

## **INFLUENCE OF LOAD AND INTERFERENCE IN VIBRATION-BASED DIAGNOSTICS OF ROTATING MACHINES**

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### **Abstract**

The paper deals with the influence of machine load and interference on symptom values in rotating machinery diagnostics. In a typical industrial plant environment, this influence often results in very irregular symptom time histories. A theoretical treatment, based on suitable models, is presented and briefly discussed. cursory analysis of available data shows that, in general, influence of load and interference cannot be neglected. It is therefore important to obtain some knowledge of its relative magnitude, as it directly affects diagnostic reasoning.

The paper presents some results obtained with available databases for large utility steam turbines. Irrespective of turbine type, these data reveal considerable qualitative similarity. On their basis, it may be concluded that in the harmonic frequency range influences of load and interference may often be considered negligible. Observed symptom time histories are thus determined mainly by machine condition evolution. In the blade frequency range, these influences are much stronger and produce considerable fluctuations. They have to be taken into account in order to arrive at a correct machine condition estimation.

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## 1. Introduction

Vibration-related condition symptoms<sup>1</sup> are perhaps the most important ones in diagnostics of rotating machines. This results not only from their high information content, but also from ease of measurement, wide range of commercially available measuring equipment, as well as comparatively well-developed data acquisition and assessment procedures [2, 12, 17]. In principle, a new machine (assumed to be in ‘good’ technical condition, i.e., with no malfunctions present and with lifetime consumption<sup>2</sup> equal to zero) is characterized by a set of reference or basic values of condition symptoms. A stepwise change (typically an increase) of one or more symptoms is indicative of a random or ‘hard’ damage, while their continuous monotonic increase is a result of natural or ‘soft’ damage [16].

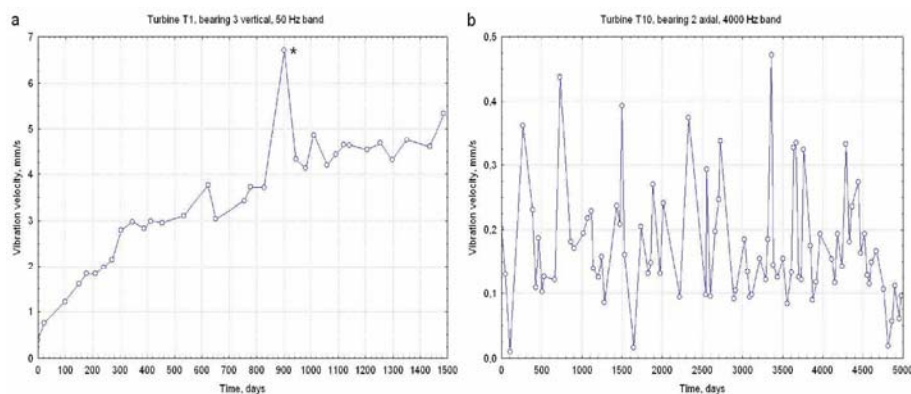
In an ideal case, a symptom time history is a continuous increasing curve that follows lifetime consumption processes. Such situation is, however, seldom encountered in practice, especially when dealing with more complex machines. What is typically obtained may be described as a continuous curve with superimposed fluctuations [9]. Relative magnitudes of the ‘natural’ increase rate and amplitudes of these fluctuations are a key factor in the symptom time history analysis. If damage is of a type that strongly influences vibration patterns and its development is fast, diagnostic reasoning and even forecasting can be fairly straightforward (Figure 1a). Much more often, however, fluctuations dominate and a monotonic trend is hardly visible in raw symptom time histories (Figure 1b). Certain smoothing procedures may be useful in such situations (see, e.g., [6, 13]), but this issue is beyond the scope of this paper.

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<sup>1</sup> To avoid any misunderstanding, symptom is defined here as a ‘measurable quantity covariant with object condition’ (see, e.g., [4]).

<sup>2</sup> Within the framework of the basic energy processor model (see, e.g., [18]), lifetime consumption is given by the ratio  $D \equiv \theta / \theta_b$ , where  $\theta$  denotes time and  $\theta_b$  is time to breakdown.

If a sudden change of the symptom value is observed, it is always necessary to determine whether it has been caused by a random damage or some other factor, other than technical condition parameters, that influences measurable symptoms. This issue shall be dealt with in more details in the next section. At this point, however, it seems necessary to point out that, for a given machine in its specific environment, such factors can be qualitatively identified, but quantitative assessment is a different issue. In the author's opinion, this problem has attracted comparatively little attention.



