CORRELATION ANALYSIS IN STEAM TURBINE MALFUNCTION DIAGNOSTICS

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Summary

Rotating machinery condition assessment employs, to a large extent, vibration-based symptoms, characterized by high information content. Diagnostic reasoning is often supported by well-developed qualitative or even quantitative relations. In many cases, however, a situation is encountered wherein several possible malfunction types give similar symptoms. Qualitative diagnosis may thus demand additional diagnostic procedures. The paper concentrates on those employing correlation coefficients, which belong to a broader class of evolutionary symptoms. Such symptoms are sensitive to the malfunction type and thus can substantially enhance qualitative diagnostic capabilities. Moreover, correlation coefficient as a function of time provides valuable quantitative information on lifetime consumption. This is demonstrated by a number of examples, pertaining to large steam turbines operated by utility power plants.

Keywords: technical diagnostics, diagnostic symptom, correlation, prognosis.

ANALIZA KORELACJI W DIAGNOZOWANIU USZKODZEŃ TURBIN PAROWYCH

Streszczenie

Ocena stanu technicznego maszyn wirnikowych opiera się w dużym stopniu na symptomach drganiowych, charakteryzujących się dużą zawartością informacji. Wnioskowanie diagnostyczne często wykorzystuje sprawdzone relacje diagnostyczne, jakościowe, a nawet ilościowe. W wielu przypadkach mamy jednak do czynienia z sytuacją, w której szereg możliwych uszkodzeń daje podobne symptomy. Diagnoza jakościowa wymaga wówczas dodatkowych procedur. Artykuł jest poświęcony zastosowaniu współczynników korelacji, należących do szerszej klasy symptomów ewolucyjnych. Są one wrażliwe na typ uszkodzenia i mogą znacznie poprawić możliwości diagnozowania jakościowego. Współczynniki korelacji w funkcji czasu dostarczają także cennej informacji ilościowej o stopniu wyczerpania żywotności. Jest to zilustrowane kilkoma przykładami, odnoszącymi się do dużych turbin parowych w energetyce zawodowej.

Słowa kluczowe: diagnostyka techniczna, symptom diagnostyczny, korelacja, prognoza.

1. INTRODUCTION

When studying technical diagnostics development, it is useful to have in mind the definition of damage as a 'continuous or sudden loss of integrity and/or operational feature' [1]. At early stages, attention was focused principally on 'sudden' losses, referred to as *random damages*: the aim of diagnostic methods was to detect such occurrences and identify malfunctions. This can be referred to as *qualitative diagnostics*. Next stages were to evaluate damage extent and eventually make a forecast (prognosis) concerning its future development: that, in turn, means focusing on 'continuous' losses, or *natural damages*, and *quantitative diagnostics*. Obviously this corresponds to the generalized damage concept (see e.g. [2,3]), wherein generalized damage can be expressed as $D = \theta/\theta_b$ (θ_b denotes time to breakdown and is determined by object parameters).

Quantitative diagnostic methods have developed considerably in recent years (a concise review for rotating machines can be found in [4]). It has to be kept in mind, however, that a large and complex machine has many possible faults, i.e. its technical condition is described by a multidimensional fault space [5]. At the same time, this condition is assessed on the basis of many diagnostic symptoms. It is convenient to express this in a vector form [6]:

$$\mathbf{S}(\theta) = F[\mathbf{X}(\theta)] \quad , \tag{1}$$

wherein $\mathbf{S}(\theta)$ and $\mathbf{X}(\theta)$ denote symptoms and condition parameters vectors, respectively. Now, consider a symptom $S_i(\theta)$. Obviously,

$$S_i(\theta) = f[X_1(\theta), X_2(\theta), \dots X_n(\theta)] \quad . \tag{2}$$

In fact, Eq.(2) may be viewed a more general form of the following relation, derived from the Energy Processor (EP) model (see [2,3]):

$$S(\theta) = \Phi\left[V_0\left(1-\frac{\theta}{\theta_b}\right)^{-1}\right], \qquad (3)$$

where V_0 denotes the power of residual processes for $\theta = 0$. It is easily seen that Eqs.(2) and (3) become equivalent if only one component of the vector **X**, i.e. $X = \theta/\theta_b$, is taken into account (this issue shall be recalled in Section 3.3). In the ideal case, we have[1]

$$\partial S_i/\partial X_k > 0 \text{ if } k = j,\ \ \partial S_i/\partial X_k \approx 0 \text{ if } k \neq j\ , \quad (4)$$

which means that, within a reasonable approximation, S_i depends on X_j only. Such assumption is sometimes acceptable for fast-developing faults with very specific representations in the symptom space; in such cases, diagnostic relations of the $S_i = f(X_j)$ type can be determined e.g. by means of regression analysis (see e.g. [8]). Usually, however, a symptom is influenced by a number of condition parameters; synchronous component of vertical vibration in rotating machines provides a good example [9,10]. Therefore, problems arise already at the stage of damage identification.

