# Short Time Vibration Analysis and Parameterisation as a Tool for Machine Prototypes Testing

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**Abstract**

The paper outlines the idea of using the short-time vibration signal processing for testing machines of a complex design, particularly the machines consisting of many subassemblies with a non-stationary (e.g. cyclic) operation modes. The method presented consists in graphic representation of vibrations of subassemblies in the form of a trace on a plane. It is created by associating instantaneous rms values of vibration acceleration and instantaneous Rice frequency values obtained as a result of short-time signal processing. On the basis of the shape, orientation of the created trajectory and / or dispersion / concentration of points on the $R_f - a_{\mathrm{rms}}$ plane, the type of non-stationarity of generated vibrations can be identified. The presented methodology can be used for testing machine prototypes. The results in the proposed form can be helpful to determine the type of vibration reduction systems for individual machine subassemblies. It is also possible to detect subassemblies for which an increase in machine capacity results in an increase in the level of generated vibrations. The location of the averaged values of measures obtained for a new machine on the $R_f - a_{\mathrm{rms}}$ plane can be a reference point for further monitoring of machine vibration and for detection of damage or malfunction of its subassemblies.

***Keywords:*** machine vibration, non-stationarity identification, short time signal analysis

## 1. Introduction

A group of machines can be distinguished for which the methodology of the assessment of their technical condition on the basis of vibration measurements, specified in standards (e.g. [1, 2]) is not adequate. This may result from the range of rotation speeds or from the variety and specifics of working movements (not necessarily rotational) performed by machine subassemblies. Machines with a complex design and additionally with many drive units (e.g. electric motors, pneumatic cylinders) are used, among others in the food, textile and tobacco industry. In this case, the evaluation of machine vibrations based on the aforementioned standards is not justified. Therefore, the problem arises how to run prototype tests and how to assess the technical condition of this class of machines during operation. Methods are sought and developed that allow detection of machine malfunctions. One of possible solutions is to use the change of poles location – natural frequency and damping ratio in order to detect and localize the machine malfunctions [3].

In this paper the author proposes a slightly different approach to running vibration tests of machines of a complex design. It uses a graphic representation of results of short-time processing of vibration signals recorded at representative measuring points located on individual subassemblies of the machine. This method involves synchronous determination and visualization of the location of instantaneous rms values of vibration accelerations $a_{\mathrm{rms}}(\tau)$ and Rice frequencies $R_f(\tau)$ on the $R_f - a_{\mathrm{rms}}$ plane. This approach allows one to determine the specifics of the operation of subassemblies e.g. during prototype testing. On the other hand, the observation of the change of the position on the $R_f - a_{\mathrm{rms}}$ plane of the averaged $R_f(\Theta)$ and $a_{\mathrm{rms}}(\Theta)$ values during operation time $\Theta$ can be the basis for detection of malfunctions of subassemblies.

The paper presents an outline of the proposed alternative methodology and sample results obtained during vibration tests of the prototype of a horizontal tray former. The purpose of this research (in the short term aspect) was to obtain data that can be helpful for optimizing the construction of the prototype of the former and for selection of vibration elimination systems. Test results (in the averaged values aspect) can be treated as reference values for further monitoring of the technical condition of the machine during operation.

## 2. The test object

The FTHT6 former (Fig. 1) is a modular machine used for the erection of cardboard trays. This machine has a cyclical operation mode. It consists of several cooperating subassemblies which make various working movements: rotations of variable speed as well as reciprocating and complex movements. Some phases of forming and bonding of a cardboard have an additional impact nature (e.g. bending, pressure during the gluing phase).

