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Detection of blade rub in jet engine turbines using strain gages

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Abstract
The operating environment of jet turbines is subject to high fluid temperature which often results in blade dilation and, consequently, a rubbing phenomenon between the blades and the stator. This paper proposes a condition monitoring strategy based on strain measurements. A mathematical model is first proposed to describe the response of the rub signal. This model is then used to de-noise the signal from interfering noises and eventually construct relevant condition indicators for each of the turbine blades rubs. The proposed technique is validated on real signals captured from a strain gauge mounted on the stator of a turbomachine.

1. Introduction

Turbomachines aim at transforming the kinetic energy of a fluid into mechanical energy (and vice versa) via a rotary assembly called a rotor. They have different designs according to their functions: turbines, pumps, compressors, turbochargers, etc. Nevertheless, most of them have a common architecture consisting of a rotor (a wheel mounted on a shaft) which is connected to a fixed housing (the stator) via a bearing. To establish a relative rotational movement between the stator and the rotor, the rotor is necessarily spaced from the stator and this spacing is commonly called the blade clearance of the turbomachine. Blade clearance is an important parameter of design and operation of a turbomachine. Its minimization prevents excessive amounts of fluid to bypass the rotating row of blades and, therefore, improves the general efficiency of the turbomachine. However, this clearance is subjected to fluctuations due to thermal expansions and mechanical phenomena dependent on the operating regime of the turbomachine as well as other environmental conditions (e.g. temperature) [1]. Today, many jet turbines feature an active clearance control system to improve fuel efficiency while dynamically controlling the clearance. In this case, clearance sensors are deployed to measure the clearance. A common strategy of such a system consists of a valve that mixes hot and cold air from the compressor outlet and the bypass duct, respectively, at a desired temperature. Air is routed to flow through tubes surrounding the housing (stator) at each stage of the turbine. This air expands or contracts the turbine housing to provide the desired JSA. Note that the previously mentioned valve is automatically adjusted by the FADEC according to the position of the thrust lever using the current clearance measured by the clearance sensor. However, eccentricities in the clearance often cause the sensor to measure a positive clearance number, even in the presence of friction. Thus, for a system in which an active slack control strategy is used, this would result in the control taking the clearance to smaller values, even in the case where a rub exists and friction occurs is produced.

Therefore, minimal clearance often induces contacts between the blade tip and the stator that leads to blade rubbing. Though being inevitable, the rubbing phenomenon decreases the engine performance and can lead to blades or/and stator damages [2]. Thus, detecting and identifying the instants of contacts between the blades and the stator is precious and offers at least two advantages. The first one is to get information about the existing wear of the turbomachine which can help to make a preventive maintenance scheme. The second one is to provide the clearance control system with the contacts information. In practice, contact can be detected using stator vibrations, acoustic signals and/or dynamic pressure [3] [4]. In this paper, it is proposed to measure the deformation (or the displacement) using strain gauges mounted on the stator. As for any sensor, it was found that the signal is highly corrupted by other interference and background noise, thus harden the detection process. This paper addresses this issue through a condition monitoring strategy based on strain measurements. This strategy is based on the fact that the rubbing phenomenon creates a wave shock that deform the structure, these latter can be efficiently measured by a strain gauge mounted on the stator. However, strain signals are generally masked by strong mechanical and electromagnetic interferences. This

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paper proposes a signal separation technique based on the cyclic and synchronous properties of the rotor component. Also, condition indicators are proposed to quantify the rubbing phenomenon of each blade. The proposed methodology is validated on real signals captured from a strain gauge mounted on the stator of a turbomachine. The proposed strategy is tested on real strain signal measured on a real turbomachine in a test experiment.

2. Problem statement

2.1 Blade rub phenomenon and measurement approach

The most natural way to detect the occurrence of rubbing is to identify the impacts generated by this phenomenon. This can be achieved through a vibration measurement captured by accelerometers and strain gauges. The major advantage of strain gauges over accelerometers is their capacity to provide an image of the strain (through the Young Modulus) which is valuable for diagnostic, component fatigue assessment and prognostic. Another desired feature of the diagnosis system is its ability to distinguish the exact source of rub i.e. the blade subjected to rub. Therefore, we use an encoder (such an optical encoder) to measure the passage of the blades. On the one hand, this measure will serve to transform the signal to the angular domain through angular resampling to compensate slight speed fluctuations in the rotating period. On the other hand, the encoder signal helps to distinguish the different blades of the turbomachine.

2.2 Mathematical modelling

The rubbing phenomenon can be seen as a brief impulse excitation generated by the contact between the tip of the blade and the surface of the stator. Consider a rotor with R blades, the stress signal generated by the r^{th} blade can be modelled as a train of transients where impulses are equally spaced by the rotating period of the rotor:

$$\forall n \in \{1, \dots, L\} \quad \begin{aligned} D_r[n] &= \left(\sum_{q=1}^{Q} a_{qr}\delta[n - q.N - \tau_r]\right) \otimes h_r[n] \\ &= \sum_{q=1}^{Q} a_{qr} h_r[n - q.N - \tau_r] \end{aligned} \tag{1}$$

where

- $\otimes$ refers the convolution operator
- $\delta[n]$ is the Kronecker function
- $h_r[n]$ is the impulse response of the r^{th} blade
- a_{qr} is the amplitude of the contact force (the excitation) generated by the r^{th} blade at the q^{th} cycle
- $\tau_r = (r - 0.5)/R$ is the phase shift in number of samples between two consecutive blades
- L is the number of samples.
- and Q is the total number of cycles

