DESIGN AND EXPERIMENTAL CHARACTERIZATION OF A NON INTRUSIVE MEASUREMENT SYSTEM OF ROTATING BLADE VIBRATION

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ABSTRACT

A measurement system for non-intrusive monitoring of rotating blade vibration in turbomachines based on fiber optic sensors is presented. The design of the whole system is discussed; the development of special purpose sensors, their interfacing to the data acquisition system and the signal processing are outlined. The processing algorithms are tested by software simulation for several possible blade vibrations. Experimental tests performed on different bladed rotors are presented. Results are compared to simultaneous strain gauge measurements.

NOMENCLATURE

- \( f \): vibration sampling frequency [Hz]
- \( f_b \): vibration frequency of blade [Hz]
- \( f_t \): transient recorder sampling freq. [Hz]
- \( s \): blade tip deflection [mm]
- \( S \): maximum blade tip deflection [mm]
- \( \Delta t \): transit time [s]
- \( v \): velocity [m/s]
- \( \omega \): rotating speed [rad/s]
- \( n \): revolutions per minute [rpm]
- \( \Phi \): phase of vibration [rad]
- \( \theta \): angular separation of sensors [rad]
- \( \lambda \): laser wavelength [nm]

SUBSCRIPTS

- \( b \): blade tip
- \( i \): general sample
- \( m \): measured
- \( R \): reference
- \( t \): transient recorder

INTRODUCTION TO THE MEASUREMENT TECHNIQUE.

Large interest is devoted since many years to non-intrusive techniques suitable for the measurement of rotor blade vibrations by non-contact sensors mounted statically on the casing of the turbomachine (McCarty and Thompson, 1978; Roth, 1981; Kiraly, 1981). They are very appealing because of their potential reliability and their non-intrusivity, due to the absence of sensing elements mounted on the rotating blades (Watkins, 1985; Paone et al., 1989; Stanoe, 1990). These techniques are all based on the measurement of the transit time of the blade tip below a set of sensing heads; time of flight variations \( \Delta t \) are due to blade deflection forward or backward. The accurate measurement of blade tip velocity and of the variations of arrival time allows one to compute one or two blade tip deflection samples

\[
S_i = \frac{v \Delta t}{2} \tag{11}
\]

when the blade passes in front of each sensor (Paone et al., 1990; Simmons et al., 1990; Paone et al., 1992; Kawashima et al., 1992). Several sensing heads along the outer circumference of the casing allow to measure different samples of the blade tip deflection. The angular displacement \( \Theta \) between each sensor head and the angular velocity \( \omega \) define the sampling frequency of the measurement system as:

\[
f = \frac{\omega}{\Theta} \tag{2}
\]

This means that the Shannon theorem, which states that sampling frequency \( f \) must be at least double than blade vibration frequency \( f_b \) can be satisfied by proper positioning of the sensors for given \( f_b \) and \( \omega \). The number of samples depends on the employed number of sensors. The uncertainty associated to each sample of blade deflection \( s_i \) affects the uncertainty in the computation of the maximum blade tip deflection \( S \) and of the vibration frequency \( f_b \) (Paone et al., 1991). The previous statement is true for every kind of vibration, either synchronous or asynchronous with rotating speed. This is an important feature, in order to measure vibrations during transient operation of the machine, when asynchronous vibrations are most important. The angular displacement \( \Theta \) can be different for each pair of sensing heads. In such a case, deflection samples \( s_i \) are non uniformly spaced. This basic method is needed to effectively measure deflections over a wide range of frequencies.

These kind of measurement techniques are accurate if the characteristics of the vibration, i.e. its amplitude and frequency, remain constant during the measurement time; accuracy decreases if these characteristics vary in time. In most cases it is
possible to estimate the vibration amplitude because the blade damping coefficient is so low that for many cycles it continues to vibrate; this allows to detect vibrations induced by events occurring before the start of the time window which do not repeat themselves at each revolution. The characteristics of typical vibrations were experimentally investigated by the same authors in 1990 and 1991; it was observed that the synchronous vibrations show constant characteristics in time, while during transients of the machine they change in time.

If just one sensor is used at each measurement point only bending vibrations can be detected and the sensor should be placed at the center of the blade chord, where torsional deformation is minimum. If more than one sensor is used at each measuring point, torsional vibrations can be detected as well.

THE MEASUREMENT SYSTEM.