**Figure 1.** Examples of vibration time histories; (a) 13CK230 unit, rear intermediate-pressure turbine bearing, vertical direction, 50Hz band; (b) K-200 unit, rear high-pressure turbine bearing, axial direction, 4000Hz band. Asterisk in (a) labels a measurement taken at about 76% of rated load, a few hours after a cold startup.

This paper is intended to throw some light on the *quantitative* influence of machine load and interference on measured values of vibration-based symptoms. In the following, these symptoms are understood as absolute vibration velocity levels in 23% CPB (constant-percentage bandwidth) spectral bands, but this has been adopted due to the available database rather than any fundamental reason and does not affect the general nature of the considerations. It has to be kept in mind that routine vibration monitoring typically employs only ‘total’ values, averaged over some frequency range. For that reason, a sudden change of a vibration component amplitude can pass unnoticed and is detected, if ever, only *post factum*, when spectra obtained from off-line measurements are analyzed.

All examples presented in the following refer to utility condensing steam turbines that are typical large critical rotating machines. It seems, however, justified to assume that many conclusions shall apply to a broader class of rotating machines or even diagnostic objects in general.

## 2. Basic Considerations

It is sometimes convenient to start from a very general relation. In technical diagnostics, this is probably given by [19]

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

where  $\mathbf{S}$ ,  $\mathbf{X}$ ,  $\mathbf{R}$ , and  $\mathbf{Z}$  denote vectors of diagnostic symptoms, condition parameters, control, and interference, respectively, all changing with time  $\theta$ , and  $\Phi$  denotes an operator. This relation just indicates the factors that influence a measured symptom value, but provides no information on the mechanisms involved.

Theoretical time dependence of a diagnostic symptom  $S_i(\theta)$ ,  $S_i \in \mathbf{S}$ , can be derived from the general energy processor (EP) model (for more details see, e.g., [3, 18]). In its basic form, the EP model is suitable only for simple objects, wherein observable symptoms depend solely on the generalized damage  $D$ . With certain modifications, this model is capable of including also the influences of control parameters and interference [14]. However, due to its accumulative nature, this requires the  $R_i$  and  $Z_i$  components be given as defined functions of time. For most practical applications, this is hardly acceptable; in particular, interference vector components must be treated as random variables.

If we limit our attention to vibroacoustic symptoms, we may recall the relation given in [21]

$$z(r, t) = h_p(r, t) * u_w(r, t) + \eta(r, t), \quad (2)$$

where  $z$  is the measured signal,  $h_p$  denotes response function for signal propagation between the source and the measuring point,  $\eta$  is the uncorrelated noise; all these quantities depend on time  $t$  and spatial variable  $r$ .  $u_w$  is given by the relation

$$u_w(r, t) = \sum_{i=1}^n h_i(r, D_i, t) * x(t) + h(r, t) * x(t), \quad (3)$$

where  $D_i$  is a dissipated variable that describes the development of the  $i$ -th defect,  $h_i$  is the response function associated with this defect,  $h$  is the response function for the object in ‘perfect’ condition (no defect present), and  $x(t)$  is the input signal, generated by the elementary vibroacoustic signal source.  $t$  denotes the ‘dynamic’ time

$$S_i(\theta) = E_t\{S(\theta, t)\}, \quad (4)$$

where  $E_t\{\cdot\}$  is the operator of averaging over  $t$  (see, e.g., [20]). Combining Equations (2) and (3) yields

$$z(r, t) = \sum_{i=1}^n h_p(r, t) * V_i(r, D_i, t) + C(r, t) + \eta(r, t), \quad (5)$$

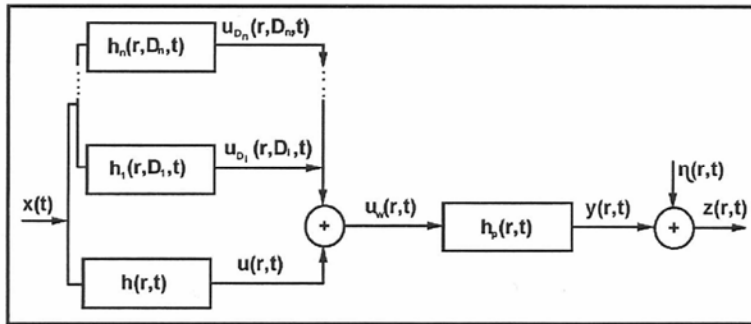
where

$$V_i(r, D_i, t) = h_i(r, D_i, t) * x(t) \quad (6)$$

represents the signal generated in the presence of the  $i$ -th defect, and

$$C(r, t) = h_p(r, t) * h(r, t) * x(t) \quad (7)$$

is the signal generated with no defects present. This model is shown schematically in Figure 2.



