UNTYPICAL VIBROACOUSTIC SYMPTOMS OF A STEAM TURBINE FLUID-FLOW SYSTEM FAILURE

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Summary

The paper deals with a fluid-flow system failure in a 200 MW steam turbine. Relevant vibroacoustic symptoms were present in the blade frequency range; what is of interest, however, certain symptoms appeared also in the harmonic range. The case has been thoroughly analyzed and probable failure scenario has been outlined. Typical vibration-based diagnostic procedures, based on the assessment of vibration levels in certain frequency bands, are augmented by correlation and vibration trends analyses. Both are shown to be of great use. Results are viewed in terms of more general considerations of evolutionary diagnostic symptoms and their applications in rotating machinery diagnostics.

Keywords: vibration, technical diagnostics, steam turbine, fluid-flow system.

NIETYPOWE SYMPTOMY WIBROAKUSTYCZNE AWARII UKŁADU PRZEPŁYWOWEGO TURBINY PAROWEJ

Streszczenie

Artykuł dotyczy awarii układu przepływowego turbiny parowej 200 MW. Symptomy wibroakustyczne były obserwowane w zakresie częstotliwości łopatkowych, jednak, co jest szczególnie interesujące, pewne symptomy pojawiły się również w zakresie harmonicznym. Przypadek został przeanalizowany i przedstawiono prawdopodobny przebieg awarii. Typowe procedury wibrodiagnostyki, oparte na ocenie poziomów drgań w pewnych pasmach częstotliwości, są wspomagane przez analizy korelacji i trendów drgań. Okazują się one bardzo przydatnym narzędziem. Wyniki są ocenione pod kątem bardziej ogólnych rozważań, dotyczących symptomów ewolucyjnych i ich przydatności w diagnostyce maszyn wirnikowych.

Słowa kluczowe: drgania, diagnostyka techniczna, turbina parowa, układ przepływowy.

1. INTRODUCTION

Generation of vibration as a result of interaction between turbine fluid-flow system and steam flow has been described in a number of publications (see e.g. [1, 2]). Detailed treatment of this complex issue is beyond the scope of this paper; we shall limit our attention to certain implications important for technical condition assessment. It has been shown [3, 4] that vibration levels in certain frequency bands are directly related to the condition of individual fluid-flow system stages and can thus be employed as diagnostic symptoms. These bands usually fall into the frequency range from a few hundred Hz to about 10÷20 kHz, which is often referred to as the *blade frequency range*.

Relation between diagnostic symptoms and condition parameters (conveniently represented by vectors, **S** and **X**, respectively), can be most generally expressed by [3]:

$$\mathbf{S}(\theta) = \Phi[\mathbf{X}(\theta), \mathbf{R}(\theta), \mathbf{Z}(\theta)] \ , \qquad (1)$$

where Φ denotes some general operator. In Eq. (1), vectors **R** and **Z** describe influences of control and external interference, respectively. All these four vectors depend on time θ.

In some cases, influences of control and interference can be neglected. Moreover, for a given symptom $S_i(\theta) \in \mathbf{S}$ we can sometimes assume that:

$$\bigvee_j \partial S_i / \partial X_k \approx 0, \quad \text{if } j \neq \text{k}. \qquad (2)$$

This yields the simplest possible form of diagnostic relation, namely:

$$S_i(\theta) \approx \Phi[X_j(\theta)] \ . \qquad (3)$$

Relations of this type are in fact often used in practice, despite the fact that underlying assumptions are not necessarily met.

It has been shown [5] that, for vibrodiagnostic symptoms pertaining to the turbine fluid-flow system condition, such simplification is not appropriate. These symptoms exhibit considerable dependences on both **R** and **Z**. While the former can, in principle, be normalized [6], some components of the interference vector are not measurable and any normalization is out of question. Time histories of such symptoms are thus usually rather irregular. Often a sudden increase is observed, even above the limit value, followed by a decrease (see Fig.1)[1]. This can trigger a false alert, so typical diagnostic reasoning based on vibration level assessment should in such cases be augmented by vibration trend analysis.