A measurement system has been designed on the basis of the above principles and it is suited for measuring synchronous and asynchronous blade vibrations in the first stages of an industrial axial compressor, which operates on a 5 MW gas turbine power plant.

Fig. 1 shows a schematic drawing of the lay-out of the measurement system.

The system is based on four optical sensing heads which detect the passage of every blade, which were described by Paone and Rossi, 1992. Optical sensors have been chosen for two main reasons: their extremely higher bandwidth and their intrinsic insensitivity to electromagnetic interference. The need for a large bandwidth is due to the necessity to measure very small vibrations; in particular transitions of a transient of time of each blade, which is about 300 ms or less for a blade moving at a tangential velocity of 300 m/s and having a tip deflection 5=0.1 mm. The rise time of the pulse produced by the blade passage in front of the sensor should be significantly smaller than that; therefore these sensors have to be designed to have bandwidth in the order of 10 MHz, quite high in mechanical applications.

There are two sensing heads for each laser and photodetector. This device, which acts as a beam splitter for any optical wave travelling along the fiber link, either emitted from the lasers and going towards the sensing heads or reflected from the blade and returning backwards to the photodetector, reduces the number of electro-optical elements, realizing a distributed fiber-optic sensor. In this case four sensing heads require only two lasers and two photodetectors. Therefore the system gains in economy and simplicity. As a drawback, this kind of optical multiplexing reduces the total power available per unit head and degrades the useful power that reaches the photodetector: the number of two heads per sensor is a good compromise.

The lasers are 10 mW He-Ne gas lasers, operating in the visible range (λ=633 nm), which allows easier tuning of the system, with respect to invisible laser diode radiation, which is often employed in electro-optic sensors.

A multimode fiber is used in the system; being the overall length of the fibers not longer than 30 m. To improve its signal to noise performances, it is operated with a bias voltage close to the break-down value: this means that very low levels of reflected light produce a very strong signal, although not linearly related to the input. Being the technique based on a time measurement, the amplitude of the signal is important only in a binary sense, in order to indicate the presence or absence of the blade under the sensing head.

The photodetector is an avalanche photodiode, with a bandwidth of more than 100 Hz. To improve its signal to noise performances, it is operated with a bias voltage close to the break-down value: this means that very low levels of reflected light produce a very strong signal, although not linearly related to the input. Being the technique based on a time measurement, the amplitude of the signal is important only in a binary sense, in order to indicate the presence or absence of the blade under the sensing head.

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SIMULATION OF THE MEASUREMENT SYSTEM.

The processing software has been validated by using simulated signals. A proper simulator of the measurement system has been developed, which operates on a 5 MW gas turbine power plant. The simulator is equipped with four sensors, not uniformly spaced. The figure shows the simulated vibration amplitude of the blade tip as a function of the angular speed, the number of blades, the vibration amplitude, its frequency and phase, the sampling frequency and memory depth of the transient recorder. The simulator allows to simulate the behavior of the measurement system when excited by a given transient signal.

The acquisition system is based on a single channel transient recorder, operating at a maximum sampling rate of 200 MHz on a 2 Mbyte buffer, with 8 bit vertical resolution. This is interfaced to a personal computer for data processing.

A special software has been developed to drive the transient recorder and subsequently, in a preprocessing phase, to identify each pulse and to reduce all the samples to a set of pulse parameters. A second step of the analysis is performed on these parameters in order to measure the time shift of each pulse and to compute blade vibration deflection. The main output of the software is the time history of blade tip deflection. The analysis is performed on these parameters in order to measure the time shift of each pulse and to compute blade vibration deflection.

The following parameters can be varied: the dimensions of the rotor, its angular speed, the number of blades, the vibration amplitude, its frequency and phase, the sampling frequency and memory depth of the transient recorder.

Fig. 3 shows some results of the simulator, in which the working conditions that will be found experimentally on the test bench are simulated; in particular it is simulated a measurement system with four sensors, not uniformly spaced. The figure shows the simulated vibration amplitude as a function of the blade tip versus time, it is imposed a sinusoidal vibration of frequency f and amplitude S5. The samples obtained by the simulation of the measurement system are drawn on the same figure; four samples per rotor revolution are shown, because it is simulated a measurement system with four sensors. In the figure below it is represented the angle of rotation versus time.