**Figure 2.** The model of vibroacoustic signal generation and propagation (after [21]).

Let us now compare Equations (1) and (5). For a given symptom  $S_i$ , influence of the  $\mathbf{R}(\theta)$  components results from the fact that they affect the input signal  $x(t)$  and/or the function describing signal propagation  $h_p(r, t)$ . Influence of the interference is represented by  $\eta$ . This does not, however, imply that interference should be identified as a ‘noise’ in its physical sense. The  $\mathbf{Z}(\theta)$  vector includes also, e.g., factors resulting from imperfect repeatability of measurement condition. It has to be kept in mind that, interference may be measurable or non-measurable.

For a given object and symptom, identification of all variables that enter Equation (5) is hardly possible. Thus, although this equation provides a *quantitative* (albeit simplified) description, it cannot form a basis for a *quantitative* assessment. The latter has to be based on experimental data.

Influence of control parameters on vibration patterns has attracted some attention (see, e.g., [5, 8]). Certain normalization procedures were also proposed [8]; for any real object, however, they can be only approximate. They are nonetheless useful, e.g., in symptom limit value estimation procedures. The problem of interference has not, to the author’s best knowledge, been studied in a comprehensive manner. It seems that much more attention has been devoted to how minimize interference influence than how to describe it.

### 3. Influence of Load

Control parameters may be defined as resulting from object operator purposeful action, aimed at obtaining demanded performance. At first approximation, they may be identified as ‘control system settings’. In many cases, the ‘demanded performance’ means demanded output power; machine load can thus be treated as a scalar measure of the vector  $\mathbf{R}$ .

In the following, attention shall be focused on condensing steam turbines<sup>3</sup>. Turbine load or active power  $P$  is, in fact, a function of a number of parameters. In general, we have [22]

$$P = (dm / dt)\Delta i\eta, \quad (8)$$

where  $dm / dt$  denotes steam mass flow,  $\Delta i$  is the enthalpy drop, and  $\eta$  is the turbine efficiency. Assuming that  $\eta$  remains constant, changes of  $P$  may be achieved by changing  $\Delta i$  (qualitative control),  $dm / dt$  (quantitative control), or both. The latter method (so-called group or nozzle control) is typically used in large steam turbines. Each control valve supplies steam to its own control stage section; the number of these valves is usually from three to six and they are opened in a specific sequence. At the rated power, the last valve is typically only partly open, or even almost closed, as it provides a reserve in a case of steam parameters drop (due, e.g., to boiler problems). Moreover,  $\Delta i$  depends also on condenser vacuum, which changes within certain limits depending on overall condenser condition, cooling water temperature, weather etc.. Thus,

$$P = f(r_1, r_2, \dots, r_k, p_o), \quad (9)$$

where  $r_i$  denotes  $i$ -th valve opening,  $k$  is the number of valves, and  $p_o$  is the condenser pressure. In fact,  $r_i$  and  $p_o$  are the control parameters vector components.

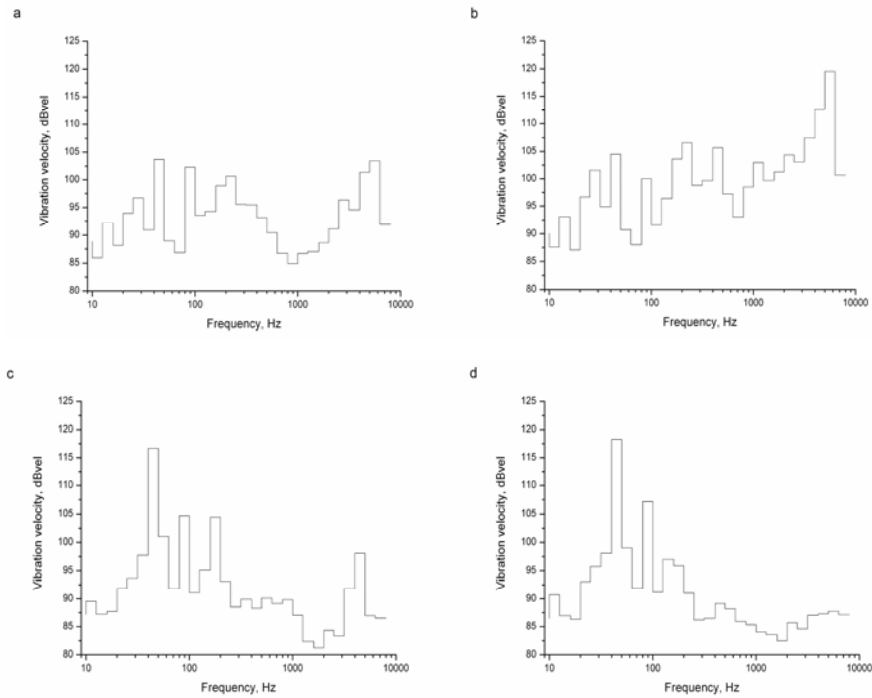
Influence of turbine power on vibration patterns can be seen as a result of competition between two phenomena. At higher power, the steam thrust is more evenly distributed over the entire fluid-flow system cross-section. Steam flow thus becomes more uniform, which is expected to reduce forces and hence vibration levels. This has been confirmed by model simulations [15]. On the other hand, increasing mass flow involves higher total thrust and higher forces, with vibration levels increasing