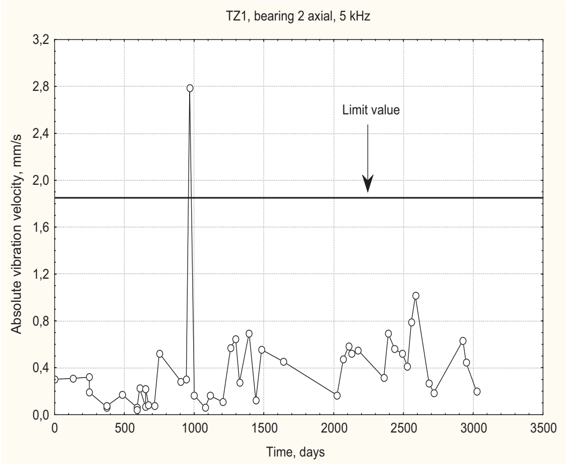


Fig. 1. Typical absolute vibration trend from the blade frequency range: 13CK230 turbine, HP/IP turbine bearing, axial direction, 5 kHz band (23% CPB spectrum). Limit value was exceeded probably due to interference

Turbine fluid-flow system failures are a rare occurrence. Thus, a case wherein symptom time histories and failure type and extent are known is certainly an interesting example. Detailed analysis can lead to important and more general conclusions.

2. DESCRIPTION OF THE CASE

PWK-200 steam turbines, dealt with in this paper, include a high-pressure (HP) turbine, intermediate-pressure (IP) turbine and two-stream low-pressure (LP) turbine. Schematic layout of the unit is shown in Fig. 2.

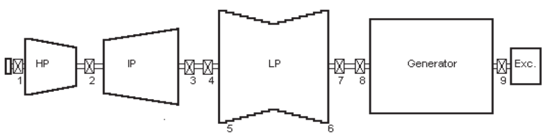


Fig. 2. Schematic drawing of a PWK-200 turbine-generator unit (numbers denote vibration measurement points)

In May 2005, PWK-200 turbine No.9 suffered an intermediate-pressure (IP) rotor failure. Several blades of the last rotor stage cracked and the turbine had to be shut down. No additional damage occurred. Immediate repair was not possible; last turbines of this type were commissioned in early 1980s, so neither spare blades nor a replacement rotor were readily available. As long outage was unacceptable, it was decided to remove the entire rotor stage blading and restore the turbine in operation.

Immediately afterwards, a major change of turbine vibration characteristics was observed. Absolute vibration levels in frequency bands pertaining to last IP turbine stages increased in a stepwise fashion, mainly in axial direction. Examples are shown in Fig. 3; they refer to bearing No. 3, which is the rear IP bearing, adjacent to the failure location.

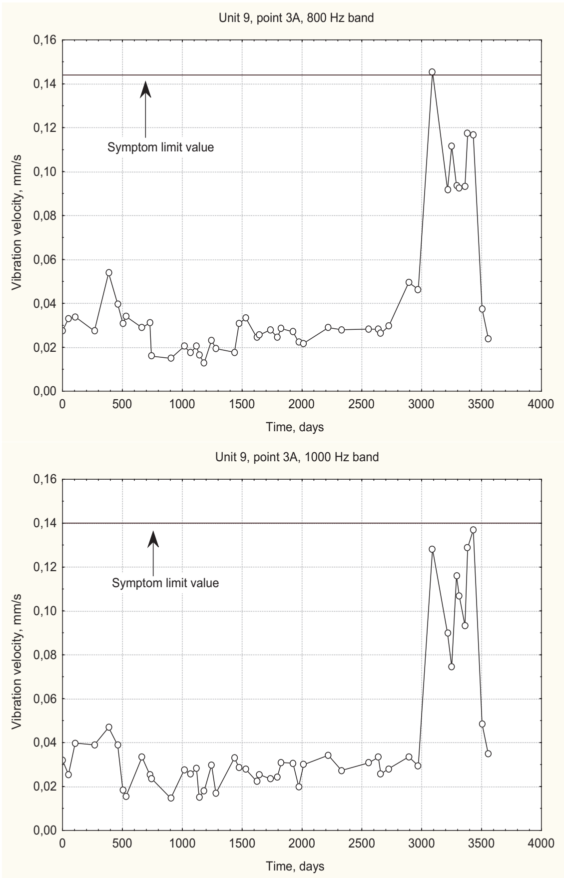


Fig. 3. Absolute vibration trends: Unit No. 9, rear IP turbine bearing, axial direction, 800 Hz (upper) and 1 kHz (lower) bands (23% CPB spectrum)

