Non-Intrusive Measurement of Blade Tip Vibration in Turbomachines

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ABSTRACT

It is discussed the need for non-intrusive monitoring of rotating blade vibration. The advantages over conventional techniques are outlined.

The vibration of a compressor blade has been investigated by strain gages, with particular attention to time evolution of the frequency spectra, at different operating speeds of the machine. These information are fundamental to the correct design of the non-intrusive measurement system.

The non-intrusive measurement technique has been presented, with special attention on uncertainty and resolution. The necessary hardware is discussed and the design of an optical sensor is presented.

NOMENCLATURE

- $E_i$: uncertainty due to the irregularity
- $E_{IS}$: uncertainty on $S$ due to $i$
- $E_{IV}$: uncertainty on $v$ due to $i$
- $f_c$: sampling frequency [MHz]
- $f_{CS}$: necessary sampling freq. to detect $S$ [MHz]
- $f_{CV}$: necessary sampling freq. to detect $V$ [MHz]
- $i$: irregularity of angular speed
- $R$: rotor radius [mm]
- $R_S$: resolution in deflection measurement
- $R_V$: resolution in velocity measurement
- $s$: blade tip deflection [mm]
- $S$: maximum blade tip deflection [mm]
- $t$: time [s]
- $T$: period of vibration [s]
- $V$: instantaneous blade tip velocity [m/s]
- $V_I$: vibratory blade tip velocity [m/s]
- $\delta$: circumferential distance [mm]
- $\Delta T_{AC}$: time interval between A and C [s]
- $\Delta T_{AC}$: mean time interval [s]
- $\Delta T_{AC}$: variation of time interval [s]
- $v$: vibratory frequency [Hz]
- $\omega$: rotating speed [rpm]
- $\omega_R$: peripheral velocity [m/s]

INTRODUCTION

Monitoring and diagnostics of rotor blade vibration are important to effectively improve performance and reliability of turbomachines. Among the measurement techniques developed for this purpose, increased attention is devoted to non-intrusive techniques, which measure the blade tip vibration by observing the blades when they pass in front of stationary sensors. A relatively vast literature already exists on this subject, proving the interest in these measurement techniques and showing the difficulties that practical applications present [1-8].

When considering the instrumentation of a machine, it should be clear if the need for measurements is limited to the research and development phase or if it is desirable to monitor the working conditions of that machine throughout its life. Both approaches aim to increase the reliability of the machine. The first approach is intended to provide information and feedback for a reliable design of the machine, while the second approach is intended to improve reliability of the machine by monitoring operating conditions and generating warnings to prevent failures or reduce their damage. Modern mechanical systems are becoming more complex and are designed for higher levels of performance. To reduce maintenance and operating costs it is necessary to increase reliability and availability of the machine. In mechanical systems, where reliability cannot be achieved by redundancy, the monitoring of machines appears the most natural choice.

Monitoring rotor blade vibration involves sensing blades rotating at very high speed while vibrating in various modes often coupled together and in a quite harsh environment. When sensors are used to increase the reliability of a mechanical system, the sensors
themselves must be highly reliable [9], otherwise their effect can be worse than without monitoring. In fact, even if the machine still works properly when a sensor is faulty, it will be necessary to stop its operation to repair the sensor, thus decreasing the availability of the whole machine.

Non-intrusive sensing techniques for monitoring blade vibration use stationary sensors mounted external to the rotating parts of the machine. They are more reliable than conventional intrusive sensing, therefore their development is of high interest.

These considerations suggest that non-intrusive methods are advantageous over intrusive sensing, but it is not clear that their performance are comparable. The "ideal goal" would be to measure amplitude and frequency of any vibrational mode of any blade of the rotor at any rotor speed; the "achievable goal" is obviously limited to some of these information. The drawback of these measurement techniques is that they rely on the validity of some modeling of the vibrations assumed as the base for the necessary computations. The following is an explanation of the relationships between the physics of rotor blade vibrations, their modelling and the measurement technique. Any assumption or approximation made in the model affects the accuracy and the meaning of the results of the measurement.

A FEW WORDS ABOUT BLADE VIBRATIONS

Rotor blades are subject to flexural and/or torsional vibrations. Several modes can be excited together, depending on the aerodynamic forces acting on the blade surfaces and on the mechanical stresses transmitted by the rotor to the blades. Two main classes of blade vibrations can be identified, namely vibrations whose frequency is an integer multiple of rotational speed and vibrations whose frequency is not a multiple of the rotor speed.