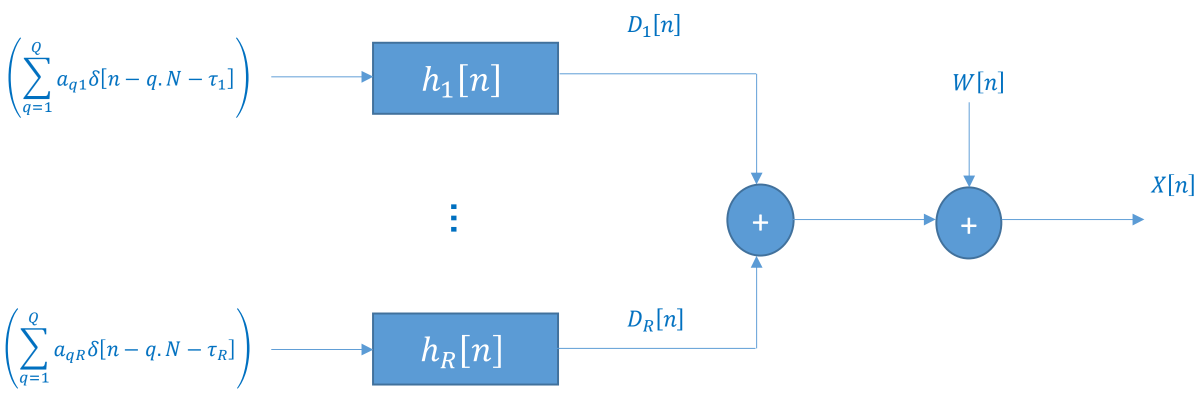

Figure 1: The mathematical model of the blade rubbing displacement signal.

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It should be noted that a_{qr} generally varies during the cycles because it depends on several factors such as the depth of the rub, the speed of rotation, the stiffness of the surfaces which can change with abrasion, etc. The latter can be considered null in the case of non-contact. Since the correlation length of the impulse response is generally much less than the cycle period N, Eq. (1) can be rewritten as:

$$\forall n \in \{1, \dots, L\} \qquad D_r[n] = a_r[n]c_r[n] \tag{2}$$

where $c_p[n] = \sum_{q=1}^{Q} h_r[n - q.N - \tau_r]$ is a periodic function of period N and $A_r[n]$ is a smooth function such as $a_r[n - q.N - \tau_r] = a_{qr}$. The mechanical response of the rotor (i.e. comprising the R blades) is written as follows:

$$\forall n \in \{1, \dots, L\} \qquad D[n] = \sum_{r=1}^{R} D_r[n] = \sum_{r=1}^{R} a_r[n]c_r[n] \tag{3}$$

It should be noted that physically, $a_r[n]$ designates the strength of the excitation of the r^{th} blade blade while $c_r[n]$ designates the periodic pattern produced by the response of this blade (therefore the stress waveform). In general, the measured stress signal contains noise from other mechanical or electromagnetic sources, these are considered independent and will be denoted by $W[n]$. Consequently, the measured stress signal is written as follows:

$$\forall n \in \{1, \dots, L\} \qquad X[n] = D[n] + W[n] \tag{4}$$

3. Proposed methodology

3.1 Monitoring strategy

The health monitoring strategy adopted in this paper is based on the processing of strain signal to extract diagnostic information related to a potential run between the rotating part of the jet engine and its stator. At first, a pre-processing step is needed to de-noise the signal from environmental noise. These noises may come from other rotating sources, electrical or/and aerodynamic sources. We use a parametric approach based on the mathematical model proposed in subsection 2.2 to estimate the strain component related to the rotating part of the turbine. This component will be ideally noise-free and contains only the source of interest. This process is described in subsection 3.2. After estimating the strain component of the rotor, the strain signal associated with each of the turbine blades is extracted separately in order to assess the rubbing phenomenon for each blade. This separation is described in subsection 3.3. Lastly, condition indicators that represent the magnitude of the rub of each blade are extracted with a statistical threshold for decision making. This will be addressed in subsections 3.4 and 3.5 respectively.

3.2 Estimation of rotor contribution

Since the excitation force of a blade does not interfere in the response of another blade, we have $a_{r_1}[n]c_{r_2}[n] = 0$ for $r_1 \neq r_2$. Considering this independence, Equation (3) is rewritten as follows:

$$\forall n \in \{1, \dots, L\} \qquad D[n] = \sum_{r=1}^{R} a_r[n] \sum_{r=1}^{R} c_r[n] = A[n]C[n] \tag{5}$$

where $A[n] = \sum_{r=1}^{R} a_r[n]$ denotes the sum of the forces exerted by all the blades being periodic of period N and $C[n] = \sum_{r=1}^{R} c_r[n]$ is the sum of R smooth functions. According to the "Weirstrass theorem", it could be approximated by a polynomial of order P

$$\forall n \in \{1, \dots, L\} \qquad A[n] = \sum_{p=1}^{P} C_p\, n^p \tag{6}$$

where C_p are constant coefficients. Equation (4) is rewritten as follows:

$$\forall n \in \{1, \dots, L\} \qquad D[n] = \sum_{p=1}^{P} C_p[n]\, n^p \tag{7}$$

where $C_p[n] = C_p A[n]$ is a periodic function of period N. According to equation (6), the stress signal generated by a rotor is modelled as a polynomial of order P with periodic coefficients having the same period of rotation as the rotor.