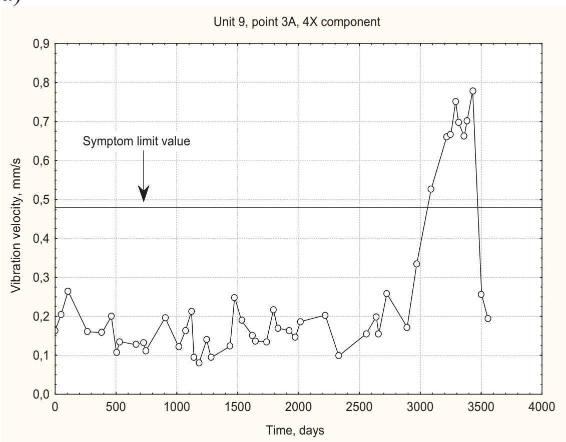
Such stepwise modification of vibration characteristics in the blade frequency range can be readily explained on the basis of turbine vibrodiagnostic model [3]. As a result of changes in the fluid-flow system itself, spatial distribution of forces produced by interaction between its elements and steam flow also changes and this modifies vibration patterns.

Since turbine was restored in operation, a steep, but continuous increase of the 4X harmonic component had been observed in many measuring points, mainly those physically related to the IP

[1] i.e. limit value determined from the symptom reliability (for details, see e.g. [7]).

turbine[2]. An example is given in Fig. 4. Initially these components did not exceed levels typical for these turbines and were much lower than first two harmonics. During about sixteen months some of them increased by four times and this was accompanied only by moderate or even negligible increase of the 1X and 2X components. It is easily seen from Fig. 4 that increase ratio decreased slightly with time, but the trend had, at least initially, been almost linear. Limit values were substantially exceeded.

a)



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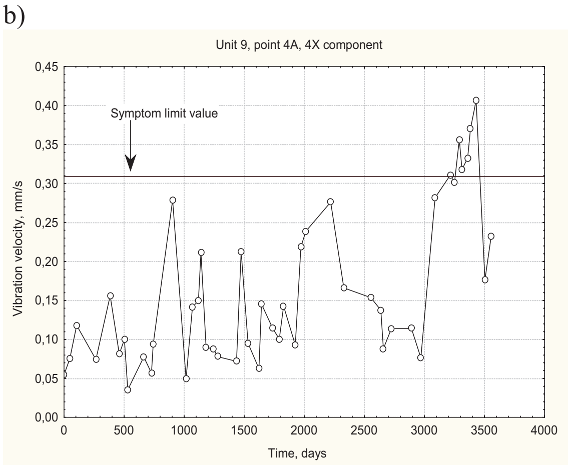


Fig. 4. Absolute vibration trends: Unit No. 9, 4X component, axial direction, rear IP turbine bearing (upper) and front LP turbine bearing (lower)

Turbine was finally overhauled in October 2006 and until that time overall vibration levels had not exceeded alert levels. It was decided to replace the entire IP rotor. After opening IP turbine casing it was found that steam flow guide fences, mounted inside the outlet part of the casing, were loose and some exhibited cracks (see Fig. 5), so they had to be repaired. One fence fell off and was later found in the lower part of the casing. After the overhaul situation changed again, this time returning to ‘normal’ (see Table 1). During subsequent period of turbine operation fluctuations of the 4X component have been relatively small and no monotonic increase has been observed.

[2] In PWK-200 turbines, bearings from 3rd to 6th (i.e. rear IP, both LP and front generator) are mounted on a common plate which comprises the lower half of the LP turbine casing.

Table 1. Comparison of the 4X component levels

Point and direction	Vibration velocity, 4X component [mm/s]		
	limit value	before overhaul	after overhaul
HP/IP bearing, A	0.250	0.292	0.197
Rear IP bearing, A	0.480	0.779	0.256
Front LP bearing, V	0.148	0.195	0.032
Front LP bearing, A	0.309	0.406	0.177