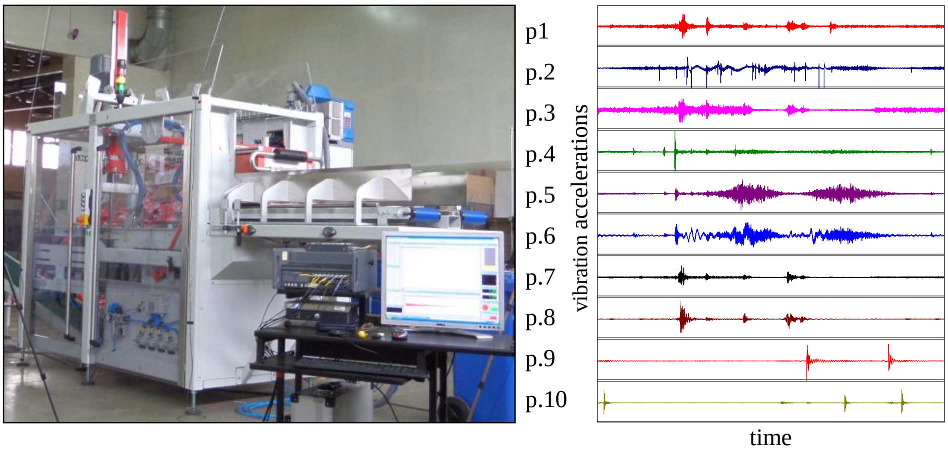



Figure 1. FTHT 6 former and vibration accelerations recorded at representative measuring points p.1 – p.10 (one machine operation cycle is presented) [4]

The research was carried out for the rated cardboard formation speed, i.e. 25 cycles per minute. Vibration accelerations were recorded at 10 representative points located on the former's prototype:

1 – cardboard manipulator beam (lower),
2 – cardboard manipulator rotary beam (upper),
3 – servo drive beam,
4 – punch gearmotor base,
5 – punch guide,
6 – punch,

7 – transverse beam for cardboard slide (at the end),
8 – the end of the guide (channel bar),
9 – cardboard bender,
10 – pneumatic clamp (pressure).

Their locations were determined on the basis of kinematic analysis of the machine and suggestions from the construction team. Figure 1 also shows a fragment of recorded vibration accelerations covering one cycle of operation. The variety of dynamic interactions of individual subassemblies is visible.

## 3. Methodology

Vibration accelerations at representative points were recorded synchronously due to the specificity of dynamic interactions, the complexity of the FTHT6 former and its high capacity. For the vibration tests of the prototype of the former eight IMI® (type 627A01, 603C01) and two MMF (type KD10) accelerometers were used. The signals from the ICP (IMI) standard transducers were recorded directly by an eight-channel recording unit TEAC LX-10. The signals from the charge transducers (KD10) were pre-conditioned in a preamplifier (Nexus B&K) and recorded by a two-channel data acquisition module VibDAQ 2+. The way the transducers were mounted and the analog-to-digital conversion parameters (sampling frequency 6 kHz, 24 bit quantization) ensured a linear vibration measurement with an appropriate dynamics in the frequency band to a minimum of 3 kHz.

The signals were analysed and parameterised in an application developed in the DASYLab (Data Acquisition System Laboratory) environment. The digital signal processing included: signal windowing, STFT (*Short Time Fourier Transform*) analysis [5, pp 46-52] as well as, basing on time-frequency (TF) maps, the determination of instantaneous rms values of vibration accelerations $a_{\mathrm{rms}}(\tau)$, instantaneous Rice frequencies $R_f(\tau)$ and creating a display of results on the $R_f - a_{\mathrm{rms}}$ plane. Further data synthesis and parameterization were enabled.

A rectangular time window with a size of 512 samples was used to segment the data in the process of signal processing. As a result, short-time, 0.085-second-long signal sequences were obtained. The time window shift $\Delta\tau$ in the STFT analysis was equal to the window size (overlapping = 0). TF maps were the basis for further parameterisation of vibration accelerations. An example of such a map (2D and 3D) is shown in Figure 2. The sonogram shows a broadband impulse and subsequent monoharmonic damped vibrations of the bender (measurement point no. 9) with a frequency of approx. 75 Hz.