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<sup>3</sup> In extraction and counter-pressure turbines, the aim of the control system is usually to maintain desired pressure, temperature, and flow rate of extraction or outlet steam, so that the following considerations do not apply. In such cases, load can no longer be accepted as a scalar measure of the control parameters vector.

accordingly. Relative influence of these phenomena depends on the turbine stage. For initial stages, immediately after the control stage, influence of non-uniformity will be comparatively strong, but its impact will dwindle as we move along steam flow path. The influence of mass flow (as well as of condenser vacuum) will in turn dominate at terminal low-pressure turbine stages, with long blades and large cross-sections.

These effects can be seen in the example shown in Figure 3. Four absolute vibration velocity spectra have been recorded at a 13CK230 turbine (nominal rating of 230MW) for two load values, namely, 227MW (Figures 3a and 3c) and 120MW (Figures 3b and 3d). This turbine is fitted with three control valves. At the higher load, valve A is fully opened, valve C is opened by about 85%, and valve B is almost closed (below 10%).



**Figure 3.** Comparison of vibration velocity spectra: 13CK230 unit, active power of 227MW (a, c) and 120MW (b, d); rear high-pressure turbine bearing, vertical direction (a, b) and low-pressure turbine casing, front part, vertical direction (c, d). See main text for details.

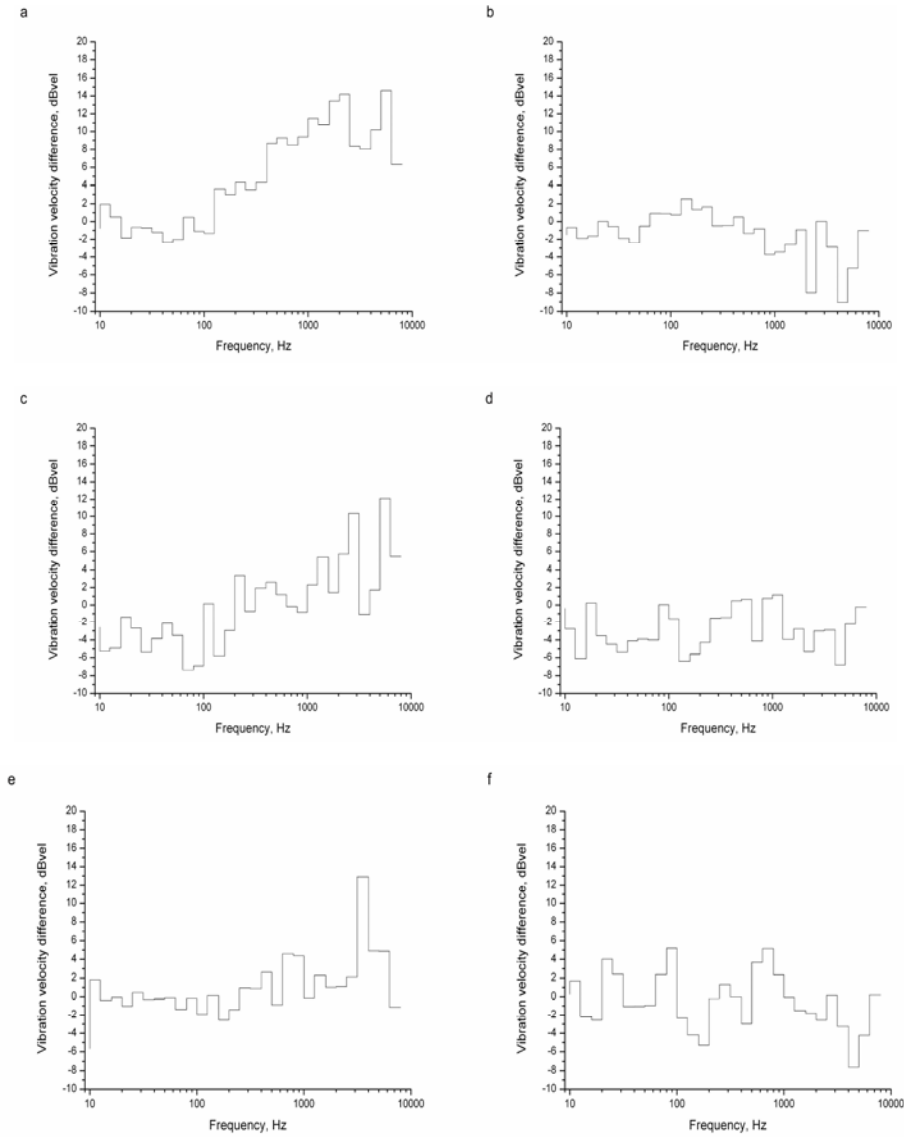


At the lower load, valve A is opened by about 50% and valves B and C are closed. The result is clearly seen in spectra recorded at the rear high-pressure turbine bearing, which is close to the steam inlet and control stage (Figures 3a and 3b). In the high frequency range, that contains vibration components generated by the fluid-flow system, operation at reduced load causes an increase in all frequency bands, by approximately six to nineteen dB - the latter corresponds to almost one order of magnitude. Similar effect, albeit less pronounced, can be seen at the front high-pressure turbine bearing. As for other measuring points, the situation shown in Figures 3c and 3d is typical: Lower load results in a decrease, but the difference is significantly smaller, about eight dB at most. We may thus infer that the effect of non-uniformity is stronger. Similar results have been obtained for other turbine types.