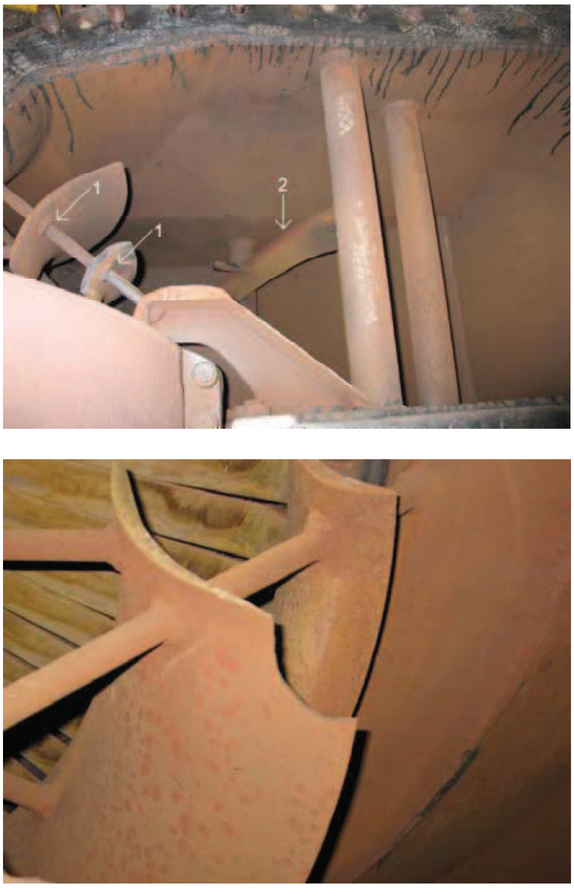


Fig. 5. Photographs of damaged guide fences; in the upper photo, arrows indicate loose welded joints (1) and the guide fence that has fallen off (2); lower photo shows damaged guide fence edge

3. ANALYSIS OF THE CASE

Steam turbine vibration spectra are often characterized by relatively high levels of higher harmonic components. Obviously, they will be present even with a purely harmonic excitation, due to nonlinearity [8, 9]. Thus, this is not the very appearance of these components that suggests a malfunction, but rather their increase above a certain level which is considered ‘normal’ and may, in terms of quantitative condition assessment, be viewed a basic symptom level [10].

While there are widely accepted guidelines concerning 1X and 2X components (see e.g. [3, 8]), interpretation of higher harmonic components is in general more ambiguous. According to some guidelines, 3X and 4X components can be associated with faulty machining of rotating elements, coupling malfunctions or rotor cracks. Sometimes they result from resonance of casings and bearing supports, which in turn suggests changes in stiffness and/or damping caused e.g. by material parameters degradation. No diagnostic relations – even of qualitative nature – suitable for practical applications have so far been developed. Example concerning the 3X component, given in [11], is of purely qualitative nature and lacks support based on a wellestablished model, although relation has been confirmed in practice.

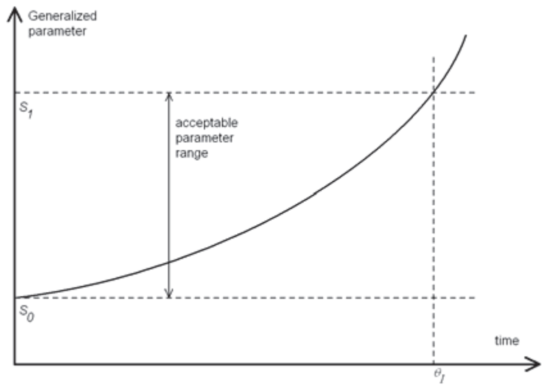
In this particular case, increase of the 4X component and its subsequent decrease are evidently coincident with removing last rotor stage blades and rotor replacement, respectively. Therefore it seems reasonable to suspect some relation between these events. What is, however, of substantial importance here is the type – or shape – of the $S(\theta)$ curve. If we consider some generalized parameter pertaining to lifetime consumption (not necessarily a diagnostic symptom), three types are possible [12], namely:

- *Natural damage* (Fig. 6a), which is a result of slow and irreversible lifetime consumption during normal operation;
- *Random* or *catastrophic damage* (Fig. 6b), characterized by very fast ($dS/d\theta \to \infty$) increase of the symptom value above its acceptable level;
- *Inertial damage* (Fig.6c), resulting from the influence of external factors if they exceed acceptable limits; such damage is in principle reversible.

While this classification and names are certainly open to discussion, one issue is certainly important: diagnostic information is contained not only in symptom value, but also in the shape of the $S(\theta)$ curve.