Fig. 3-a and 3-b represent the results obtained with the rotor running at 2400 rpm (40 Hz) and the blade vibrating at f=395 Hz, i.e. asynchronous to the rotor, with amplitude S5=0.11 mm and two different initial phases Φ=0 and Φ=π/2. The sampling frequency of the simulated transient recorder is f=12.5 MHz, its memory depth 2 Mbyte. The samples clearly show that no synchronization exist between vibration and rotation, in fact at each revolution they occur at different positions along the sinusoid. Both the results in fig. 3-a and fig. 3-b allow to estimate a vibration amplitude S5=0.11 mm.

Fig. 3-c and 3-d represent the results obtained by the simulation of the rotor running at 4200 rpm (70 Hz) and the same blade vibrating at f=420 Hz, which is synchronous to the rotor, with amplitude S5=0.75 mm, again with two different initial phases Φ. The sampling frequency of the simulated transient recorder is f=25 MHz, its memory depth 2 Mbyte. The synchronization existing
between vibration and rotation appears from the samples measured by the system, in fact at each revolution they occur at the same positions along the sinusoid: the different phase in fig.3-c and 3-d depends on the initial phase of the vibration at the beginning of the acquisition time window. The results in fig. 3-c allow to estimate a vibration amplitude \( S_5 = 0.56 \) mm, those of fig. 3-d give \( S_5 = 0.73 \) mm. Both amplitude measurements are underestimated: this appears as a proximity sensor. They have allowed to observe that the sensor output is affected by the blade surface finishing. Roughness effects can be observed at the output as pulse amplitude distortion, which can reach the order of 30% in presence of rough surface. This implies a very careful triggering to measure the start and finish instants of each pulse. Also the focal spot diameter and the range of operating distance of the sensor for different focal lengths have been measured on the static bench. These measurements confirm that the sensor which has been developed is capable of detecting the blade presence for blades which are 2 to 6 mm distant from the sensors. These values make it suitable for most turbomachinery applications.

A second series of experimental tests have been performed on a small bladed rotor which provides a maximum blade tip tangential velocity of 30 m/s. The blades are not vibrating in this configuration. These tests have allowed to verify the correlation of the shape of each signal produced by one blade with its surface roughness. Fig. 4 represents the superposition of several pulses produced by one particular rotating blade. Each blade shows a characteristic signature in terms of shape of the pulses. This feature could be implemented in the software to enforce the signal processing, direct recognition of the pulses of each blade can be developed, further to the recognition by counting, which has already been implemented. Damages due to the accidental contact between blade tip and casing could also be detected.

**EXPERIMENTAL TESTS OF THE MEASUREMENT SYSTEM.**

A high speed test bench has been developed for further testing of the measurement system; its rotor can reach a maximum tangential speed larger than 300 m/s and it has vibration frequencies ranging from 200 to 2500 Hz. The blades installed on the rotor are instrumented with conventional strain gauges and slip rings for signal transmission, so that the actual vibration can be measured independently by eddy current probes on the steady bladed rotor. These results describe the operation of the measurement system which has been developed and confirm the good performance of the software and validate the processing algorithms that have been implemented. They also suggest the correct measurement procedure to be used. The stationarity of blade vibrations was confirmed to be sufficient for the operation of the measurement system by Paone et al. (1991).

The simulation of the measurement system by Paone et al. (1991) relates the strain gauge output signal to the actual blade deflection, which was measured independently by eddy current probes on the steady bladed rotor. These measurements have been performed in different conditions. Fig. 5-a, 5-b and 5-c show the output of the measurement system at 2400 rpm, 3000 rpm and 4200 rpm with air excitation. When the rotor runs at 2400 rpm and 2550 rpm the signals of the sensors are digitized at \( f_s = 12.5 \) MHz, while at 4300 rpm a transient recorder is operated at \( f_s = 25 \) MHz. Being its memory depth equal to 2 Mbytes, it is sufficient to digitize 6 revolutions of the rotor in all cases. The
three figures show the samples of blade tip deflection versus time; at each revolution 8 samples are measured by 4 optical sensors so that 48 deflection samples are available in each case, with exception of case 5-b where only two sensors were used. The sensors are not equally spaced along the circumference of the casing. The transient recorder is triggered by the signal of the sensors; no key phase sensor is used. Uncertainty in the measurement of each sample can be estimated, according to a methodology presented by the same authors in 1991, to be about 2%.