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Let us denote by $\bar{n} = \lfloor (n-1)/N \rfloor + 1$ the position of the sample in period N ($\lfloor a/b \rfloor$ denotes the remainder of the division of a by b). Because $C_p[n]$ is periodic of period N, we have $C_p[\bar{n}] = C_p[\bar{n} + (q-1)N]$ for all integers $q = 1, \dots, Q$ (Q denotes the number of cycles). Equation (6) is therefore equivalent to

$$\forall q \in \{1, \dots, Q\} \ \forall \bar{n} \in \{1, \dots, N\} \quad d[\bar{n} + (q-1)N] = \sum_{p=0}^{P} C_p[\bar{n}](\bar{n} + (q-1)N)^p \tag{8}$$

Using Newton's binomial formula,

$$(\bar{n} + (q-1).N)^p = \sum_{i=0}^{p} l_i^p N^i \, (\bar{n} - N)^{p-i} q^i \tag{9}$$

where l_i^p stands for the binomial coefficient), we can deduce that the sample associated with position $\bar{n}$ in the period, $s_q[\bar{n}] = D[\bar{n} + (q-1).N]$ for all integers $q \in \{1, \dots, Q\}$, defines a polynomial of order P with constant coefficients, i.e.

$$\forall q \in \{1, \dots, Q\} \ \forall \bar{n} \in \{1, \dots, N\} \quad s_q[\bar{n}] = \sum_{p=0}^{P} b_p[\bar{n}] q^p \tag{10}$$

where $b_p[\bar{n}] = N^p \sum_{j=p}^{P} C_p^j \, (\bar{n} - N)^{j-p} c_j[\bar{n}]$. It is important to note that $b_p[\bar{n}]$ is parametrized by $\bar{n}$. In the case of a noisy signal $X[n]$, a good estimate of the deterministic part is to find the right curve $\boldsymbol{s}[\bar{n}] = \left[s_1[\bar{n}], \dots, s_Q[\bar{n}]\right]^T$ for any $\bar{n} \in \{1, \dots, N\}$ which reduces the noise to find an estimate of $\boldsymbol{b}[\bar{n}] = \left[b_1[\bar{n}], \dots, b_P[\bar{n}]\right]^T$ for each $\bar{n} \in \{1, \dots, N\}$ for an order P of the polynomial. This could be done by finding the curve that minimizes the error in the sense of least squares, i.e.:

$$\begin{aligned} \forall \bar{n} \in \{1, \dots, N\} \quad \hat{\boldsymbol{b}}[\bar{n}] &= \operatorname{argmin}\left(\sum_{q=1}^{Q} W[\bar{n} + (q-1)N]^2\right) \\ &= \operatorname{argmin}\left(\sum_{q=1}^{Q} \left(D[\bar{n} + (q-1)N] - x_q[\bar{n}]\right)^2\right) \\ &= \operatorname{argmin}\left(\sum_{q=1}^{Q} \left(\sum_{p=0}^{P} b_p[\bar{n}] q^p - x_q[\bar{n}]\right)^2\right) \end{aligned} \tag{11}$$

And the solution is written as:

$$\forall \bar{n} \in \{1, \dots, N\} \ \hat{\boldsymbol{b}}[\bar{n}] = (\boldsymbol{\Phi}^{\mathbf{T}} \boldsymbol{\Phi})^{-1} \boldsymbol{\Phi}^{\mathbf{T}} \boldsymbol{x}[\bar{n}] \tag{12}$$

Once the coefficients calculated, the signal of interest estimated at location $\bar{n}$ is found as;

$$\forall \bar{n} \in \{1, \dots, N\} \ \hat{\boldsymbol{s}}[\bar{n}] = \left[\hat{s_1}[\bar{n}], \dots, \widehat{s_Q}[\bar{n}]\right]^T = \boldsymbol{\Phi} \hat{\boldsymbol{b}}[\bar{n}] = \boldsymbol{\Phi}(\boldsymbol{\Phi}^{\mathbf{T}} \boldsymbol{\Phi})^{-1} \boldsymbol{\Phi}^{\mathbf{T}} \boldsymbol{x}[\bar{n}] \tag{13}$$

The estimation of the rotor contribution in the signal is deduced as:

$$\forall n \in \{1, \dots, L\} \ \ \hat{D}[n] = \hat{s_q}[\bar{n}] \quad \text{where} \quad \bar{n} = \lfloor (n-1)/N \rfloor + 1 \ \text{ and } \ q = 1 + \ (n - \bar{n})/N \tag{14}$$

3.3 Separation of blades contributions

Now that the contribution of the rotor in the stress signal $\hat{D}[n]$ is estimated, it is important to separate the contributions coming from these blades. This is possible thanks to the signal coming from the top-blade sensor which measures the passage of each blade. Since these positions are well known in angular terms, we therefore propose an estimator of the signal generated by the blade r by applying a periodic window favouring the instants linked to the blades of interest:

$$\forall n \in \{1, \dots, L\} \qquad \hat{d}_r[n] = \hat{D}[n] F(r - r/R) \tag{15}$$

with

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$$F(n) = \sum_{q=1}^{Q} f[n - q.N] \tag{16}$$

and $f[n]$ is a weighting window having as effective length (or standard deviation) equals to r/2R. A possible example of such a window is the rectangular window centred on zero defined as:

$$f[n] = \begin{cases} 1 & si \quad -r/2R \leq n \leq r/2R \\ 0 & elsewhere \end{cases} \tag{17}$$

The process of blade contribution separation is illustrated in Fig. 2.