As we can easily see in Fig. 3, increase of the components from the blade frequency range is in fact instantaneous and their further fluctuations, albeit large, do not exhibit any monotonic trend. On the other hand, levels of the 4X component in various measuring points, after the initial jumps, tend to increase monotonically until finally exceeding limit values. There are almost no signs of ‘saturation’ and it is reasonable to expect that, had the replacement not been made, there would have been further increase to even higher values.

a)



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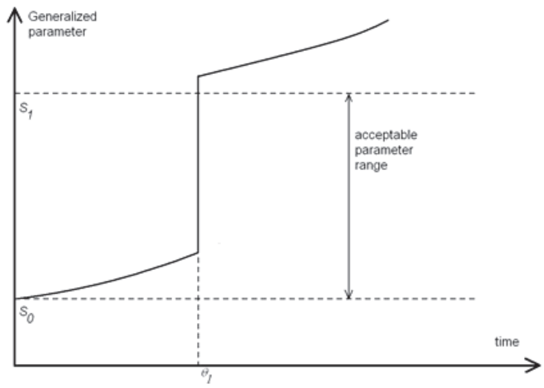

c)

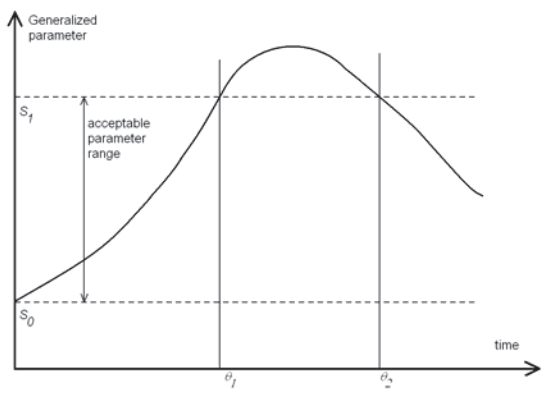

Fig. 6. Three theoretical types of $S(\theta)$ curves: natural damage (a), random damage (b) and inertial damage (c) (after [12])

The above observation becomes even more clear if we analyze the behavior of normalized dimensionless symptom values versus time; normalization is here done with respect to symptom values immediately before initial failure, denoted by S_0. Corresponding plots of $s = S/S_0$ are shown in Fig. 7. We can easily see that, for two bands from the blade frequency range, values of s immediately increase to about 3.1 and 4.3, respectively, and then show considerable fluctuations. This is equivalent to the theoretical case illustrated in Fig. 6b. On the other hand, for the 4X component there is an almost monotonic increase during the entire period until repair and maximum value of s, which is reached at the end of this period, is about 2.3. This in turn corresponds to the theoretical case from Fig. 6c.

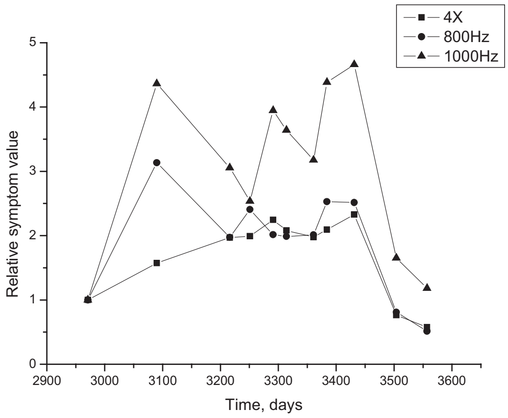

Fig. 7. An example of normalized symptom S/S_0 time histories; rear IP turbine bearing, axial direction

Important diagnostic information is also provided by correlation analysis. Every $S(\theta)$ given in the form of a trend (not a continuous function – cf. Fig.4) may be referred to as a random variable. It is known that, for two independent random variables, their correlation coefficient is equal to zero. The reverse is not necessarily true. Speaking in terms of diagnostic symptoms, we may infer that if two experimental trends $S_1(\theta)$ and $S_2(\theta)$ exhibit a high correlation coefficient value, they probably depend on the same physical phenomenon.

Statistical analysis has shown that correlation between 1X and 4X components is poor. Correlation coefficients[3] for three measurement points (HP/IP bearing, rear IP bearing and front LP bearing) in three directions are typically below 0.3 and only in two cases (both in vertical direction) slightly exceed 0.5. These values should be considered low. In a case of an excessive 3X harmonic component increase, briefly described in [11], correlation coefficients between 1X and 3X components for five PWK-200 turbines ranged from 0.55 to 0.89. Of course, this can hardly be considered decisive; we can, however, suspect that prime reason of the 4X component increase is not directly related to the rotary motion.