Spectra shown in Figure 3 indicate that significant differences between two load conditions can be observed mainly in the high ('blade') frequency range, roughly above 1kHz. This is easily understood, as both above-mentioned competing phenomena pertain to vibration components generated as a result of interaction between turbine fluid-flow system and steam flow (for more details see, e.g., [19]). In the low ('harmonic') frequency range, differences are typically smaller and often negligible. This is shown by differential spectra, obtained for three different turbine types, some of which are presented in Figure 4. Nominal load condition has been taken as a reference in all cases; in order to minimize the interference impact, five spectra obtained at  $P \cong P_{\text{nom}}$  have been averaged. Results of this averaging have been compared with spectra recorded at low loads, namely;

- 13CK230 turbine (as in Figure 3): 120MW;
- 16K260 turbine (nominal rating of 260MW, four control valves): 105MW;
- K-200 turbine (nominal rating of 200MW, four control valves): 140MW.

It should be noted here that, in the latter case, 140MW is the lowest load that can be continuously maintained, due to boiler operation stability. By convention, positive difference means higher level at the low load.



**Figure 4.** Differential vibration velocity spectra: 13CK230 unit, rear high-pressure turbine bearing, axial direction (a) and front low-pressure turbine bearing, vertical direction (b); 16K260 unit, front high-pressure turbine bearing, horizontal direction (c) and rear low-pressure turbine bearing, axial direction (d); K-200 unit, rear high-pressure turbine bearing, vertical direction (e) and rear intermediate-pressure turbine bearing, horizontal direction (f); see main text for details.

Spectra shown in Figure 4 in general confirm the above observations. In particular, differences observed in measuring points located at the high-pressure turbine are much larger and mostly positive, while that in points located at the intermediate - and low-pressure turbine are significantly less pronounced. In the low frequency range, all differences are comparatively small. In principle, at low loads forces resulting from steam flow non-uniformity can ‘push’ the rotor slightly upwards or downwards, thus influencing harmonic and sub-harmonic components, but this effect is usually quite weak.

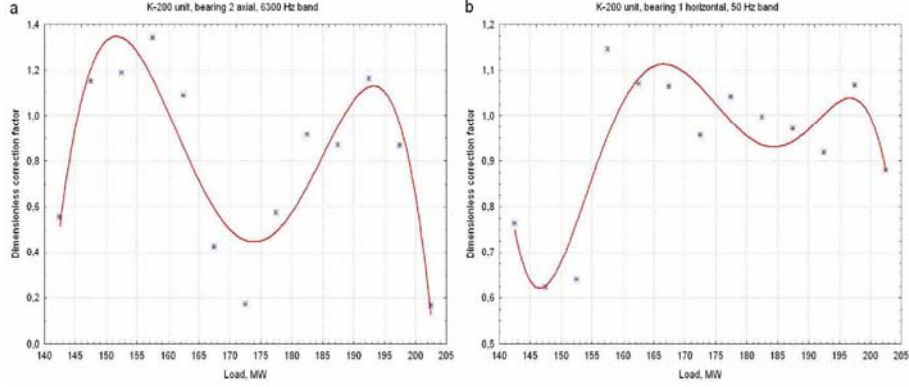
At this point, at least three remarks are necessary.