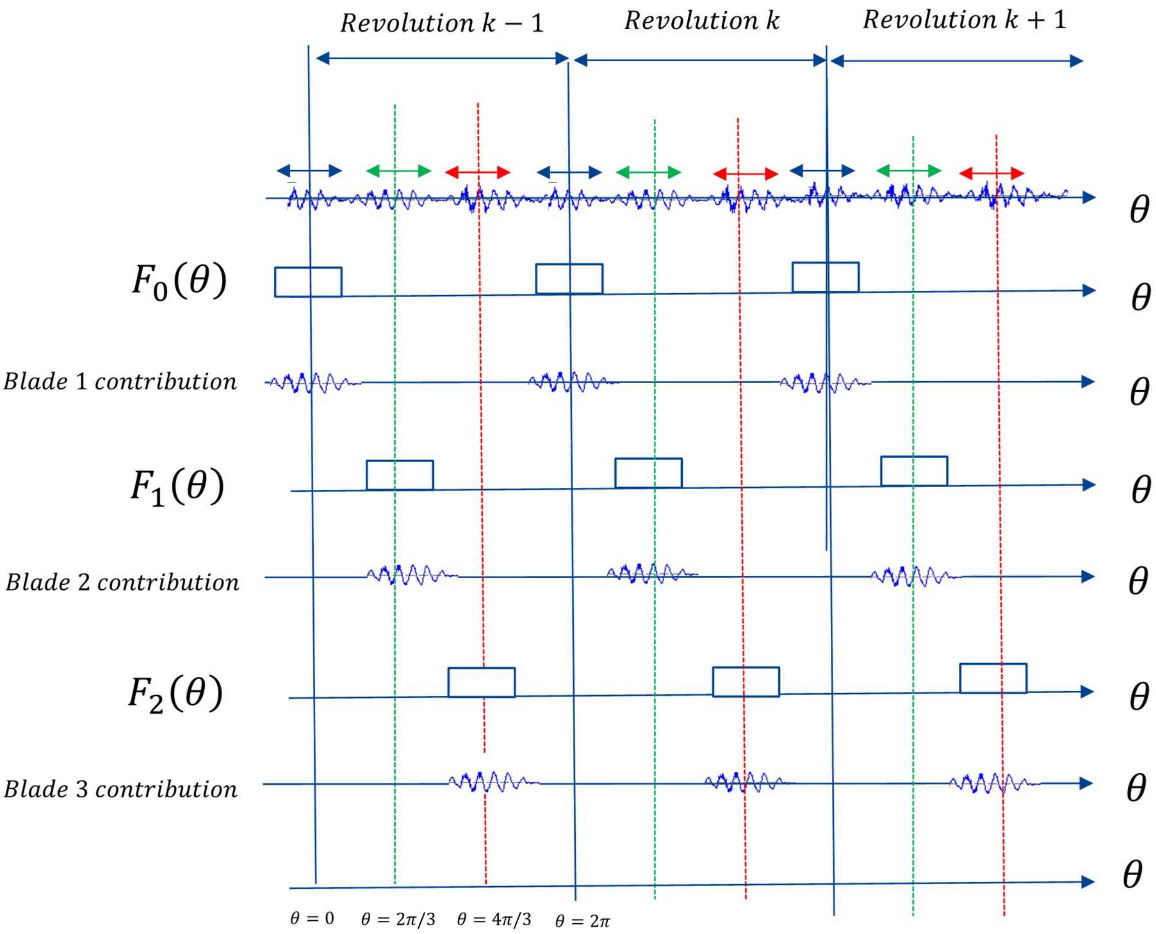

Figure 2: Illustration of the blade separation process.

3.4 Condition indicators

After the estimation of $\hat{d}_r[n]$, we propose the condition indicator of the blade rub. The latter will be function of the cycles of the rotor (i.e. a value for each cycle). We define this indicator for the r^{th} blade as the root mean square value of the contribution of the same blade calculated for each cycle:

$$\forall q \in \{1, \dots, Q\} \qquad I_r[q] = \sqrt{\frac{1}{N} \sum_{\bar{n}=1}^{N} \hat{d}_r[\bar{n} + (q-1).N]^2} \tag{18}$$

It is recalled that the index q designates the number of the cycle and Q the total number of cycles.