The problem is further exacerbated by the consequences of the fact that general relation of the type given by Eq.(1) is in most cases only approximate. Usually we have

$$\mathbf{S}(\theta) = F[\mathbf{X}(\theta), \mathbf{R}(\theta), \mathbf{Z}(\theta)]\ , \qquad (5)$$

where vectors $\mathbf{R}(\theta)$ and $\mathbf{Z}(\theta)$ describe control and interference, respectively. Control parameters R_i are determined by the operator's purposeful action and their influence can, in principle, be normalized (see e.g. [11]). Influence of interference, in particular non-measurable, cannot be normalized. A good example is provided by two vibration trends, recorded for a 230 MW steam turbine in a utility power plant (see Fig. 1). Both refer to the blade (high) frequency range, wherein vibration components are very sensitive to the **R** and **Z** vectors components [12]. It is easily seen that, for the 5 kHz component (Fig. 1a), there is a single 'peak', which corresponds to operation at a very low load, about 40% of the rated power. In such situations, pressure distribution over the fluid-flow system cross section, especially immediately downstream the steam inlet, is very uneven and this causes excessive vibration in this frequency range [12,13]. For the 3.15 kHz component (Fig. 1b) the same phenomenon can be observed, but there are two more 'peaks', obviously comparable in magnitude, which are absent in the trend shown in Fig.1a[2]. The first (indicated by *) can also be attributed to low load (50% of the rated power), but the second (indicated by +) occurred at 87% of the rated power. Most probably it can be attributed to some interference, but this cannot be resolved on the basis of vibration trends analysis only.

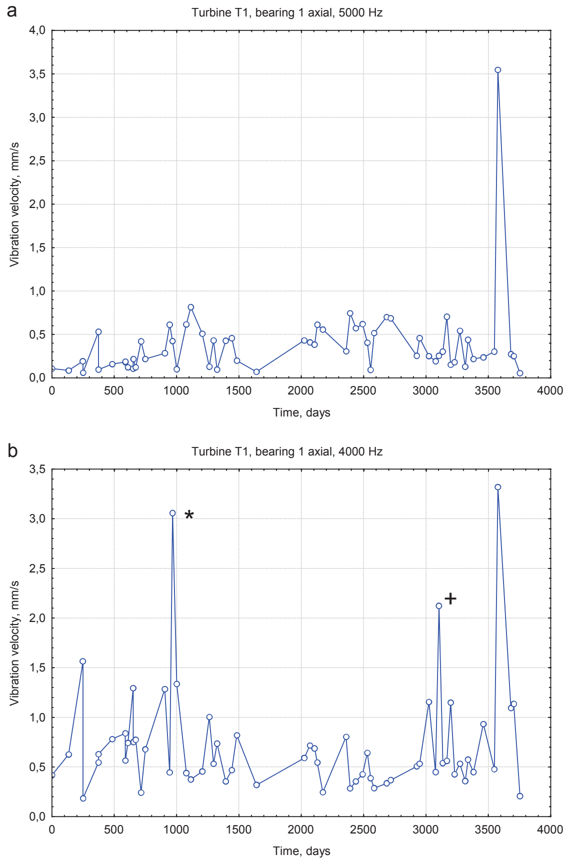


Fig. 1. Vibration velocity trends (23% CPB spectra) for a 230 MW steam turbine, front bearing, axial direction; a, 5 kHz; b, 4 kHz frequency band (see main text for details).

With this all in mind we are have to admit that typical procedures, such as analysis of vibration spectra obtained during steady-state operation and even trends analysis, are sometimes not sufficient for a qualitative diagnosis. This draws our attention to additional symptoms that can augment diagnostic reasoning – among them those employing quantitative measures of correlation. In fact, suitability of such symptoms has been pointed out earlier by some authors (see e.g. [10, 15]), albeit in somehow cursory and basically qualitative manner. In the following this issue shall be dealt with in more detail.

Although in this paper attention is focused on steam turbines and vibration-based symptoms, some results and conclusions can obviously be generalized, to cover a broader class of rotating machines or even technical objects and other types of diagnostic symptoms, not necessarily related to vibration.

[1] With the assumption that S_i monotonically increases with damage extent, in accordance with the EP model (for more details, see e.g. [3,7]).

[2] Influence of 'peaks' such as shown in Fig.1 can be, to some extent, reduced by averaging raw data; for details see e.g. [14].

2. BASIC CONSIDERATIONS

To begin with, let us consider a case wherein observation of turbine condition evolution covers a period from $\theta = 0$ to $\theta = \theta_0$ and furthermore that θ_0 is sufficiently large, typically of the order of magnitude of machine service life – for steam turbines this means at least a few years. In such case we may usually assume that components Z_i of the interference vector have no long-time evolution trends, i.e. do not systematically increase or decrease with time. This can be written as

$$\bigwedge_i \theta \to \infty \Rightarrow \frac{\Delta Z_i(\theta)}{\theta} \to 0 \quad , \qquad (6)$$

where $\Delta Z_i(\theta) = Z_i(\theta) - Z_i(0)$. The same usually applies to the control vector components R_i,[3] which means that, in a similar manner,

$$\bigwedge_i \theta \to \infty \Rightarrow \frac{\Delta R_i(\theta)}{\theta} \to 0 \quad . \qquad (7)$$

The above implies that, when considering symptom evolution between $\theta = 0$ and $\theta = \theta_0$, we may neglect influences of control and interference. This in fact means that we accept approximate relations of the type given by Eq.(2), and forms the basis for diagnostic reasoning based on vibration trends, both $S_i(\theta)$ function types [12,15] and evolution rates [16].