First, while evaluating results obtained at low loads, care must be taken to assure that they refer to a stable thermal condition. Typically shortly after a cold startup the turbine is kept at a low load level for some time - even a few hours - in order to stabilize thermal fields, especially in thick-walled elements. Such condition may strongly affect vibration patterns. The asterisk in Figure 1a labels a measurement taken at about 76% of the rated load, a few hours after a cold startup. As already noted, such dramatic increase in the low frequency range should by no means be attributed solely to the influence of load. It should be noted that all results used to produce differential spectra shown in Figure 4 have been obtained in stable thermal conditions - in large steam turbines, this means at least a few days after a startup.

Second, it has to be kept in mind that  $S_i(P)$  functions are typically strongly non-linear. This issue has been dealt with by the author in several earlier publications (see, e.g., [7, 8]). As it will be explained in the next paragraph, such functions, estimated from experimental databases, can be only approximate. Their non-linearity is nonetheless evident. In principle, symptom value  $S_i$  measured at an arbitrary load  $P$  can be ‘normalized’, so that a value corresponding to the rated load  $P_n$  could be obtained

$$S_i(P_n) = S_i(P) \cdot c(P). \quad (10)$$

Two examples of  $c(P)$  functions, obtained by fitting fifth-order polynomials to experimental data, are shown in Figure 5 (details of the procedure can be found in [7]). It is easily seen that, over the entire load range, any linearization is out of question.



**Figure 5.** Dimensionless correction factor  $c$  as a function of turbine load: K-200 unit, rear high-pressure bearing, axial direction, 6.3kHz band (a) and front high-pressure bearing, horizontal direction, 50Hz band (b). Asterisks show calculation results and continuous line is the fifth-order polynomial fit.

Finally, let us now recall the basic relation given by Equation (1) and consider any specific symptom  $S_i$ . If the period under consideration is sufficiently small to assume that  $\mathbf{X}(\theta) = \text{const.}$  and the influence of  $\mathbf{Z}(\theta)$  is neglected, we obtain

$$S_i = \Phi(\mathbf{R}) = f(r_1, r_2, \dots, r_k, p_o). \quad (11)$$

Combining Equations (9) and (11), we immediately see that  $S_i$  and  $P$  are functions of the same variables and, in principle, any specific value of  $P$  may be attained with different values of control parameters. Vibration patterns are thus not influenced by  $P$  itself, and  $S(P)$  in general cannot be a single-valued function. Thus, as already mentioned, any relation of the type given by Equation (10) can be only approximate.

#### 4. Influence of Interference

In general, two types of interference can be distinguished: *External interference* (the source is outside the object under consideration) and *internal interference* (the source is within the object). It may be noted that within the framework of the EP model, we may speak in terms of the external interference only. In order to facilitate any mathematical

description, interference must thus be viewed as some additional energy supplied to the object. Formally, it is possible to introduce some  $Z(\theta)$  as a scalar measure of the  $\mathbf{Z}(\theta)$  vector. However, as already mentioned, the accumulative nature of the EP models requires that  $Z(\theta)$  must be given as a deterministic function of time.

External interference may also influence  $S_i$  directly. This means that the basic EP model equation, which for this specific case may be written as [18]

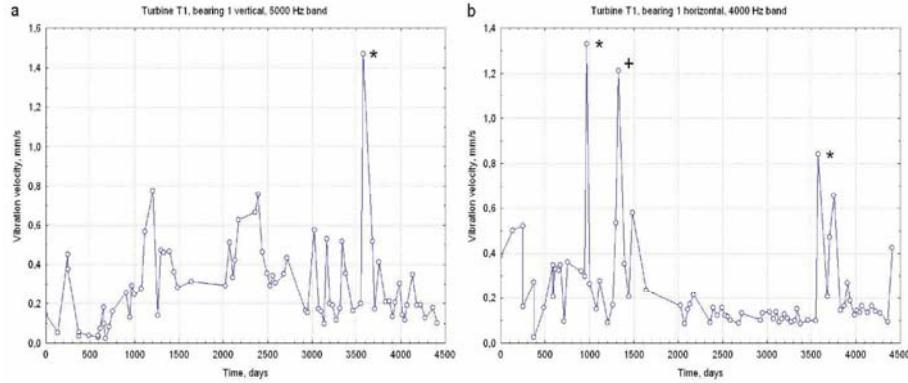
$$S_i(\theta) = \Phi_i[V(\theta)], \quad (12)$$