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3.5 Decision

Although the indicator should theoretically return a zero value in the absence of a rub, it is never zero in practice because of estimation errors. For this, it is important to set thresholds to confirm the occurrence of a hit. We thus propose to calculate the average and the standard deviation of the indicator on Q_{ref} cycles taken as reference (no rotor/stator contact). Let $I_r^{(ref)}[q]$ be the indicator of the r^{th} blade on the reference cycles, the average is calculated as

$$\mu_r^{(ref)} = \frac{1}{Q_{ref}} \sum_{q=1}^{Q_{ref}} I_r^{(ref)}[q] \tag{19}$$

While the standard deviation is calculated as

$$\sigma_r^{(ref)} = \sqrt{\frac{1}{Q_{ref}} \sum_{q=1}^{Q_{ref}} I_r^{(ref)}[q]^2} \tag{20}$$

We set the threshold indicating the occurrence of the rubbing phenomenon at the reference mean plus three times the standard deviation:

$$\lambda_r = \mu_r^{(ref)} + 3\sigma_r^{(ref)} \tag{21}$$

4. Application and results

4.1 Experimental setup

To test blade rubbing phenomenon and its consequences on the stator, a specific rubbing test bed has been used (Figure 3 a). The experimental setup consists in a rotor combining three identical metallic blades, and a fully integrated stator section mounted on a sliding table (Figure 3 b). Several strain gauges have been bonded on the stator parts (Figure 3 c) to be able to measure strain values near the contact location during tests. Several tests have been conducted using a rotor speed of 15000 rpm. The penetrating speed (relative approach between stator and rotor) has been fixed between 0.01 and 0.15mm/s. Maximum penetrating depth of 0.4mm has been reached during tests. To be able to measure properly dynamic phenomena, the strain sampling frequency has been fixed to 240 kHz during tests.

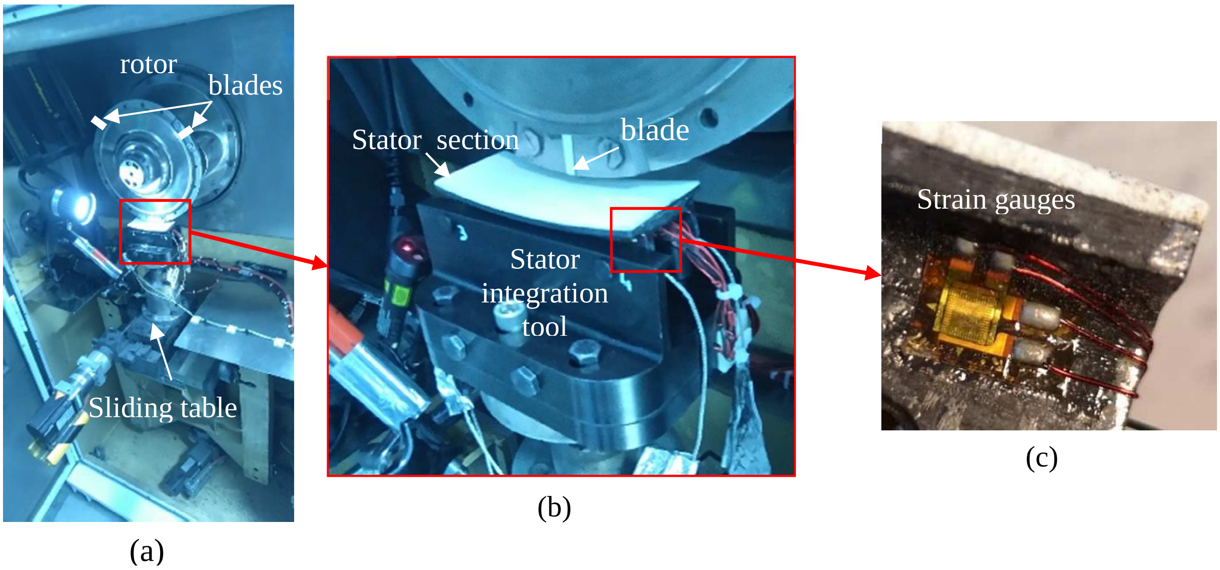

Figure 3. Illustration of the rubbing test bed (a), focus on the integrated stator section (b) and the strain gauges bonded on the stator section (c).

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4.2 Application of the monitoring strategy

This section is dedicated to validating the proposed approach on the signal obtained from the experiment described in section 3. The time signal measured by the strain gauge is displayed in Fig. 4 (top plot) together with its spectrogram. The spectrogram indicates the presence of strong electromagnetic interferences that mask the mechanical component generated by the blades; these components appear as horizontal lines in the spectrogram (being typically harmonics with fixed frequencies asynchronous to the shaft rotation). The rubbing occurs after 18 s and stay almost two seconds during which is the depth of the rubbing changes. The occurrence of this phenomenon is seen in the spectrogram and manifests through a wideband waveform that extends over a large frequency band. As expected, the excitation is angle-periodic with respect to the blade rotation and blade pass frequencies. Nevertheless, it is important to estimate the strain signal associated with the rubbing itself to clearly identify and quantify the phenomenon. Though being acquired under a stationary condition, the time signal is first resampled using a blade position sensor that measures the passage of the blades to compensate small speed fluctuations. The obtained signal is now synchronized with the blade positions and its plot is exposed in Fig 2 (top plot).