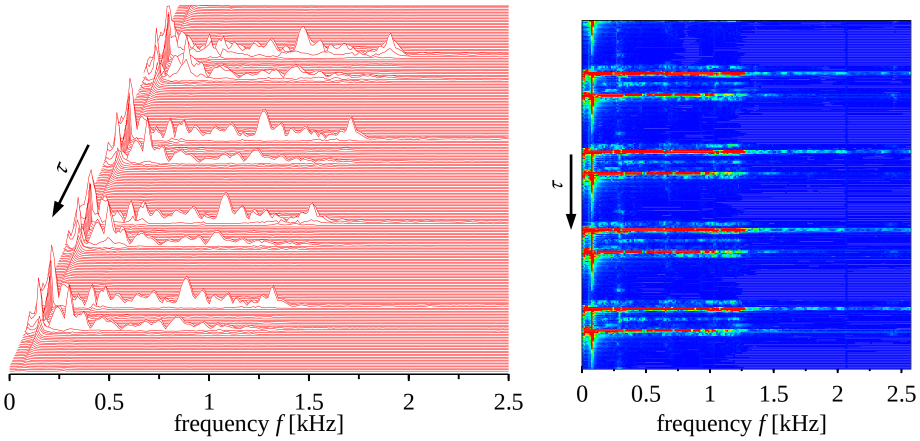



Figure 2. An example of STFT analysis covering four operation cycles of the former (accelerations recorded at the measurement point no. 9 – cardboard bender) [4]

It should be mentioned that Rice frequency $R_f$ describes the average frequency of the analysed process [6]. It is used, among others, in the diagnostics of rolling bearings, for the detection of damage of valves in internal combustion engines [7] and cavitation [8]. The Rice frequency can be determined based on power density spectra [9 pp 84-85]. Bearing this in mind, the instantaneous Rice frequency values $R_f(\tau)$ can be obtained as a result of post-processing of subsequent short-time spectra with the following formula [10]:

$$R_f(\tau)=\left[\frac{\int_0^\infty f^2 X_a(f,\tau)df}{\int_0^\infty X_a(f,\tau)df}\right]^{\frac{1}{2}} \tag{1}$$

where:

$X_w(f,\tau)$ – sonogram obtained as a result of STFT of $a(t)$ signal,
$f$ – frequency,
$\tau$ – shift of the analysis time window.

The properties of the obtained $R_f(\tau)$ characteristics and methods of its further parameterisation are shown in a monograph by the author [11]. Bearing in mind the Parseval's theorem [12, p. 134] instantaneous $a_{rms}(\tau)$ values can be easily determined by synthesizing short-time spectra. A similar method of determination (calculation) of the rms value of the signal from the spectrum was used in another work [13]. As part of the research, averaged values of the measures $\check{R}_f$, and $\check{a}_{rms}$, for 20 s of signal covering 10 complete machine cycles were also determined.

It should be added that the determination of the above measures based on post-processing of TF maps gives a possibility of conducting a selective analysis in any frequency range. This is done by taking into account in the processing of components of a TF map located between arbitrarily set cutoff frequencies.

## 4. Results

An idea of interpreting the results of short-time parameterisation of vibrations is shown in Figure 3. It can be intuitively determined whether the signal is stationary or non-stationary in the amplitude and / or frequency terms. The proposed method can be an alternative to the methods of detecting signal instability described by Plazenet et. al [14].

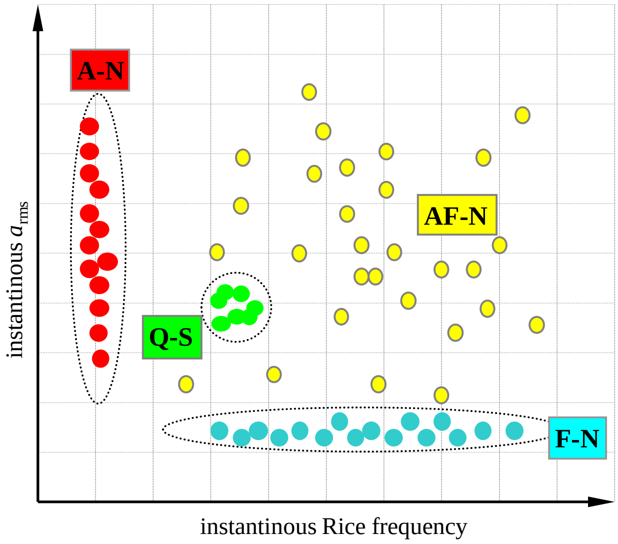



Figure 3. The idea of interpreting the results of short-time parameterisation of vibrations; a representation of various vibration signal types on the $R_f - a_{\mathrm{rms}}$ plane: A-N non-stationary amplitude; F-N non-stationary frequency, Q-S quasi-stationary signal, AF-F non-stationary signal in amplitude and frequency terms