where  $V$  is the power of residual processes and  $\Phi_i$  denotes the symptom operator, takes the form of

$$S_i(\theta) = \Phi_i[V(\theta)] + f[\mathbf{Z}(\theta)]. \quad (13)$$

This in turn implies that the influence of interference is identified with the measurement error. Obviously, this error is not related to the object condition parameters and thus is not a function of  $\theta$  or  $D$ , unless the measurement method and/or equipment has been changed during the object life. It is worth mentioning here that, relations of the type given by Equation (13) have been introduced at a comparatively early stage of vibration-based diagnostics development, which is perhaps indicative of the external interference being seen basically as the measurement error.

In many cases, interference is intuitively suspected of having caused an abrupt change of vibration patterns, which is manifested in symptom time histories by 'peaks'. An example is provided by Figure 6, which refers to the front low-pressure turbine bearing of a 13CK230 unit. The first time history (Figure 6a-vertical direction, 5kHz band) reveals a single peak, labeled with an asterisk, which corresponds to operation at 94MW-the lowest load attainable. The second one (Figure 6b-horizontal direction, 4kHz band) also reveals a peak at this load, but two more are present. The first of these, also labelled with an asterisk, corresponds to the load of 115MW, but the second, labelled with the '+' sign, has occurred at 220MW, only slightly below the rated load. Most probably, it has been caused by an interference - in this case, a steam flow instability.

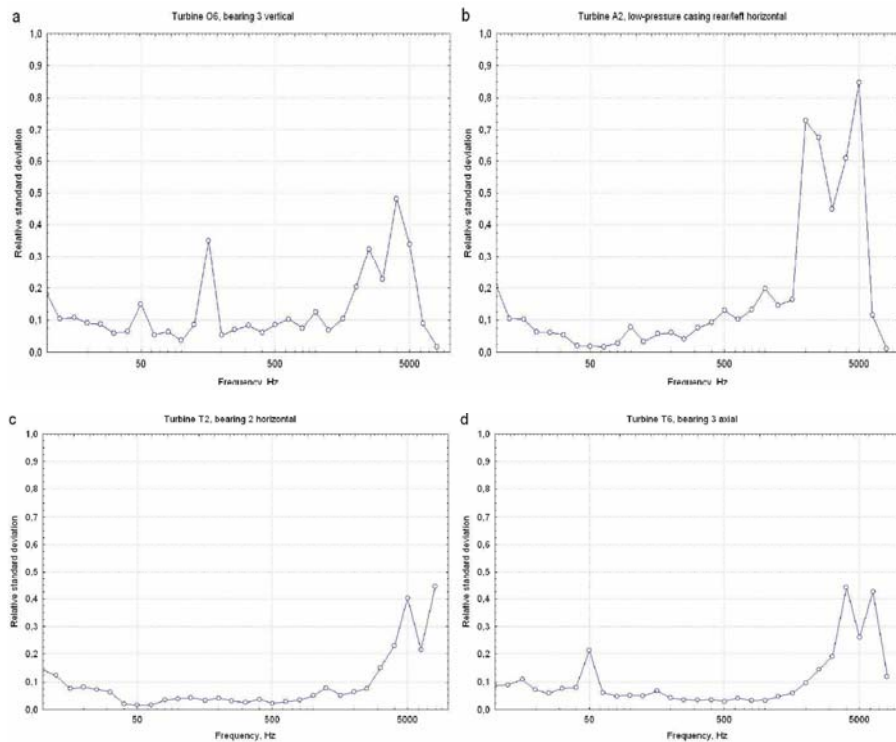


**Figure 6.** Vibration time histories recorded at the front high-pressure turbine bearing of a 13CK203 unit: (a) vertical direction, 5kHz band; (b) horizontal direction, 4kHz band (see main text for details).

Time histories of the type shown in Figure 6 are, in fact, typical for the blade frequency range: A slow, often almost unnoticeable evolution is observed, with superimposed peaks or fluctuations. Figure 6 also suggests that amplitudes of these peaks can be very high. On the other hand, as seen in Figure 1, such occurrences are seldom observed in the harmonic (low) frequency range, wherein time histories are usually dominated by technical condition evolution rather than interference. A question therefore arises how a quantitative measure of the interference influence depends on frequency.