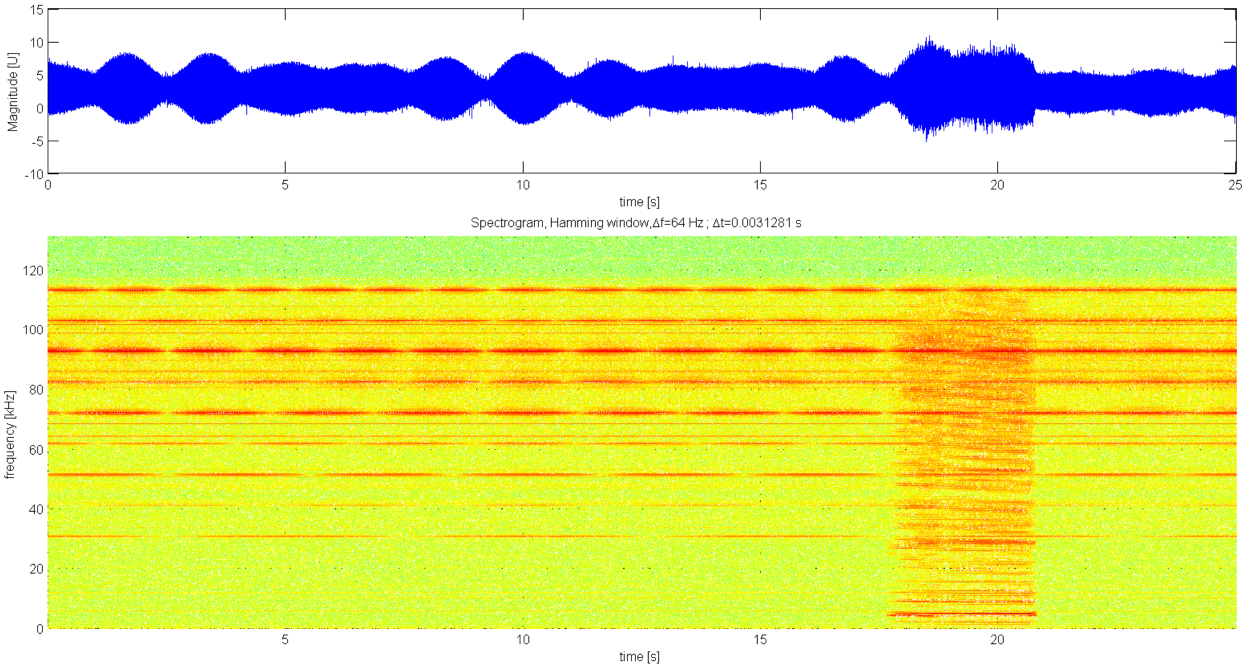

Figure 4: Raw strain gauge signal and its spectrogram.

The strain component related to the rotor set is estimated as described in subsection 3.2 using a polynomial of order 6. The obtained results are displayed in Figure 5. Very interestingly, the estimated rotor signal returns small values before and after the rubbing phenomenon, but higher values during the rub. This clearly indicates the presence and the strength of the rubbing. The residual signal, which basically consists of electromagnetic interferences, seems more regular and has similar time-statistics along the acquisition. This is expected as this latter is independent from the mechanical condition of the system. It is important to note that the strain signal contains time-varying DC components which turns out to be useless in the detection process. This component is separated by simply applying a narrow-band low-pass filter and then disregarded. Eventually, a close-up of the waveform is displayed in Fig. 6 during the rubbing, showing three events in each cycle associated with the response of each blade in the rotor (the rotor consists of three blades).

In the following, the three contributions of the rotor blades are computed as described in subsection 3.3 and the obtained results are displayed in Figure 7. The plots clearly distinguish the transient waveforms generated by the rub between each of the turbomachine blade with the stator. This is eventually used to construct the blade rub condition indicators as described in 3.5. The statistical threshold is computed as described in 3.6 taking the first 500 cycles of the turbomachine as a reference (with no rub). Results are displayed in Figure 8, showing clearly the instants of the rubbing in the turbomachines. These results confirm the effectiveness of the proposed strategy in detecting blade rub in turbomachine.

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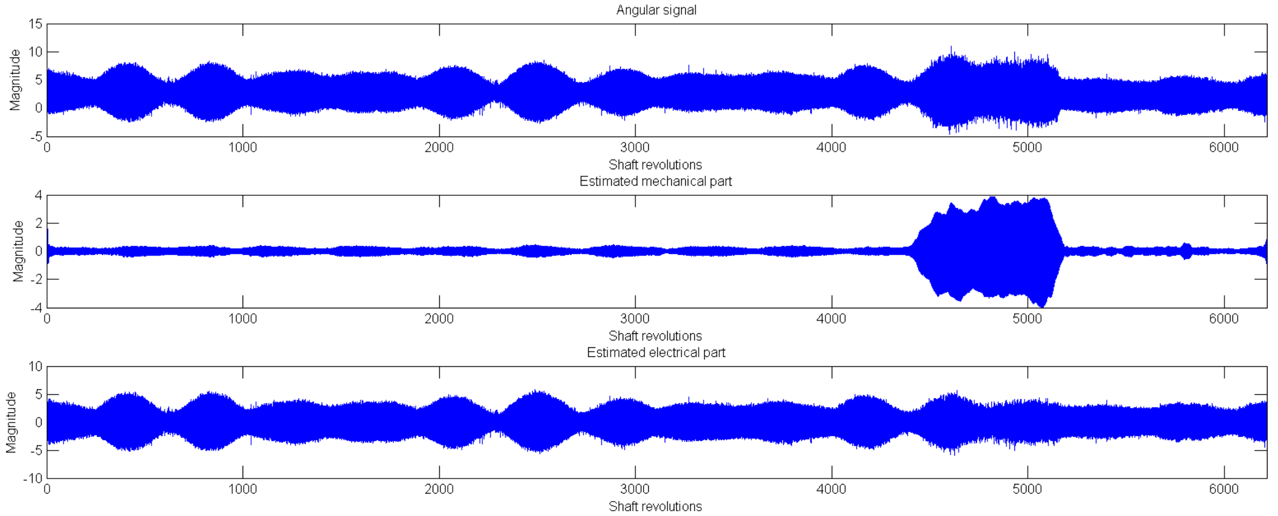