Let us now return to Eq.(2) and assume that the function given by this equation is differentiable with respect to every X_j, $j = 1, 2, \ldots, n$. We can thus define a set of partial derivatives $s_{ij} = \partial S_i/\partial X_j$ (cf. Eq. (4)), each representing the 'sensitivity' of i-th symptom to j-th condition parameter. Due to nonlinearity, s_{ij} in general will change with time. This set can be conveniently written in the $m \times n$ matrix form:

$$[s] = \begin{bmatrix} s_{11} & s_{12} & \ldots & s_{1n} \\ s_{21} & s_{22} & \ldots & s_{2n} \\ \ldots & \ldots & \ldots & \ldots \\ s_{m1} & s_{m2} & \ldots & s_{mn} \end{bmatrix} \quad , \qquad (8)$$

each row representing a symptom and each column a condition parameter. If we can assume that in each row one element is substantially larger than all other, we have an ideal situation with no ambiguity in *qualitative* diagnosis. This is, however, seldom the case with complex machines, even if we neglect the fact that the s_{ij} elements do change with time. Typically a malfunction (which is equivalent to a condition parameter change) influences a number of symptoms and each symptom can be influenced by a number of malfunctions. Some additional measures are thus necessary for failure identification.

In general, any symptom will depend on a number of condition parameters, so there will be no deterministic *functional relation* of the $S_i = f(X_j)$ type. We may, however, expect that changes of X_j will affect probability distribution of S_i, which implies a *stochastic relation*. Moreover, it is justified to assume that if X_j changes substantially, then with various values of X_k, $k \neq j$, the value of S_i will fluctuate about some expected value $\hat{S}_i = f(X_j)$. This means a *statistic* or *correlative relation*. If X_j influences a number of symptoms, its changes will affect their probability distributions accordingly. Thus, if two symptoms can be shown to be *correlated*, we may infer that they are *dependent*, i.e. that their changes have been caused by the same condition parameter X_j. The opposite is not true: if two random variables are not correlated, this does not necessarily mean that they are independent [17].

Strength of correlation between symptoms S_1 and S_2 is often described by the Pearson linear correlation coefficient, i.e. a normalized covariance [17]:

$$r = \frac{E\{(S_1 - \eta_1)(S_2 - \eta_2)\}}{\sqrt{E\{(S_1 - \eta_1)^2\}E\{(S_2 - \eta_2)^2\}}} \quad , \qquad (9)$$

where E denotes expected value and

$$\eta_1 = E(S_1),\ \eta_2 = E(S_2) \quad . \qquad (10)$$

It can be shown that $|r| \leq 1$; $r = 0$ means no correlative relation between S_1 and S_2, while $|r| = 1$ means a functional relation.

A formal question may be asked whether r can be used as a diagnostic symptom. It should be stressed here that we are speaking of a symptom in the broader sense (a function, usually continuous, rather than an event), in accordance with the EP model [2,3]. We may define such symptom as 'a measurable quantity which is covariant with object condition' [5]. Correlation coefficient can be regarded a measure of probability that changes of S_1 and S_2 have been caused by a change of the condition parameter X_j, qualitative relation being determined from the diagnostic model of the object under consideration. We can thus see that r is sensitive to the failure type and conforms to the above diagnostic symptom definition.

According to [10], a measure of correlation between vibration amplitude and an operational parameter is an important diagnostic symptom for steam turbines. This point needs some explanation. In practical applications, correlation (or, more precisely, covariance) between vibration amplitude and turbine load is sometimes used to augment diagnostic reasoning. The same refers to the correlation between vibration amplitude and rotational speed, which can help in identifying malfunctions such as shaft crack or rotor bend; detailed treatment can be found in [9]. In such cases, correlation is usually treated in a purely qualitative manner: investigations are intended to determine whether vibration am-

[3] Exceptions to this rule do exist. If a steam turbine is shifted from base-load to peak-load operation, this usually implies that it will be operated in considerably broader load range, hence values of control vector components will also vary within broader limits. In the following we shall neglect such cases.

plitudes change significantly with operational parameters or not. Some remarks on this issue can be found in [18]. In the following, attention shall be focused on the correlation *between two vibration components*.

3. EXAMPLES

All results dealt with in the following have been obtained with large steam turbines, rated between 200 and 260 MW, operated by utility power plants. These machines differ in details, but all are condensing, base-load units; each consists of a high-pressure (HP), intermediate-pressure (IP) and two-stream low-pressure (LP) section and drives a two-pole generator. Absolute vibration velocity (23% constant percentage bandwidth spectra, 10 kHz frequency range) has been recorded on bearing outer casings and LP turbine casing.

3.1. Turbine fluid-flow system failure

This case has been described in detail in references [19,20]. Here we shall concentrate on issues directly related to correlation analysis.

Turbine T9 suffered an IP rotor failure (crack of several last stage blades) and was shut down. It was decided to remove the entire rotor stage blading and keep the turbine in operation until replacement rotor could be supplied. Immediately afterwards, a significant increase of the $4{\times}f_0$ component in a number of measurement points (mostly in axial direction) was observed. After opening IP turbine casing during overhaul it was found that steam flow guide fences, mounted inside the outlet part of the casing, were loose and cracked and had to be repaired. Detailed analysis revealed that turbine operation without last stage IP rotor blading resulted in substantial increase of guide fences vibration, probably of resonance nature; this was caused by changes of magnitude and distribution of forces imposed by steam flow. Excited vibration, much higher in amplitude than during normal operation, caused rapid damage of the steam flow guide fences and intensified their vibration; this is a good example of the destructive feedback [3].