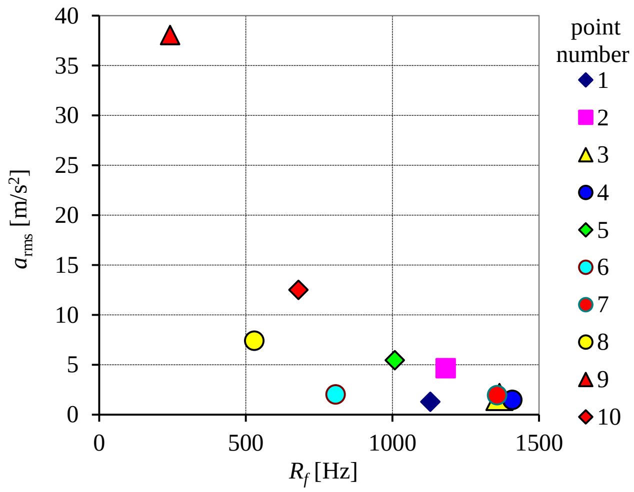



Figure 4. Results of acceleration parameterisation on the $R_f - a_{rms}$ plane, averaged values for 20 cycles of machine operation [4]

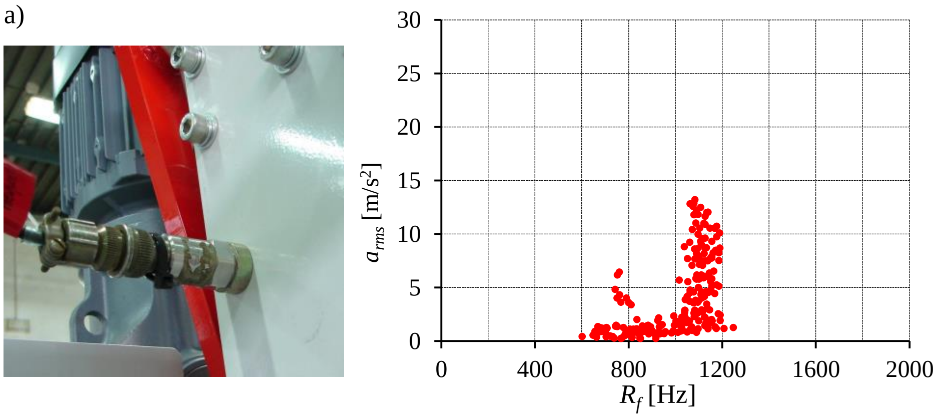
The results shown in Figure 4 allow one to estimate which subassemblies are characterized by the highest vibroactivity. The results also allow one to determine the average vibration frequency. Three subassemblies can be pointed out: cardboard bender (no. 9) pneumatic clamp (no. 10) and guide (no. 8) for which reduction of vibration levels may be considered. Signal parameterisation results (Table 1) are helpful in this aspect. The results are supplemented with acceleration peak values $a_{peak}$ and crest factors $k$.

Table 1. The results of parameterization of vibration accelerations (frequency band up to 2.5 kHz) for 10 operation cycles of the FTHT6 former (approx. 20 s)

