b) hammer test on dummy disk 2: 1FR mode identification.

6.1 Preliminary test - Hammer test

A preliminary hammer test was performed to detect the modal parameters of the disk in static conditions (not rotating). In order to have the same constraints with the shaft as during the rotation, the disk was kept mounted on the spinning rig during the hammer test. The response of each blade was measured by a laser scanner. It was observed that the cleanest disk responses for both the 1FR and 1T modes occur at nodal diameter ND = 7 that is at 1029 Hz and 1402 Hz respectively. In Figure 8-b an example of the response detected by the laser scanner is shown. Considering this preliminary results, it was chosen to compare the BTT and SG measurements during rotation for the 1FR and 1T family modes at ND = 7.

6.2 Test campaign

According to Eq. 2 considering that the number of blades is N_b = 32, and assuming j = 2, it can be deduced that the ND = 7 can be excited by an engine order EO = 57. Moreover from Eq. 1 it can be derived that assuming i = 3 a number of equally spaced magnets m = 19 positioned in front of the rotating disk can produce an excitation force at EO = 57 in the disk. A test campaign was then performed with 19 permanent magnets (Figure 9) at a gap of 5 mm from the blade leading edges in order to produce in the disk a travelling excitation force with EO_{19}^{1} = 57 which in a given speed range excites the modes at ND = 7.

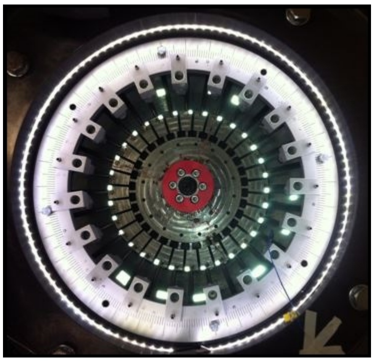

Figure 9: Spinning test rig with 19 permanent magnets.

The main tests parameters are listed in Table 5. Strain and displacement were acquired simultaneously for the 2 speed ranges (Table 5), including the values of n_{res} at which the resonances occur. For each test a linear speed sweep with an acceleration of 3.2 s/Hz was performed. The test campaign was repeated 3 times.

EO	*ND*	n-range
-	-	rpm
57	7	1000 - 1150
57	7	1350 - 1500

Table 5: Test campaign for the dummy disk 2.

6.3 The test results

The same procedure used for the previous disk was used to process the strain gauges signals. In the BTT data processing both the SDOF and CFF post processing methods were adopted. The displacement amplitudes in resonance obtained by the BTT were averaged over three acquisitions. Even for this disk the measured values showed high repeatability (standard deviation $\bar{\sigma} < 0.21\%$).

Tables 6 and 7 report the displacement amplitude values and the natural frequencies detected both by stran gauges (SG) and by the BTT system. In Table 6 the BTT data come from SDOF post-processing, while in Table 7 the BTT data come from the CFF post-processing method.
From the comparison of the e_f and e_u values (defined in Eq. 6) of Table 6 and Table 7 it can be noticed that they are both higher in Table 6 where the SDOF method is used than in Table 7 where the CFF method is used. It can be then be observed that with the dummy dusk 2 the acceptable method to post-process the BTT data is only the CFF where for the frequency $e_f < 2,5\%$ for the and vibration amplitudes $e_u < 5\%$. This differences between the two methods was explained considering the low vibration amplitude values of the dummy disk 2 which leads to measured points detected by the BTT probes more affected by the noise. The

Mode	*ND*	$\bar{f}_{SG}$	$\bar{f}_{BTT}$	e_f	$\bar{u}_{SG}$	$\bar{u}_{BTT}$	e_u
-	-	Hz	Hz	%	μm	μm	%
1FR	7	$1029,40$	$1144,00$	$11,13$	$136,43$	$173,30$	$27,02$
1T	7	$1402,20$	$1496,84$	$6,75$	$155,48$	$230,90$	$48,51$

Table 6: BTT-strain gauges comparison for the dummy disk 2 when the SDOF method is adopted.

Mode	*ND*	$\bar{f}_{SG}$	$\bar{f}_{BTT}$	e_f	$\bar{u}_{SG}$	$\bar{u}_{BTT}$	e_u
-	-	Hz	Hz	%	μm	μm	%
1FR	7	$1029,40$	$1022,75$	$0,65$	$136,43$	$131,37$	$3,70$
1T	7	$1402,20$	$1368,05$	$2,43$	$155,48$	$162,54$	$4,54$

Table 7: BTT-strain gauges comparison for the dummy disk 2 when the CFF method is adopted.

CFF method proved to be more robust for these noisy initial data since it uses the measurement coming from all the probes in one fitting procedure, while the SDOF method uses the data coming only from one probe.

7 CONCLUSIONS

A new measurement configuration for a BTT system usable with optical probes, the so called beam shutter or beam interrupted configuration, was tested by comparing the BTT results with the strain gauges results. The comparison was performed on two different dummy disks.

The beam shutter configuration proved to work properly since it gives the expected collection of measured data from the time of arrivals of each blade under each probe. These data has to be post processed to obtain frequency and amplitude in resonance condition.

The dummy disk 1 has a simple geometry and in resonance for the selected modes high blades displacement amplitudes (1600 - 2300 μm) were obtained. In this case the BTT system gave accurate values of frequency and amplitude in resonance using indifferently the two methots (SDOF or the CFF) for BTT data post-processing.

The dummy disk 2 is more similar to a real disk, it presents family of modes close to each others. Using a high EO value it was possible to excite two isolated modes at ND = 7 for two different modal families. The comparison between BTT and SG measurement was performed for these two modes. The amplitude of the displacements for both modes at the measured point was in this case smaller (130-150 μm) than in the previous disk. The low amplitude values leads to a set of data collected by the tip timing probes more noisy. In this case the final values of the dynamic parameters coming from the post-processing of the measured data proved to be more affected by the method selected for the post-processing. The CFF method which uses the signal of all probes in one fitting procedure, gave more accurate results.

8 ACKNOWLEDGEMENTS

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