To a stationary observer these two classes appear to behave very differently. A stationary observer sitting on the external casing of the machine, looks at the blade tips passing in front of him; he is able to measure time intervals, so he can also compute blade instantaneous or mean velocity. If the blades are not vibrating, their passing frequency must match the rotating speed and, at steady angular speed, the stationary observer always measures the same inter-arrival time. If the blades are vibrating at a frequency which is not a multiple of the rotating speed, the time of arrival and the instantaneous blade tip velocity will change in time, being the blade deformed, so that it arrives sooner or later. The observer can thus measure these time fluctuations and, from them, once a model of the vibration is assumed, he can theoretically compute frequencies and amplitudes. If the vibrations have a frequency multiple of the rotating speed, no difference in time of arrival will be detected; each blade will pass in front of the observer in the same deformed configuration. The measurement of vibration is still possible if more than one observer sit along the casing; they will in fact see differences between their time measurements and, within the framework of a model of the vibrations, it will be possible to compute their characteristics.

All that has been said is valid, if the characteristics of the vibration, i.e. amplitude and frequency, do not change in time, and therefore the vibration is coherent. If coherence is lost, the blade time of arrival will change from one revolution to the next one in a way that cannot be only related to the vibratory motion. In this case it is impossible to determine the frequency, while it may still be possible to measure the maximum amplitude of the vibration, which, in general, is related to blade stress, the quantity of primary interest.

This description of blade vibrations explains why the model of vibration used for the computations is fundamental to the results; vibrations of one kind cannot be absolutely measured by these non-intrusive techniques if it is assumed to be of the other kind!

AN EXPERIMENTAL INVESTIGATION OF BLADE VIBRATION

Not many data are available for blade vibration characteristics that are helpful in evaluating the practical application of non-intrusive techniques, such as the ones discussed here.

In order to develop a prototype system for measuring blade vibrations in a compressor, an experimental investigation by conventional strain-gage techniques has been done. The machine under study is the compressor of a 10 MW industrial axial gas turbine, rotating at 10600 rpm. Strain gages are mounted on some blades. Signals are recorded via slip-rings; time records of blade vibrations at different rotor speeds have been Fourier transformed to have information on the frequency content of the vibrations. The aim is to determine if vibrations characteristics are stationary in time, i.e. if their amplitude and frequency remain the same during many rotor revolutions, and also to see if their frequency is an integer multiple of rotor angular speed. All these characteristics strongly affect the implementation and the performance of the technique.

Results show two main classes of behaviour of a vibrating and rotating blade.

Figs.1-a) and 1-b) show typical spectra of the vibration measured respectively during transient operations (5900 rpm) and at full operating speed (10600 rpm). They appear very different; the frequency content is limited to only one harmonic at operating speed, but much more rich in harmonic content during transient operation. The figures show that frequencies lie on values that are an integer multiple of the rotational speed, indicated by dotted vertical lines; the spectra at transient operation show also some energy coupled to non-integral frequencies. Fig.2 shows the time evolution of the amplitude and frequency of the first harmonic in both cases. Amplitude and frequencies have been measured during 10.24 s, which correspond to 1800 rotor revolutions and to 1000 respectively at 10600 rpm and at 5900 rpm. The results show that at operating speed very little amplitude and frequency fluctuations are present; the vibration remains highly coherent during many revolutions. Fluctuations are present during transient operation, mainly in amplitude. At 5900 rpm amplitude fluctuations equal to ±30% and frequency fluctuations of ±0.5% have been measured. Figs. 3-a) and 3-b) show the time evolutions of the spectra of vibration in the same conditions as above. Results clearly show the differences of frequency content and of stability in time between the two different regimes. These, together with other tests, have shown that the behaviour found at steady operating speed extends down to speeds about 80% of full speed.
A blade rotating on its rotor can be considered as a cantilever beam which rotates while subject to flexural or torsional vibrations; for both kinds of vibrations a point of the blade tip is displaced forward or backward, with respect to the sense of motion, as in fig.4. The tip displacement $s(t)$ can be expressed by:

$$s(t) = S f(t)$$

(1)

where $f(t)$ is a periodic function of period $T$ and fundamental frequency $f_0=1/T$ and $S$ is the blade tip vibration amplitude; both $S$ and $f_0$ are of interest.

The measurement of $s(t)$ is accomplished by time interval; the time interval which carries information on the blade tip deflection is dependent on the kind of vibrations under exam, either multiple or non-multiple of the frequency of rotation.