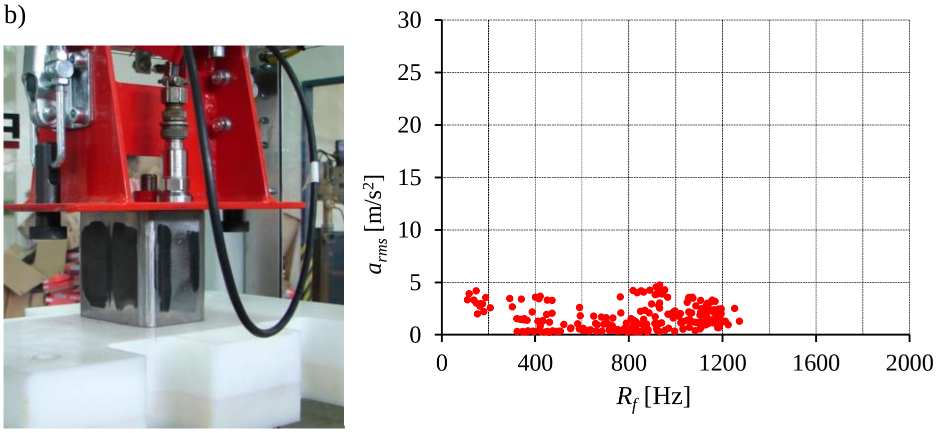
| Measure [unit] | measurement point number | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
| $\check{a}_{rms}$ [m/s$^2$] | 1.3 | 4.6 | 1.7 | 1.5 | 5.4 | 2.0 | 1.9 | 7.4 | 38.1 | 12.5 |
| $a_{peak}$ [m/s$^2$] | 15.8 | 59.1 | 20.9 | 53.2 | 45.5 | 16.1 | 46.2 | 168.6 | 810.0 | 451.9 |
| $k$ [–] | 12.5 | 12.7 | 12.0 | 36.3 | 8.4 | 8.0 | 24.0 | 22.8 | 21.3 | 36.2 |
| $\check{R}_f$ [Hz] | 1130 | 1182 | 1365 | 1409 | 1008 | 806 | 1358 | 529 | 241 | 680 |

An example of a graphic representation of the short-time parameterisation of vibration accelerations on the $R_f - a_{rms}$ plane is shown in Figure 5.

a)




b)




c)

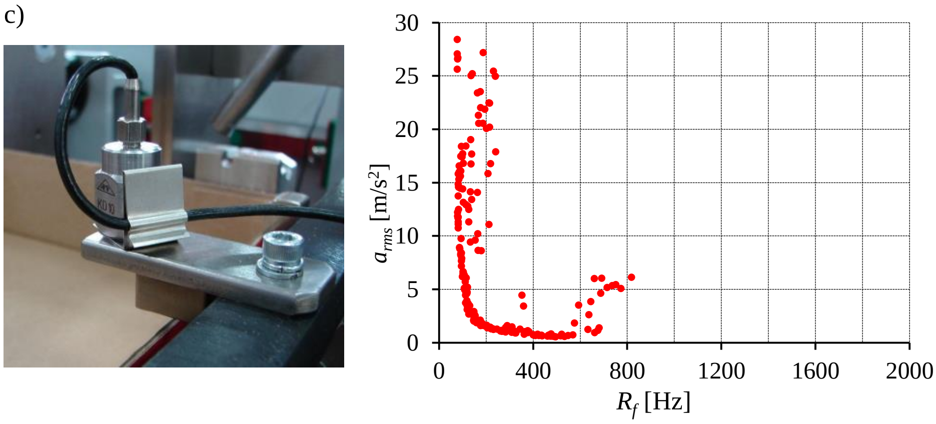



Figure 5. Example results of parametrisation of accelerations on the $R_f - a_{\mathrm{rms}}$ plane; a) punch gearmotor base b) punch c) cardboard bender [4]

A gearmotor is a device consisting of an electric motor and a gearing. The specificity of vibrations generated by the gearmotor is shown in Figure 5 a. Non-stationarity in both the amplitude and frequency terms is visible there. The cyclical start of this subassembly causes an increase in instantaneous $a_{\mathrm{rms}}(\tau)$ values as well as instantaneous $R_f(\tau)$ values. It should be noted that $R_f(\tau)$ takes into account all signal components in the considered frequency band. They are related to the rotation frequency, the gear meshing frequency and its superharmonics as well as components from magneto-electric phenomena (magnetostriction). In the case of the cardboard bender (Fig. 5 c), a non-stationarity in the amplitude terms is mainly visible. This is due to the cyclical activation of this subassembly (natural frequency around 75 Hz, see Fig. 2). The vibration acceleration signal recorded on the punch (Fig. 5 b) is non-stationary in the frequency terms, which is indicated by the large variability of $R_f(\tau)$ with only slight changes in $a_{\mathrm{rms}}(\tau)$.