Interpretation of higher harmonic components in absolute vibration spectra is by no means straightforward. $3{\times}f_0$ and $4{\times}f_0$ components are sometimes associated with faulty machining of rotating elements, coupling malfunctions or rotor cracks [10, 15]. They can also result from resonance, which in turn suggests changes in stiffness and/or damping, caused e.g. by substantial material parameters deterioration. It is thus difficult to point out the reason for a dramatic increase of the $4{\times}f_0$ component, which became dominant in vibration spectra and caused a substantial increase of overall vibration levels. It can, however, be noticed that this component is, to a high degree, correlated with those from the blade frequency range, which are sensitive to the condition of the fluid-flow system [8]. This can be easily seen from scatterplots (see Fig.2). In turbine T9, scatterplot clearly indicates correlation, while in turbine T8 (of the same type) vibration components are obviously not correlated.

At the same time, correlation with the f_0 and $2{\times}f_0$ components is much weaker than in turbines T8 and T10, which suggests that untypical vibration patterns cannot be attributed to phenomena directly related to the rotational motion (see Table 1) and influence of the fluid-flow system failure is dominant.

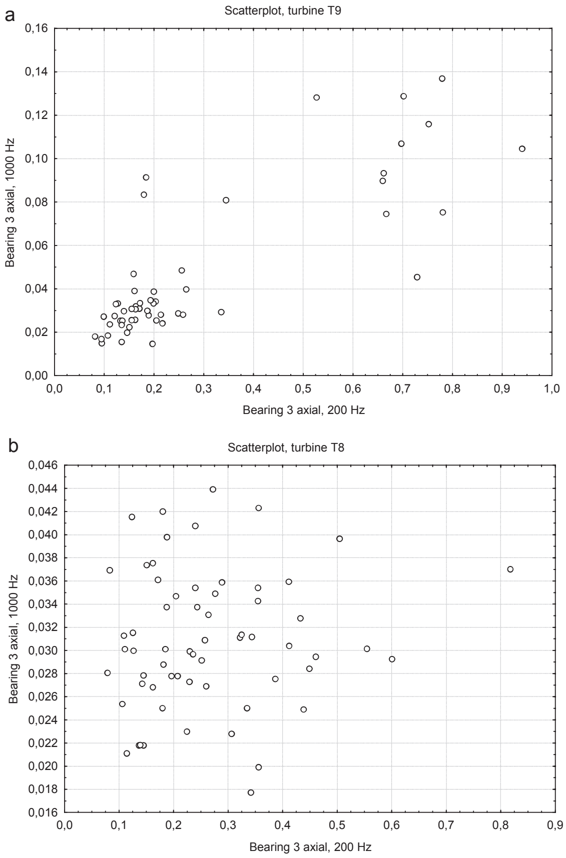


Fig. 2. Scatterplots of 200 Hz vs. 1000 Hz components for turbines T9 (a) and T8 (b); rear IP turbine bearing, axial direction (after [18])

Table 1. Linear correlation coefficients for turbines T8, T9 and T10: rear IP bearing axial (after[18])

Frequency band [Hz]; 23% CPB spectrum	Coefficient of correlation with the $4 \times f_0$ component		
	T8	T9	T10
50	0.19	-0.19	0.77
100	0.23	0.00	-0.35
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

Results for turbine T9 clearly indicate that the $4\times f_0$ and certain blade range components exhibit strong correlation, i.e. behave in a very similar fashion, which suggests that they probably have the same origin, in terms of object technical condition. In fact, for some frequency bands correlation coefficients are only slightly lower than 1. This facilitates a qualitative diagnosis, as many possible (and more typical) causes can be readily eliminated on this basis.

3.2. IP turbine rotor bow

This case has also received some attention in references [16, 18]. Again, issues relevant to correlation analysis shall be dealt with in the following.

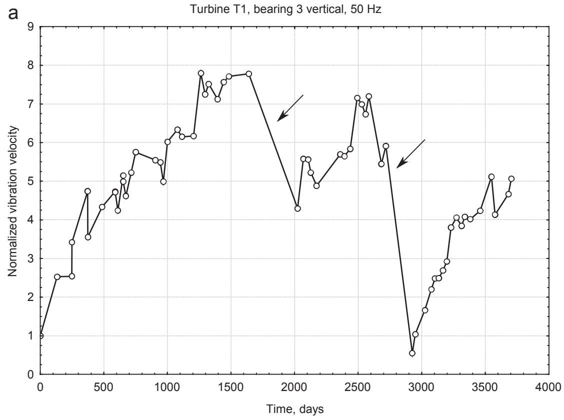
Permanent rotor bow (i.e. involving plastic deformation) is a rare, but serious failure. Repair is costly and time-consuming; in many cases rotor replacement is the only solution. Basically, rotor bow produces symptoms similar to that of a 'plain' unbalance and in fact of many common failure types [9]. Additional tests and measurements are thus required for unambiguous identification. These include vibration monitoring at slow rotational speed [9], monitoring of harmonic components during rundowns in various thermal conditions [21] or phase angle variations at startup [22]. All this requires measurements during transient operation.