Figure 5: Raw strain gauge signal in the angular domain (top plot), the estimated rotor component (middle plot) and the corresponding residue that basically consists of the electrical part and other noises (bottom plot).

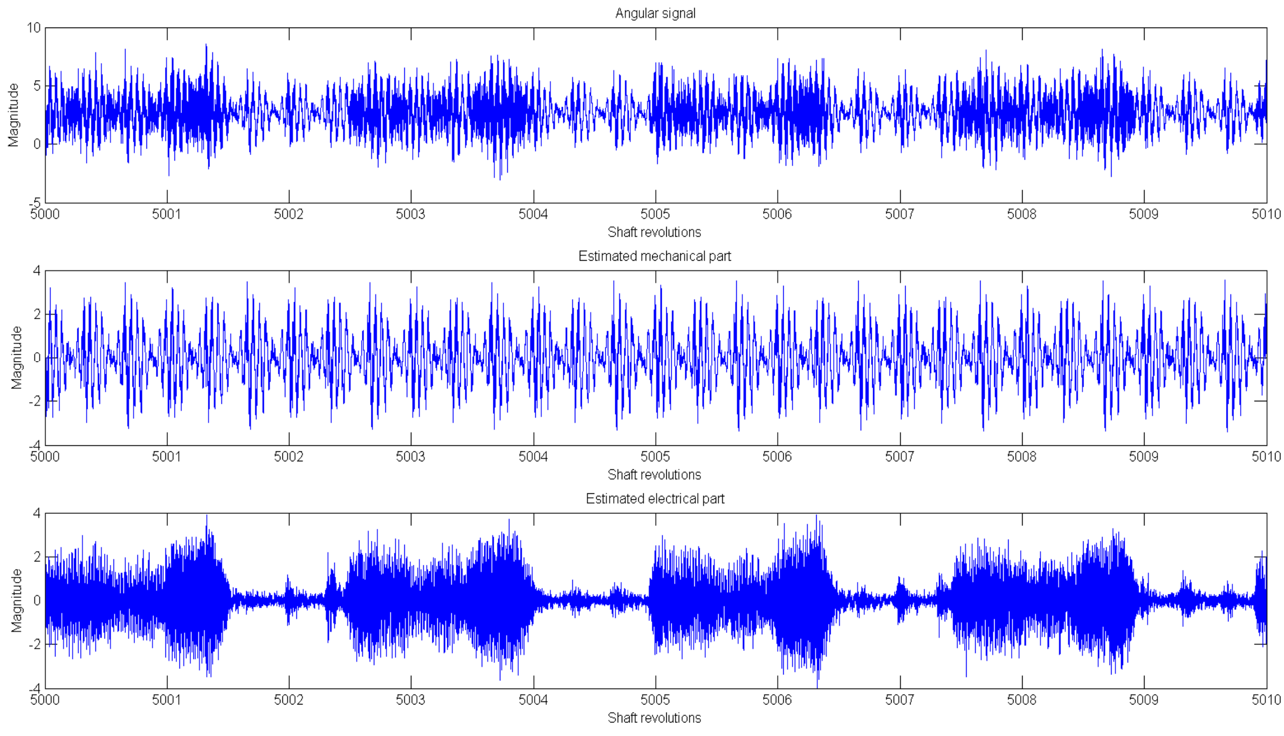

Figure 6: Close-up of figure 5 in the rubbing zone.