Fig.4 refers to the case of non-multiple vibrations; the time interval to be considered is $\Delta t_{AC}$ between a trigger signal produced by an element which rotates, but does not vibrate, and a second pulse produced by the blade tip. This interval would be constant if the blade would not vibrate, while it is variable if the blade tip vibrates. To compute blade tip deflection it is necessary to measure the variations of $\Delta t_{AC}$ with respect to its mean value:

$$\Delta t_{AC} = \Delta t_{AC} - \Delta \Delta t_{AC}$$

(2)

It is also necessary to measure the instantaneous blade tip velocity:

$$v' = \frac{\delta_{CD}}{\Delta t_{CD}}$$

(3)

which requires the measurement of the time interval $\Delta t_{CD}$ during which the blade tip, having thickness $\delta_{CD}$, passes in front of the stationary sensor. Once $\Delta t_{AC}$ and $v'$ are known, it is possible to compute the blade tip deflection $s_1$ at each passage of the blade under the measurement station; it is:

$$s_1 = \frac{\delta_{CD}}{\Delta t_{CD}} - \Delta \Delta t_{AC}$$

(4)

It is also possible to measure the blade tip velocity due to the vibration, by subtracting from the instantaneous velocity the tangential velocity due to the rotation:

$$v_1 = \frac{\delta_{CD}}{\Delta t_{CD}} - \frac{\delta_{AB}}{\Delta t_{AB}}$$

(5)

This implies the assumption of steady rotational velocity during the "sample interval" $\Delta t_{AC}$. The validity of this assumption depends on the constancy of the rotational velocity, which is generally very stable.

Thus, it is possible to determine a sample of blade tip deflection $s_1$ and vibratory velocity $v_1$ at each passage of the blade in front of the measurement station, once a trigger signal for each blade is available and a stationary sensor is installed on the casing. The trigger signal can be produced by an encoder mounted on the shaft, with as many teeth as the number of blades. In case this is not available, the signal of the previous blade can be used. In this case the signal processing is more complex, because the variations of the time interval $\Delta t_{AC}$ are due to simultaneous deflections of the two adjacent blades. Once a sequence of data $s_1$ and $v_1$ is acquired, it is possible to compute some characteristics of the vibration. For example, maximum amplitude is computed by:

$$S = \text{MAX}(s_1)$$

(6)

Under the assumption that no phase changes occur in the vibration while the machine is rotating, it is possible to reconstruct its harmonic content, knowing many samples of deflection $s_1$ and its derivative $v_1$ at given instants $t_i$. It should be noted that, since the frequency of the vibration generally higher than the frequency of rotation, it is not a simple task to reconstruct the signal, because if one measurement station is used only aliased samples are available.

When the fundamental frequency of the vibration is an integer multiple of the rotating speed, the measurement technique must be implemented in a different fashion, because the blades will always pass in the same deformed configuration at the measuring point.

The scheme of fig.4 must be modified into that of fig.5, which has more than one sensor at the periphery of the rotor. This allows the measurement of the deflection of the blade tip again by time interval measurements. If the blade is not vibrating the time interval between two sensors would be:

$$\Delta t_{AC} = \frac{\delta_{AC}}{(\delta_{AB}/\Delta t_{AB})}$$

(7)

where the distances $\delta_{AC}$ and $\delta_{AB}$ are known, while the time interval $\Delta t_{AC}$ is measured. Because of vibration, the time of flight from A to C will actually be different, equal to $\Delta t_{AC}$; the difference is a measurement of blade deflection at point C, with respect to point A:

$$s_i = \frac{\delta_{AB}}{\Delta t_{AB}} (\Delta t_{AC} - \Delta t_{AC})$$

(8)

$\delta_{AB}/\Delta t_{AB}$ is a measurement of the instantaneous rotor tangential velocity. In the same way as above it is possible to measure the instantaneous blade vibration velocity. This is valid as long as the rotational velocity does not fluctuate during $\Delta t_{AC}$.

If more than two sensors are used, it is possible to have more than one sample of the vibration $s(t)$, this time not aliased, because the measurement points are close to each other, and in any case their distance is fixed to be shorter than the wavelength of the vibration. This makes it possible to compute the vibration characteristics, by a dedicated digital data processing (11). The same configuration can also be used for vibrations with frequencies that are multiples of rotational speed.

UNCERTAINTY AND RESOLUTION OF THE MEASUREMENT TECHNIQUE

The measurement technique relies on time interval measurements; the hypothesis and assumptions taken in the modeling of the vibration, which is at the base of the data processing, have influence on the uncertainty of the method, on its resolution and, in some cases, limit its applicability (5), (7).