IP turbine rotor bow has recently occurred in two steam turbines. Vibration trends shown in Fig.3 reveal that in both cases this has resulted in continuous and rapid increase of the $1\times f_0$ component of vertical vibration velocity, measured at the rear IP bearing. Rotor balancing, intended as a 'symptom treatment' provided in order to keep turbines in operation until next scheduled overhaul, resulted in some improvement (indicated by arrows), but it is evident that the root case has not been eradicated. It can be easily seen that, in each cycle, $S_i(\theta)$ is almost linear and $\partial S_i/\partial\theta$ is hardly influenced by rotor balancing. Moreover, influences of interference and control are much weaker than that of the condition parameter (in this case, unbalance resulting from the rotor bow), so trends are fairly regular.

Application of correlation analysis for distinguishing between rotor bow and other possible faults that produce similar behavior of the $1\times f_0$ component is based on the following reasoning. Normally in a rotating machine with the horizontal shaft line there will always be a rotor sag under its own weight, but resulting rotor shape is maintained during a full rotation [9]. On the other hand, if there is a permanent bow, this shape will change during the period $T = 1/f_0$, so that rotor disks will exhibit a time-dependent tilt in a plane parallel to the rotor axis. For a given disk, the angle α between the disk plane and a vertical plane perpendicular to the rotor axis will behave approximately as [18]

$$\alpha \approx \alpha_s + \alpha_b \sin 2\pi f_0 t \ , \qquad (11)$$

where α_s is the angle resulting from rotor sag due to gravity load and α_b is the amplitude of angle determined by permanent bow. This will result in an additional periodic axial force, with a period of T. We may thus expect axial vibration component with the frequency of f_0 and amplitude increasing with the permanent rotor bow. At the same time, the permanent bow will produce an unbalance with accompanying increase of the f_0 component of vertical vibration. It is therefore justified to suppose that, if this type of fault is present, there will be a correlation of the f_0 components in vertical and axial directions.



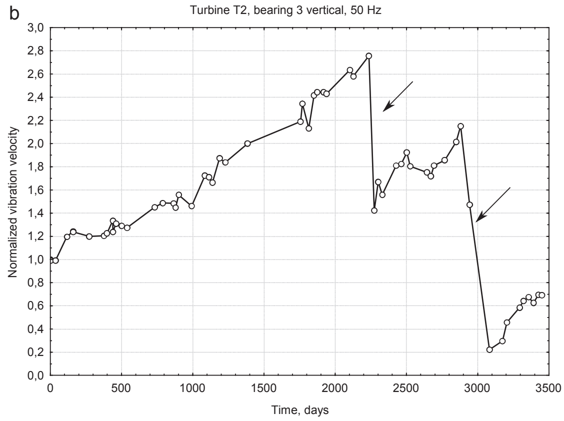


Fig. 3. 50 Hz component of vertical vibration velocity, measured at rear IP rotor bearing in turbines T1 (a) and T2 (b) as a function of time. Values are normalized with respect to those measured for θ = 0. Arrows indicate IP rotor balancing.

Correlation coefficients calculated for turbines T1 (three cycles, determined by rotor balancing attempts – cf. Fig.3a) and T2 (two cycles – the third one in Fig.3b is too short for a meaningful analysis) are listed in Table 2, together with corresponding values for turbine T3 of the same type, in which no permanent bow has occurred. Values are given for rear IP rotor bearing (i.e. the point at which trends shown in Fig.3 have been recorded) and two adjacent bearings (front IP and front LP), as periodic axial force is transmitted along the shaft line. It is clearly seen that for turbines T1 and T2 correlation is very strong, with seven values of r equal to or higher than 0.9. All fifteen values are positive. For turbine T3 correlation is much weaker, maximum value of |*r*| being 0.15; moreover, two values of three are negative, which indicates that an increase of one symptom is accompanied by a decrease of the other. Correlation coefficient thus turns out to be sensitive to the fault type and can be used to provide a reliable qualitative diagnosis. This is accomplished on the basis of steady-state measurement results only.

Table 2. Linear correlation coefficient values for turbines T1, T2 and T3 (see main text for details).

Turbine and cycle	Coefficient of correlation between $1{\times}f_0$ components of vertical vibration velocity at rear IP bearing and axial vibration velocity at		
	front IP bearing	rear IP bearing	front LP bearing
T1, cycle 1	0.69	0.90	0.91
T1, cycle 2	0.94	0.79	0.73
T1, cycle 3	0.87	0.91	0.88
T2, cycle 1	0.97	0.97	0.95
T2, cycle 2	0.65	0.60	0.51
T3	-0.15	0.14	-0.04

3.3. Fluid-flow system condition deterioration

As already mentioned, so-called blade components, usually easily distinguished in absolute vibration spectra, convey information on the fluid-flow system condition; details can be found in references [8,12,23]. Amplitudes of these components, with frequencies usually in the range from a few hundred hertz to 10÷20 kilohertz, can be used as diagnostic symptoms. This is particularly useful in old turbines, for which residual lifetime assessment is often of vital importance.