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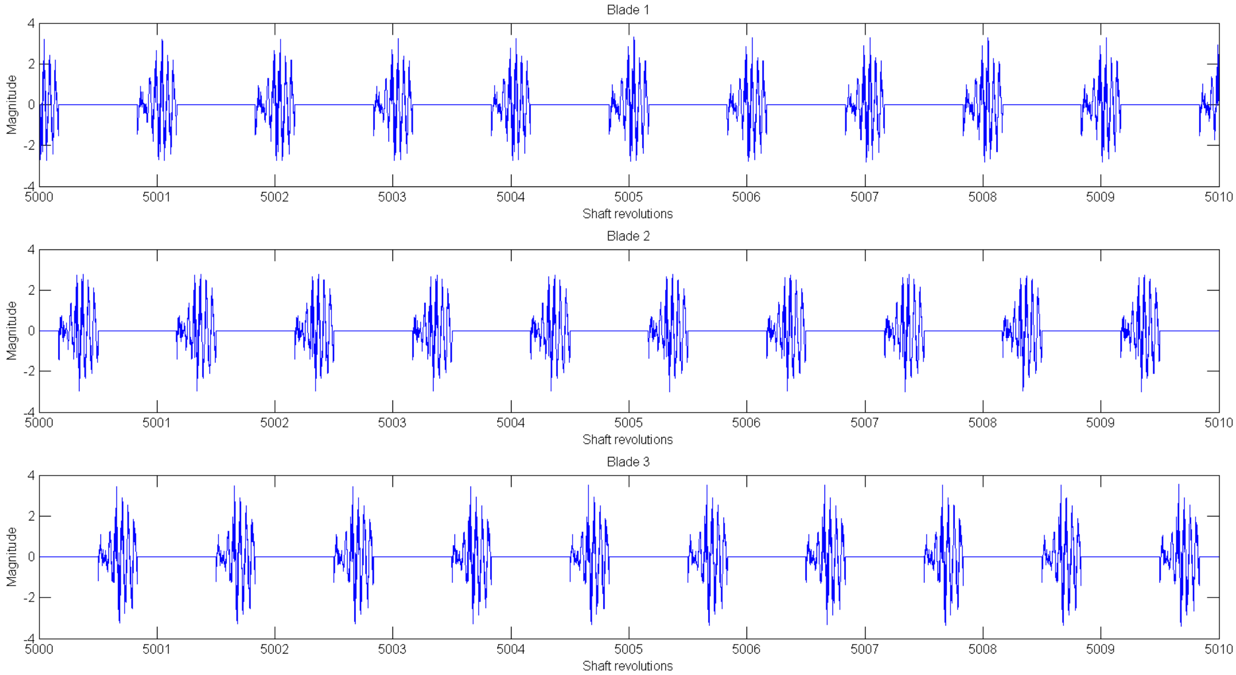

Figure 7. The estimated contributions of the three turbomachine blades.

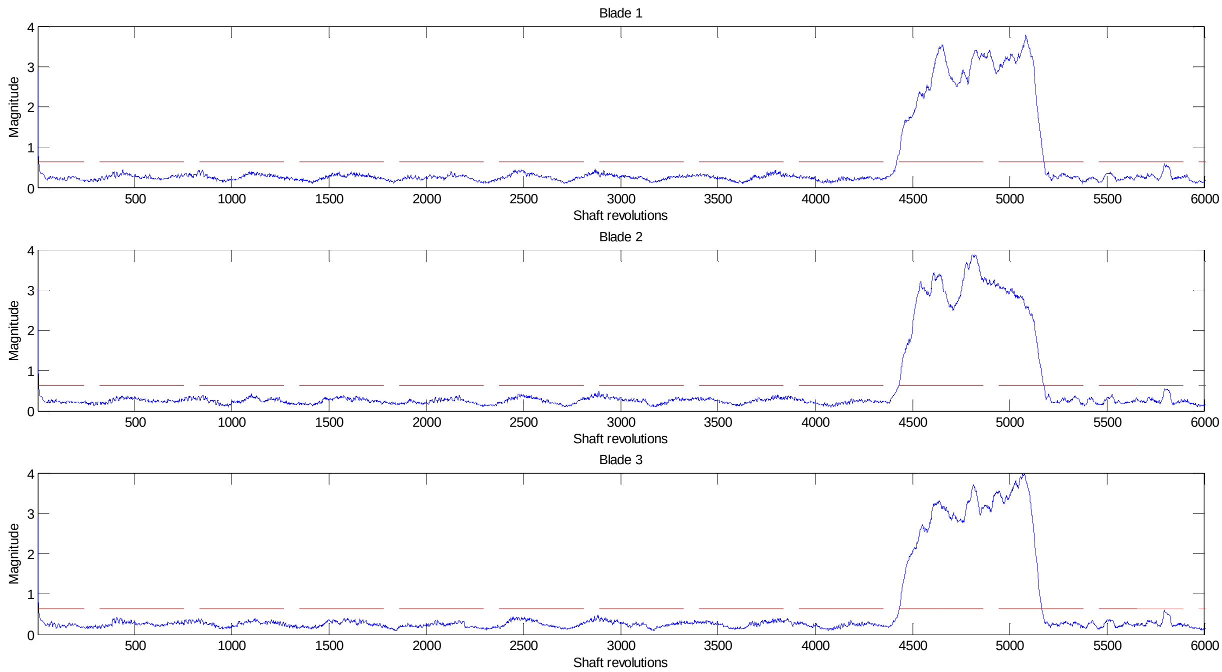

Figure 8. The condition indicators associated with the three blades of the turbomachine.

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5. Conclusion

This paper addresses a proposed condition monitoring strategy for blade rubs in turbomachines based on strain measurements. This strategy is based on the fact that the rubbing phenomenon creates a wave shock that deform the structure, these latter can be efficiently measured by a strain gauge mounted on the stator. However, strain signals are generally masked by strong mechanical and electromagnetic interferences. This paper proposes a signal separation technique based on mathematical modelling of the rubbing phenomenon. Condition indicators are proposed to quantify the rubbing phenomenon of each blade. The proposed methodology is validated on real signals captured from a strain gauge mounted on the stator of a turbomachine, and the obtained results assert the relevance of the proposed strategy in detecting blade rubs in turbomachines.

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