Flutter, one of the most dangerous kind of vibration, shows the coupling of torsional and flexural deformations. Our treatment is based on the implicit assumption that flexural and torsional modes are not present together, or one kind has much larger amplitude than the other; their simultaneous presence needs the use of more than one line of sensors, along the blade...
span and renders the technique more complex, but still feasible, although it has not been treated here.

The main source of uncertainty is caused by random phase fluctuations of the vibration; if they occur, it will be impossible to measure the frequency content of the vibration by these techniques. Only maximum amplitudes can be measured.

If the maximum amplitude of the vibration fluctuates in time it is again possible to measure only its peak value, while the rest of the information of the vibration is lost.

During the time interval $\Delta t_{AC}$ it has been assumed that the rotational speed of the rotor remains constant; if it has an irregularity

$$i = \frac{(T_{MAX} - T_{MIN})}{T_{AVERAGE}} \tag{9}$$

some errors arise, because $\Delta t_{AC}$ varies due to both vibrations and irregularities. These uncertainties can be estimated if the vibration $s(t)$ is known; let us assume a simple sinusoidal law, of frequency $V$, with the rotor of radius $R$ rotating at an angular speed $\omega$.

$$s(t) = S \sin(2\pi V t)$$ \tag{10}

An estimate of the variations induced on $\Delta t_{AC}$ by $i$, can be expressed by:

$$E_i = i \Delta t_{AC} \tag{11}$$

it generally produces overestimated errors, to be considered as upper bounds. This uncertainty has to be related to the time interval fluctuations that should be measured, i.e. $\Delta t_{AC}$ for the estimate of the deflection $s$ and $\Delta t_{CD}$ for the estimate of the vibrational velocity $v$. It results in uncertainties equal to:

$$E_s = i \Delta t_{AC} / \Delta t_{AC} = i \Delta t_{AC} / S \tag{12}$$

and

$$E_v = \frac{1}{2} \Delta t_{AC} / \Delta t_{CD} = i \left(\frac{2\pi R^2 - 4\pi v S}{2\pi v W S}\right) \tag{13}$$

Table 1 shows some results about the maximum errors for given parameters of the machine and of the vibration; to be noted is the lower accuracy of vibration velocity measurements, with respect to blade tip deflection. This implies that the estimate of frequencies is inherently less accurate than that of amplitudes.

Non intrusive techniques are based on the measurement of small fluctuations of time intervals, namely $\Delta t_{AC}$ for the evaluation of deflections $s_i$ and $\Delta t_{CD}$ for the determination of $v_i$. To obtain a resolution $(R_S=\Delta s/S)$, in the measurement of blade tip deflection, it is necessary to measure time with a clock period equal to:

$$t_{RS} = R_S \left(\Delta t_{AC}\right) = R_S \left(S/\omega R\right) \tag{14}$$

In the same way it can be computed the clock period necessary to measure velocity with a resolution $R_v$:

$$t_{RV} = R_v \left(\Delta t_{CD}\right) = \frac{2\pi SV C D R_v}{\omega (2\pi R^2 - 4\pi v S)} \tag{15}$$

Table 1 presents some results about the necessary clock frequency $f_C$ for a given machine and given vibration. Results show that, in order to measure correctly velocity fluctuations, it is necessary to use a very high clock frequency. Much less demanding is the measurement of blade tip deflection. High clock frequencies imply also larger memories if signals are stored before processing; at present these are the main limits to the applications of these techniques to turbomachines.

The measurement of vibration whose frequency is an integer multiple of the rotational speed need the use of more than one sensor and the measurement of longer time intervals $\Delta t_{AC}$. This results in even larger memories than for the case of vibration whose frequency is not an integral multiple of rotational frequency.

A MEASUREMENT SYSTEM

The requirements for high resolution in time measurements also determine the characteristics of the measurement system. In particular its bandwidth must be as high as possible, because time interval measurements are more accurate if squared pulses are used.

Signals are produced by the blade tip passing in front of a sensor which is mounted stationary on the casing. Typical duration of the passage of a blade in front of a sensor is very short, for example 10 µs in the case of a 3 mm blade moving at 300 m/s. To respond to such an event with a squared signal, only optical sensors are well suited.