The main problem in vibration patterns assessment in the blade frequency range is in fact a direct consequence of Eq.(5). As already mentioned, influence of interference and control on vibration components in this range is in most cases much stronger than on harmonic (low-frequency) ones. This is illustrated by Fig.4, which shows normalized standard deviation σ/S_a (S_a is the mean value) as a function of frequency for three different turbines and measuring points. Data have been obtained during sessions lasting about 1.5 h each, so that resulting scatter is due mainly to interference. Similar results have been obtained for other measuring points and other turbine types. It is clearly seen that for the harmonic components σ/S_a is of the order of a few percent, while for the blade components σ/S_a is roughly one order of magnitude higher. This explains why vibration trends pertaining to this frequency range are often very irregular (cf. Fig.1).

Moreover, in certain measuring points, there are vibration sources other than turbine fluid-flow system. This refers mainly to front HP bearing, which in typical large steam turbines is usually housed in a casing together with main oil pump drive, speed governor and several other drives, which generate their own characteristic vibration patterns. Some of these components fall within the blade frequency range and, if spectral resolution is not high enough to distinguish them, which is often the case, interpretation of vibration patterns and trends can be vague.

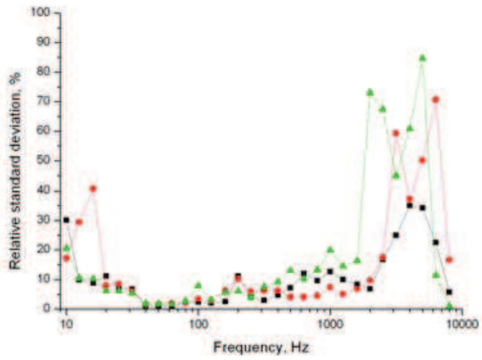


Fig. 4. σ/S_a as a function of frequency, determined from 23% CPB vibration velocity spectra; (squares) K-200 turbine, front LP bearing axial, 58 measurements; (circles) K-200 turbine, front HP bearing horizontal, 55 measurements; (triangles) TK-120 turbine, rear part of the LP casing horizontal, 90 measurements

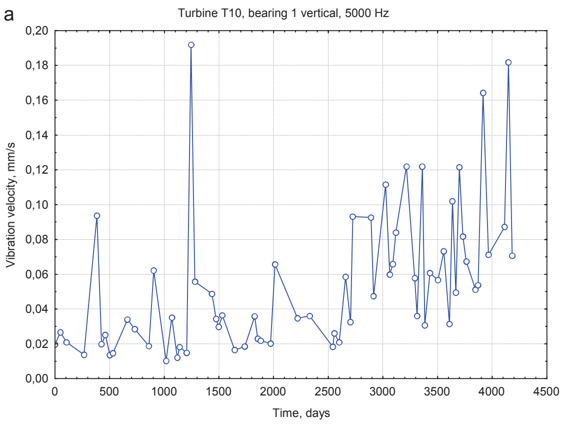
Can correlation analysis be useful in such cases? It seems reasonable to assume that technical condition deterioration affects the rotor as a whole[4], although degradation rates will of course be different for individual stages. Thus, if there is a *positive* correlation between vibration amplitudes in spectral bands that contain vibration generated by the fluid-flow system, we can infer that the increasing trend, if present, can be attributed to its technical condition deterioration.

Correlation analysis has been performed for the front HP bearing of turbine T10, which was commissioned in early 1960s and modernized in early 1990s (modernization concerned mainly LP turbine). Two examples of vibration trends, recorded at this point, are shown in Fig.5. Accelerated deterioration of fluid-flow system condition can be suspected, as there is a marked increase and noticeable departure from linearity in $S_i(\theta)$ histories [12,16]. Considerable fluctuations, due to the influences of control and interference, are noteworthy; that of interference is probably dominant, as the turbine in most cases ran at 90 to 100 % of the rated power and in this range load influence is comparatively small [11]. On the basis of these (and also other) vibration trends it is not possible, however, to confirm this conjecture.

In this case there are ten frequency bands in a CPB spectrum that contain blade components, and hence ten symptoms for each measuring direction.

[4] Of course this refers to 'natural damage' only. With random failures, caused e.g. by foreign objects, this is in general not the case.

Correlation data seem, at a glance, inconclusive. Generally correlation between blade components is not strong and many values are negative (20, 27 and 20 of 45 in vertical, horizontal and axial directions, respectively). If we, however, distinguish those exceeding +0.5 (which is obviously an arbitrary limit), we immediately notice that they fall into two groups: the first one comprises three components (1.6 kHz, 2 kHz and 2.5 kHz) and the second two (5 kHz and 6.3 kHz). Correlation *between* these groups is weak and almost all values (16 of 18) are negative.