Fig. 5-b shows a series of samples which repeat themselves quite similar from one revolution to the other; this is the experimental result that is obtained in case of synchronous vibrations, as explained above by the use of the simulator software. Fig. 5-a and 5-c show instead two cases in which asynchronous vibrations are measured: the samples appear randomly distributed across the vertical scale. In these cases the angular velocity of the rotor was varying.

The measurements of fig. 5-a and 5-c are made while the vibration was monitored also by strain gauges. Aside of each figure are reported the values of twice the vibration amplitude, measured independently by the strain gauges and by the non intrusive measurement system and indicated respectively as $2S_5$ and as $2S_6$. $2S_5$ is measured by FFT analysis of the strain gauge signals, while $2S_6$ is computed as the difference among maximum and minimum amplitude. The results confirm what observed by using the simulator; i.e. the vibration amplitude is underestimated, due to the statistical reasons discussed previously. At 2400 rpm the fiber optic measurement system measures $2S_5=0.08$ mm against $2S_6=0.11$ mm measured by the strain gauges, while at 4300 rpm it results $2S_5=0.42$ mm against $2S_6=0.75$ mm. The difference is the same order of magnitude as the predictions of the simulator. As explained above, the increase in the number of samples leads to a population which is statistically meaningful and whose analysis allows a more accurate measurement of vibration amplitude.

Many hours of extensive testing in an industrial environment have proved the reliability of the sensing elements and of the whole measurement chain. In fact the dust and the oil particles which were present in the test rig did not significantly affect the signal to noise ratio of the sensor. Furthermore the intense electromagnetic fields, caused by the 500 hp electrical engine which drives the rotor did not affect the performance of the system. Air cooling of the fiber optic head is under development and air is also used to clean the front lens. The sensors will then be suited for industrial applications on operating turbomachines.

CONCLUSIONS

A measurement system for detecting synchronous and asynchronous rotating blade vibrations based on static fiber optic sensors has been realized and tested. Its performance has been characterized by software simulation. Its different performance in case of synchronous and asynchronous vibrations is put in light; in all cases, if the number of samples of vibration is little, the vibration amplitude is underestimated. The means to increase the number of samples by expansion of the memory of the transient recorder and by acquisition of successive time windows have been proposed. The experiments performed on a test rig in conditions very close to those of an industrial axial compressor, have confirmed the capabilities of the measurement system to detect blade vibrations. Comparison with strain gauge measurements shows the underestimation of vibration amplitude that was investigated by the simulator. Therefore the conclusions derived by using the simulator can be implemented in the measurement system in order to improve its accuracy. The system has also proved to be reliable when operating in difficult environmental conditions. Further work is necessary to verify its performance on a real machine and to implement refined algorithms for data processing.

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200 Msd/s data acquisition system

Fig.1: Components of the measurement system

Fig.2: Software block diagram
3-a) \( n = 2400 \text{ rpm} \)
\( f = 395 \text{ Hz} \)
\( S_p = 0.11 \text{ mm} \)
\( \Phi = 0 \)
\( S_m = 0.11 \text{ mm} \)

3-b) \( n = 2400 \text{ rpm} \)
\( f = 395 \text{ Hz} \)
\( S_p = 0.11 \text{ mm} \)
\( \Phi = \pi/2 \)
\( S_m = 0.11 \text{ mm} \)

rotation angle \([\text{deg}]\)

3-c) \( n = 4200 \text{ rpm} \)
\( f = 420 \text{ Hz} \)
\( S_p = 0.75 \text{ mm} \)
\( \Phi = 0 \)
\( S_m = 0.73 \text{ mm} \)

3-d) \( n = 4200 \text{ rpm} \)
\( f = 420 \text{ Hz} \)
\( S_p = 0.75 \text{ mm} \)
\( \Phi = \pi/2 \)
\( S_m = 0.56 \text{ mm} \)

rotation angle \([\text{deg}]\)

Fig. 3: results from the simulator of the measurement system.
Fig. 4: Typical signals of the sensors produced by one particular blade passing in front of one sensing head at different successive revolutions

a) n=2400 rpm
   $2S_m=0.08$ mm
   $2S_r=0.11$ mm

b) n=2550 rpm
   $2S_m=0.38$ mm
   (only two sensors have been used and no strain gauge)

c) n=4200 rpm
   $2S_m=0.42$ mm
   $2S_r=0.75$ mm

Fig. 5: Blade tip displacement samples measured by the system.