The analysis of changes in the averaged values of $\check{R}_f$, i $\check{a}_{\mathrm{rms}}$ as a function of machine capacity is important at the stage of testing the prototype of the former. One can identify subassemblies for which increased capacity changes the nature and intensity of vibrations. This type of information can be premises for optimizing the machine design.

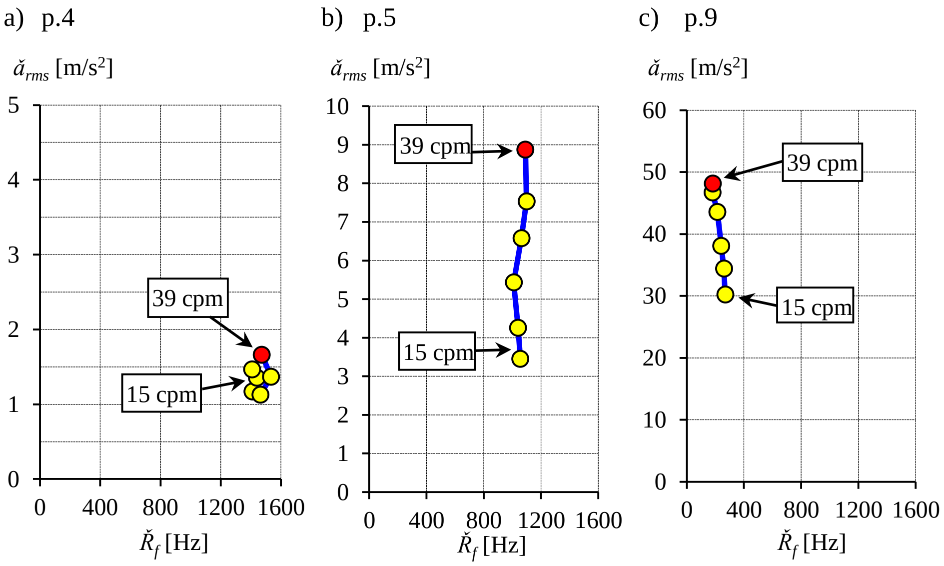



Figure 6. The change in the position of the result of the parameterisation of vibration accelerations on the $R_f - a_{\mathrm{rms}}$ plane resulting from the increase in machine capacity from 15 to 30 cycles per minute (cpm) a) gearmotor base b) punch guide c) cardboard bender [4]

Based on the images in Figure 6, it can be stated that an increase of the capacity of the former:

- has no significant effect on the vibration generated by the punch gearmotor (Fig. 6a),
- causes an increase in the vibration intensity of the punch guide and slightly changes the spectral composition (shift of $\check{R}_f$ towards higher frequencies, Fig. 6b)
- causes an increase in the vibration intensity of the cardboard bender and a decrease in $\check{R}_f$ due to the greater energy share of the subassembly's own frequency. This is due to the shortening of the time interval between successive excitations of the subassembly to vibrate.

## 5. Conclusions

The graphic representation of the $R_f(\tau)$, $a_{\mathrm{rms}}(\tau)$ results on the $R_f$ – $a_{\mathrm{rms}}$ plane in the short-time terms gives information about the nature of signal instability in both amplitude and frequency terms as well as the intensity of vibrations of individual subassemblies.

The averaged values $\check{R}_{f,}$ and $\check{a}_{\mathrm{rms}}$ obtained for a new machine should be treated as reference values in the vibration monitoring process. Changing the location of the point specified by $\check{R}_{f,}$ and $\check{a}_{\mathrm{rms}}$ on the $R_f - a_{\mathrm{rms}}$ plane during operation will be a signal of a change in the technical condition of the monitored subassembly. The way the results are displayed enables simultaneous monitoring of many subassemblies of the machine. The proposed method can be used to monitor complex machines with many subassemblies and drive units with different kinds of working movement.

This method can be used at the stage of machine prototypes testing as an effective tool for detection of subassemblies that increase vibroactivity along with the increase of machine capacity.

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