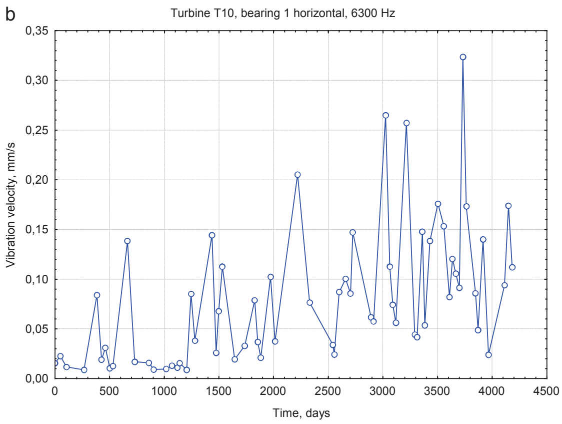


Fig. 5. Vibration velocity trends, recorded at the front HP bearing of turbine T10; a) 5 kHz band, vertical direction; b) 6.3 kHz band, horizontal direction

For this particular turbine type, frequency bands up to 3150 Hz contain components generated by bladed diaphragms, while higher frequency bands contain those generated by rotor stages. It may thus be inferred that both bladed diaphragms and rotor stages exhibit symptoms of substantial lifetime consumption degree, but quantitatively behave in different manners. In general, correlation is stronger for rotor stages, which suggests that their deterioration is more advanced; this is, however, just a conjecture, as quantitative relations still remain to be established.

It may be argued that in all above cases correlation coefficient is treated as a two-value (binary) symptom: in fact the question is whether there is a correlation or not. It seems natural to check how this coefficient changes with time (or, more precisely, with D). In order to deal with this issue, it is useful to recall the so-called 'old man syndrome': as θ_b is approached, all condition symptoms become more and more correlated [24]. This can be understood on the basis of Eq. (5) if we assume – in accordance with suggestions put forward e.g. in [25] – that D should be treated as a condition parameter, i.e. a component of the $\mathbf{X}(\theta)$ vector. For small values of D each symptom is dominated by specific condition parameters (or control and/or interference), so correlation between individual symptoms is expected to be weak. If, however, D is large, or θ is close to θ_b, its influence on all symptoms becomes dominant; note that, according to the EP model, each S_i is a monotonically increasing function of D[5]. This implies that, for a pair of symptoms,

$$\theta \rightarrow \theta_b \;\Rightarrow\; r \rightarrow 1 \;. \tag{12}$$

Can this syndrome be observed? Fig. 6 refers to turbine P4 similar to T10 dealt with earlier in this section, in which HP rotor and casing (together with bladed diaphragms) were replaced after about 220,000 hours in operation; last measurements were performed immediately before replacement. A marked increase of vibration velocity amplitudes in frequency bands containing components generated by rotor stages can be observed (Fig. 6a). On the other hand, those that contain components generated by bladed diaphragms show no distinguishable monotonic trend (Fig. 6b). In fact, trend shown in Fig. 6a is qualitatively similar to that shown in Fig. 5 – no wonder, as a similar phenomenon of the HP turbine fluid-flow system lifetime consumption is observed.

Correlation coefficients, shown as functions of time in Figs. 6c and 6d, have been calculated in such a manner that, for a given point on the time axis, they refer to ten preceding measurements. Fig. 6c refers to two velocity amplitudes from the blade frequency range. Initially correlation between them is quite weak and r is negative. Then a fast increase is observed, up to r = 0.82, followed by a decrease. Last section of the $r(\theta)$ curve exhibits a monotonic increase, with r = 0.95 shortly before the overhaul; the length of this increase period is about three years. Detailed analysis has revealed that after about 1500 days several components of the HP turbine (inlet nozzles and several bladed diaphragms clamping rings) were replaced. This is most probably responsible for the temporary correlation strength decrease. Vital fluid-flow system elements, however, were not affected, so after a comparatively short period the increasing tendency reappeared.

Fig. 6d, on the other hands, shows that correlation between components generated by rotor stages and bladed diaphragms is weak. Time history

[5] It has to be kept in mind that s_{ij} values (see Eq. (8)) usually increase with D, which intensifies the effect.