On the other hand, correlation between the 4X component and amplitudes in spectral bands that contain blade components turns out to be very good, especially for the lower part of the blade frequency range. Corresponding values are listed in Table 2 (rear IP bearing, axial) for three turbines, namely No.9 considered here and two identical units, No. 8 and No.10, at the same power plant. Differences are evident indeed; below 2500 Hz correlation coefficients for the turbine No.9 range from 0.69 to 0.93, while for other two units – from 0.04 to 0.24. On the basis of this data we can suspect that time histories of the 4X component and of amplitudes in spectral bands from the blade frequency range have been determined by the same physical phenomenon or, more strictly speaking, by the same damage development mechanism.

[3] Pearson linear correlation coefficients.

Table 2. Correlation coefficients for three turbines (rear IP bearing, axial direction)

Frequency band [Hz] 23% CPB spectrum	Coefficient of correlation with the 4X component		
	Turbine No. 8	Turbine No. 9	Turbine No. 10
800	0.16	0.91	0.06
1000	0.11	0.93	0.17
1250	0.07	0.83	0.13
1600	0.22	0.79	0.16
2000	0.04	0.69	0.24
2500	0.06	0.55	0.18
3150	0.15	0.19	0.10
4000	0.09	0.20	0.22
5000	0.01	0.15	0.39
6300	0.01	0.23	0.41

4. PROBABLE EVENT HISTORY

In this particular case, we have been in a lucky position, as damage type and extent had been known before all symptom time histories analyses were performed. This, however, does not mean that the case is not worth attention.

On the basis of all observations described here, the following possible event scenario can be drawn up:

1. Cracks (initially minor) in the IP turbine last rotor stage blades appear, most probably due to high-cycle fatigue (the turbine was commissioned in late 1960s).
2. Turbine is restored in operation with removed last rotor stage blades. This causes a dramatic change in the distribution of forces resulting from interaction between the fluid-flow system and steam flow. As a result, vibration levels in certain bands in the blade frequency range immediately increase, even by a few times.
3. Turbine operation in such condition results in a change of forces acting on the steam flow guide fences. As steam flow velocity is higher, we may presume that these forces have increased in magnitude; moreover, their distribution has changed substantially. This in turn excites vibration of fences, with frequency close to that of the 4X component, i.e. 200 Hz. This phenomenon is in fact a resonance.
4. Excited vibration, much higher in amplitude than during normal operation, causes rapid damage of the steam flow guide fences and intensifies their vibration (destructive feedback principle – see [13, 14]). Increase of the 4X component amplitude is monotonic and much faster than that resulting from normal degradation, but almost

linear. This phenomenon is particularly evident in the rear IP turbine bearing vibration patterns, as this bearing is close to the IP turbine outlet.

We can thus say that a reliable symptom-condition relation has been determined, albeit still of only qualitative nature.

5. CONCLUSIONS

The case itself is certainly a specific one, the first of that type known to authors. However, several more general conclusions are justified.

1. Harmonic components in turbine bearings absolute vibrations spectra are usually attributed to phenomena directly related to the rotary motion. This case certainly shows that this is not always correct and the prime cause can be different. Routine application of typical vibration pattern assessment guidelines can thus lead to an incorrect diagnosis.
2. Practical vibrodiagnostic procedures are usually based on vibration spectra evaluation, often supplemented by analysis of trajectories and/or relative vibration vectors. Diagnostic relations are usually, however, more complex than that generally given by Eq.(3). In most cases there are several phenomena (or, more precisely, several condition parameters) that contribute to the amplitude of a given vibration component. As a result, diagnosis can be ambiguous. In such cases, evolutionary symptoms can be very useful.
3. Important and valuable diagnostic information is contained in correlation coefficient values. In principle, a correlation coefficient can be considered a diagnostic symptom [3], although such symptom is not directly related to residual processes in the sense of the energy processor (EP) model [14]. Data given in Table 2 suggest that symptoms based on correlation coefficients might be very sensitive to damage type. Of course, reliable determination of a correlation coefficient value demands suitable database, but with state-of-the-art monitoring systems for critical rotating machines such database is usually readily available. It seems justified to conclude that diagnostic symptoms of this type deserve further attention and evaluation, both from theoretical and practical points of view (suitable model and case analyses).

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