Such measure may be provided by relative standard deviation  $\sigma_r = \sigma / E(S)$ , where  $\sigma$  is the standard deviation and  $E(S)$  is the symptom expected value. Both  $\sigma$  and  $E(S)$ - or, more precisely,  $\sigma$  and mean value  $S_{av}$  - are obtained from databases obtained in such a way that the influence of load is minimized. Figure 7 shows the  $\sigma_r(f)$  functions, estimated for four turbines and different measuring points. In all four cases, measurements took about 1 to 1.5 hours, with negligible load variations. Figures 7c and 7d refer to turbine types already mentioned in Section 3. Figure 7a has been obtained from data recorded at a 25MW *Lang* turbine (an old unit, commissioned in mid-1950s), while Figure 7b refers to a 120MW TK-120 unit (*AEI* design, license-built in Poland in

early 1960s). It can be easily seen that, despite different machine types and measurement points, all four plots are qualitatively similar in that there is a significant increase in the blade frequency range (roughly above 1kHz). It should be added here that the peak at 200Hz (4th harmonic component) in Figure 7a is probably a result of resonance condition. In the low (harmonic) range,  $\sigma_r$  is typically below 0.2, while in the blade frequency range, it can be as high as 0.7 to 0.8. Similar results have been obtained with other turbines and measurement points.



**Figure 7.** Relative standard deviation vs. frequency: (a) Lang 25MW unit, front low-pressure turbine bearing, vertical direction, 56 measurements; (b) TK-120 unit, low-pressure casing rear, left side, horizontal direction, 90 measurements; (c) 13CK230 unit, rear high-pressure turbine bearing, horizontal direction, 56 measurements; and (d) 16CK260 unit, front low-pressure turbine bearing axial, 61 measurements.

## 5. Summary and Conclusion

In principle, components of both interference and control parameters vectors (or machine load, which may be considered a scalar measure of the latter) should be treated as random variables. This means that speaking in terms of symptom values, and diagnostic reasoning on their basis, inevitably involves an approximation: A symptom is, in fact, also a random variable. A question immediately arises when the above approximation can be considered valid.

Figures 4 and 7 in fact make an interesting comparison.

The former shows that, in the harmonic (low) frequency range, differences caused by operation at extremely low loads are typically of the order of a few dB. On the other hand, in the blade frequency range, these differences are typically much higher and on occasions amount to an order of magnitude.

The latter reveals that, in the harmonic frequency range, fluctuations that may be attributed to the influence of interference are characterized by standard deviation levels typically below  $0.2 \times E(S)$ , while in the blade frequency range, this factor is significantly higher, usually by two to four times.

It should be kept in mind that components from the harmonic frequency range are related to failures and malfunctions typical for all rotating machines, such as unbalance, misalignment, rotor bow or crack, problems with journal bearings etc. [1, 12, 17, 19]. These failures often develop comparatively fast and observable symptoms are characterized by high sensitivity to their extent. This means that the influences of load and interference may often be neglected and, if an abrupt symptom change is observed, this can be unambiguously attributed to a failure, qualitatively identified from the machine diagnostic model.

On the other hand, evolution of the fluid-flow system condition is usually slow. In the linear range (linearity referring to  $V(\theta)$ , not  $S_i(\theta)$  -



cf. Equation (12)), and with the exponential symptom operator  $\Phi_i$ , which is justified for these vibration components [9], a function given by

$$S_i(\theta) = S_i / S_i(\theta = 0) = e^{a\theta} \quad (14)$$

is fitted to the measured symptom time history. It has been estimated [9, 10] that the exponential factor  $a$  is typically of the order of  $10^{-5} \div 10^{-4}$  / day and tends to increase only for  $D$  close to 1, i.e., with substantial lifetime consumption degree, where linearity can no longer be assumed. In a number of cases, the above condition has been fulfilled for turbines with almost 150,000 hours in operation, which in practice means about 25 years. Assuming  $a = 10^{-4}$  / day and  $\sigma_r = 0.4$ , a simple calculation shows that the influence of interference in the blade frequency range results in fluctuations characterized by standard deviation equal to symptom value increase attained in over nine years.

This said, it is necessary to point out that, an abrupt increase of the symptom value caused by a fluid-flow system random or 'hard' damage does happen. An example is provided, e.g., in [11]. In such cases, correlation analysis proves very useful.

Finally, it has to be stressed that sensitivity to the  $\mathbf{R}$  and  $\mathbf{Z}$  vectors components turns out to be dependent on condition parameters. This provides a basis for diagnostic reasoning employing so-called statistical symptoms (see, e.g., [14]). Their importance results from the fact that, in a real industrial plant conditions, influences of load and interference cannot be eliminated. This field of research seems to hold much promise.

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