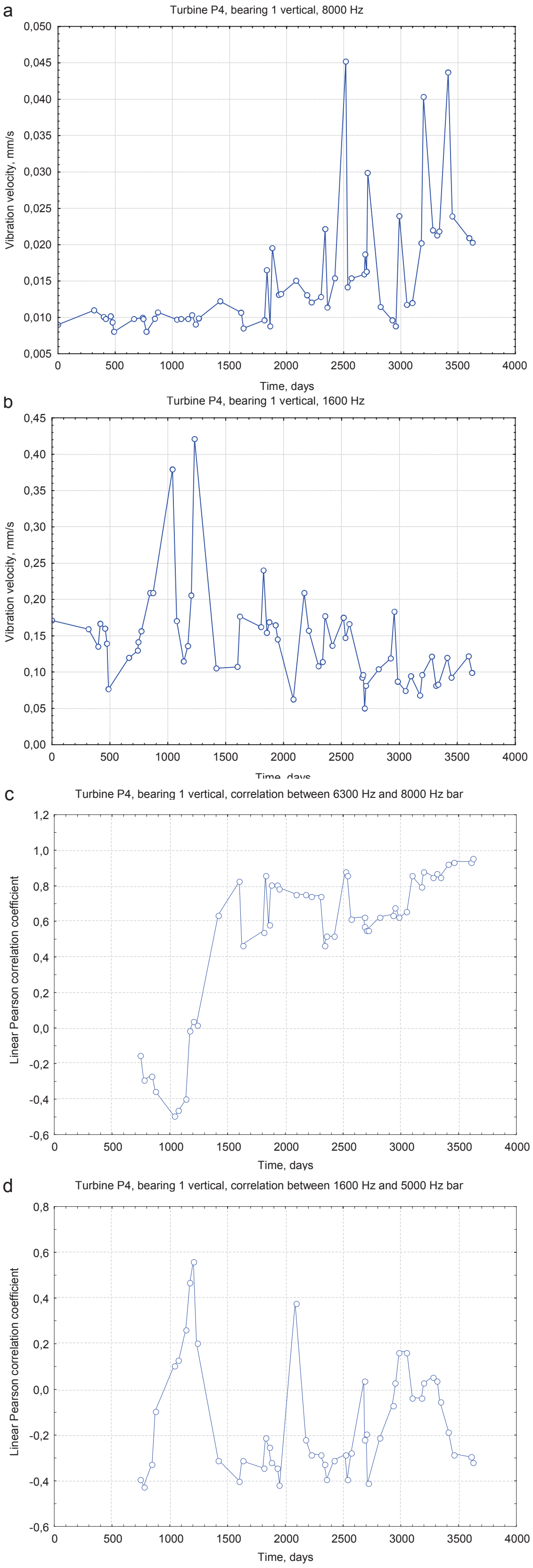


Fig. 6. Vibration velocity amplitudes and correlation coefficients vs. time; turbine P4, front HP bearing vertical. a) vibration, 8 kHz band; b) vibration, 1.6 kHz band; c) correlation between 6.3 and 8 kHz bands; d) correlation between 1.6 and 5 kHz bands

$r(\theta)$ is very irregular and most values fall within the range from r = -0.4 to r = 0.2. This confirms the observations already mentioned in this section that rotor stages and bladed diaphragms behave in different manners and that rotor condition deterioration is more advanced. It may be added here that, for rotor stages, correlation between results obtained for different measuring directions is weaker, but the 'old man syndrome' can nevertheless be observed; two examples are shown in Fig. 7.

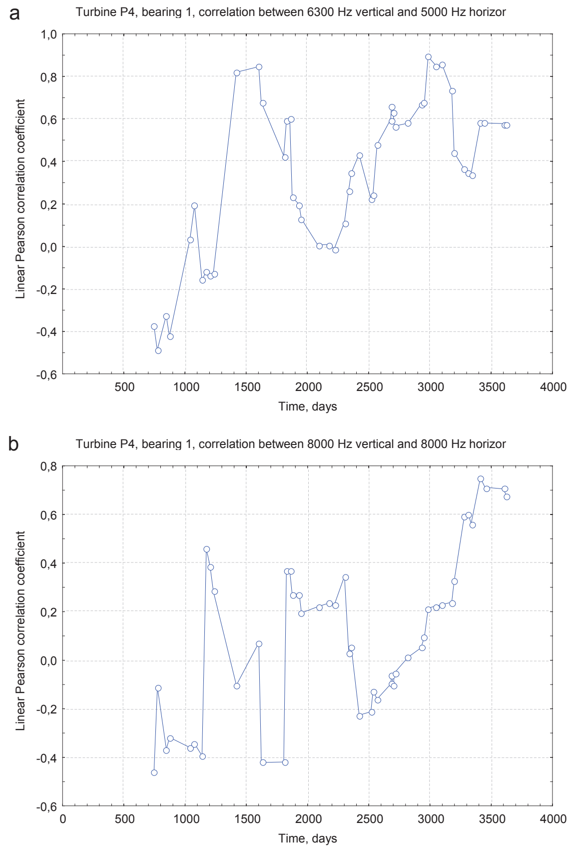


Fig. 7. Correlation coefficients vs. time; turbine P4, front HP bearing. a) correlation between frequency bands 6.3 kHz (vertical) and 5 kHz (horizontal); b) correlation between frequency bands 8 kHz (vertical) and 8 kHz (horizontal)

It should be kept in mind that the above results have been obtained from databases recorded during routine measurements rather than by means of a purpose-designed diagnostic experiment. This is probably responsible for irregularities of the $r(\theta)$ time histories, at least to a certain extent. Similarly, the method employed for determining r as a function of time is responsible for the lag in the correlation coefficient decrease after an overhaul. The potential of the method is, however, clearly demonstrated.

4. CONCLUSION

Vibration-based symptoms, in particular absolute vibration amplitudes in certain frequency bands and their time histories, are widely employed in the diagnostics of rotating machines. It has been shown that correlation between such symptoms contains valuable information on machine technical condition and hence a measure of this correlation can itself be employed as a symptom.

In qualitative diagnostics, presence (or absence) of correlation is indicative of the possible malfunction or damage type and therefore can augment diagnostic capabilities in dubious cases. This is particularly useful in the harmonic (low) frequency range, wherein several phenomena have very similar representations in vibration characteristics. In such approach, correlation can be even treated as a binary symptom.

In quantitative diagnostics, analysis of correlation coefficient as a function of time can reveal the 'old man syndrome', which is indicative of a substantial lifetime consumption degree. This seems particularly important for machines operated beyond their design lifetime (which is by no means an exceptional practice), where residual life assessment becomes vital. In such approach, correlation coefficient is treated as a continuous function of time or of generalized damage.

Examples presented in this paper reveal a substantial potential of diagnostic methods based on correlation analysis. Further research in this field is therefore justified.

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