We propose a fiber-optic sensor designed for this purpose [10]; its scheme is represented in fig.6. It is a reflection type sensor: laser light from a 5 mW He-Ne laser is focused on a small spot on the blade tip and reflected light is collected and carried to a fast avalanche photodiode. The design is intended to satisfy the requirement for a high bandwidth and to produce a compact and robust probe, which could be used in an environment with large temperature fluctuations and strong vibrations without loss of alignment and thus efficiency. The fiber link is composed of multimode fibers 50/125 µm: a 10/90 fiber directional coupler is used in the system, so that only one fiber arrives at the probe end. Light is coupled into the fiber by graded index lenses: the same kind of lens is used at the probe end, where it focuses light at 3 mm from its flat surface. This results in an extremely rugged probe, with the lens and the fiber glued together inside the fiber collimator, to form a solid unique piece. Probe external diameters smaller than 10 mm can be realized. The bandwidth of the fiber link is extremely high with respect to the frequencies involved in this problem; the bandwidth of the photodiode is above 1 GHz, more than sufficient for this purpose. Analysis shows that the main limit to frequency response of the sensor is due to the interaction between the light spot and the blade tip. Having the focal spot diameter $d=3 \mu m$, when the blade tip enters the illuminated area at $v=300 \text{ m/s}$ it takes a time $t=0.01 \mu s$ to enter it completely: this is a limit to the rise time of the pulsed signal that cannot be overcome and will be present to some extent in any kind of optical probe. This limit, together with the fact that the edges of the blades are not very sharp, implies working with signals with lower slope, thus reducing accuracy of time interval measurements.

Time measurements can be performed either on the digitized signals by digital signal processing or on the analog signal by counting techniques.

Digital signal processing implies high sampling frequency and large memories. Commercial transient recorders have lower memory size for increasing sampling.

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frequency. The proceeding analysis has put in evidence the need for much higher frequencies to accurately measure velocities than those needed to measure displacements. Commercially available transient recorders are suited to measure displacement, but are still limited with respect to what is needed to measure velocities. It should be noted that memory segmentation techniques can help in data storage with memory saving. Furthermore, if vibration maintains the same characteristics in time, it is possible to download stored data to disk memories and restart data acquisition without loss of information. Another advantage of the transient recorders over counters is the possibility of using the pulses to measure blade clearances, by analysing the intensity of light reflected by the passage of each blade.

Counting techniques do not present problems with frequency, but do not register the signals. It means that a complex and reliable logic for the trigger system is necessary to measure the desired time intervals [2]; little margin for external control is left to the operator.

CONCLUSIONS

What has been presented above has covered different aspects of the application of non-intrusive measurement techniques for turbomachinery blade vibration: the object of the measurements (i.e. the vibrations), the theory of the technique, the characteristics of a measurement system.

The experimental investigation of some compressor blades operating at different rotor speed has put in evidence different possible kinds of vibration that should be measurable by non-intrusive techniques. Stationary vibration characteristics with frequencies that are multiples of the rotational speed have been observed at velocities above 80% of the regime: their power spectrum is made of one frequency, highly stable in time. Different and more complex vibrations arise during transient regimes of the compressor.

The theoretical presentation describes the basics of the method and the different kinds of vibrations that can be measured, and shows the performance of the measurement technique in terms of uncertainty and resolution of the results. The mathematical relationships that have been presented are a useful guide in the design of a measurement system based on these techniques. They compare the characteristics of the system to be implemented, with respect to the characteristics of the vibrations to be measured and of the machine inside which they occur.

The same discussion focuses on the main requirements for the design of the measurement system. Since the sensor is a particularly critical element in the system, a design developed for this application has been presented, which shows a rugged construction coupled with high performances characteristics.

This work will proceed with the implementation of the technique on an industrial machine, based on the principles and on the discussion presented above.

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BIBLIOGRAPHIC REFERENCES


fig. 1 - Adimensional power spectrum of blade vibration (linear scale):
   a) at 55% of maximum speed, b) at maximum speed (10600 rpm).

fig. 2 - Adimensional plot of amplitude and frequency versus time
   at partial speed and at full speed.

fig. 3 - Time evolution of power spectrum of blade vibration:
   a) at 5900 rpm, b) at 10600 rpm.
fig. 4 - Scheme with one reference sensor and one sensor on the periphery; only non-integral vibrations are measurable.

fig. 5 - Scheme with more than one sensor on the casing; both integral and non-integral vibrations are measurable.

Table 1 - Parametric study of uncertainty and time resolution.

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<td>13.0</td>
<td>20.9</td>
</tr>
</tbody>
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