

RESEARCH ON IMPROVING SEALS TO SUPPRESS VIBRATION OF ROTARY MACHINES

Li Tao¹, Lei Shao¹, Olha Pazdrii²,
Oleksandr Gerasin³, Tymur Parashchenko⁴,
Volodymyr Zaytsev⁵, Wang Li^{6✉}

Received on February 21, 2025

Presented by Ch. Roumenin, Member of BAS, on April 29, 2025

Abstract

Based on the analysis of vibration signals modelled on a complex rotor system's impeller, such as an aviation gas turbine engine operating under steady-state conditions, this study demonstrates the influence of methodological factors – such as discrepancies in the natural frequencies of blade models, the sample length of the time series for the vibration process, and the sampling frequency – on the Hurst exponent value. The R/S analysis, or rescaled range method, proposed by H. Hurst for studying random processes, has provided a new perspective on the theory of random processes. This method has also become a highly effective and widely applied technical approach for examining a wide range of processes, from telecommunications and bioelectric signal analysis to solving functional diagnostic tasks for complex spatial objects. The findings of this study may be implemented within functional diagnostic systems for complex rotor systems to detect and predict the behaviour of damage in their rotating components.

Key words: vibration signals, Hurst method, R/S analysis, antipersistence, noise, gas turbine engine

This study was supported by the “Tangshan Science and Technology Plan Project” of Tangshan City (Grant No. 21130216C) and the “Central Guidance for Local Scientific and Technological Development Funding Projects” (Grant No. 246Z1912G).

<https://doi.org/10.7546/CRABS.2025.05.10>

Introduction. This work focuses on diagnosing the technical condition of complex systems, such as aviation gas turbine engines (GTE), using functional diagnostics systems that analyze vibrational and vibroacoustic processes during operation to prevent catastrophic failures [1–4]. Vibrodiagnostics is one of the most effective methods for determining the technical condition of gas turbines, offering high informativeness through vibration and acoustic signals that can be easily converted into electrical signals for analysis in operational conditions [2, 3]. Processing of vibroacoustic signals is a critical stage in vibration diagnostics, requiring constant improvement in algorithmic and methodological support for signal processing [5–7].

Signal processing methods, such as fractal analysis [8, 9], could be used to isolate informative components from vibration noise, enabling a more accurate assessment of the technical condition of the GTE. In [6] the application of a multi-level signal processing system initially used time-frequency analysis, but the obtained results were difficult to interpret and automate the process of diagnosing the engine's technical condition. To address this, an additional level of processing based on fractal analysis was introduced, allowing for better distinction between the GTE's technical condition, both with and without element damage. The aim of this study is to explore the influence of methodical factors on the determination of fractal characteristics in simulated vibration signals (VS) of the GTE impeller during stationary operation.

Problem statement. The physical processes that reflect the technical state of complex rotor systems are characterized by high levels of noise and chaotic behaviour. Typically, the vibration of the GTE includes numerous components of various origins, with the time series of the process displaying fractal properties [1–3]. The application of classical methods for processing VS does not fully allow for the isolation of useful signal components or the accurate interpretation of the GTE's technical state. To address this, we propose utilizing fractal analysis of VS R/S analysis, or the rescaled range method proposed by the English researcher H. Hurst [10–14].

This study proposes to determine the influence of methodological factors on the Hurst exponent using R/S analysis of complex VS from a gas turbine engine's impeller under stationary operating conditions. The analysis focuses on factors such as discrepancies in the natural frequency values of the object's model components, the sample length of the vibration process time series, and the sampling frequency.

Modelling and analysis of vibration signals. *Vibration signal modelling.* The modelling focuses on the compressor wheel, which contains 21 blades, under stationary operating conditions of an aviation GTE as presented in [12, 15–17]. The generation of vibroacoustic signals was carried out according to the numerical modelling scheme shown in Fig. 1, where: 1 is the vibration disturbance generation block $P(t)$; 2 are the blade models, $n(t)$ is the additive white

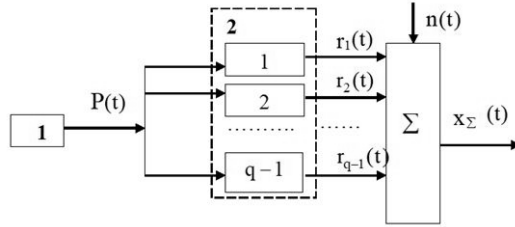


Fig. 1. Diagram of modelling vibroacoustic signals

noise [2, 12]. Blade models were represented by oscillatory systems with one degree of freedom, which in the modelling scheme were defined by their impulse responses [2]:

$$(1) \quad g_*(t) = \frac{1}{\omega_{1*}} e^{-ht} \sin \omega_{1*} t,$$

where $\omega_{1*}^2 = \omega_*^2 - h^2$, ω_* is the natural frequency of the blade model and h is the damping coefficient. The natural frequency was assumed to be the same for all blades, $\omega_* = 628 \text{ s}^{-1}$ ($f_* = 100 \text{ Hz}$).

Vibrational disturbances in steady-state operation occur at the rotor rotation frequency ω_p , which is referred to as the first (fundamental) rotor harmonic. There are also superharmonic components present in GTE with frequencies that are multiples of the fundamental rotor harmonic frequency $\omega_{pi} = i \cdot \omega_p$, ($i = 2, 3, 4, \dots$). Their level is significantly lower than that of the fundamental harmonic and decreases rapidly with increasing multiplicity. Additionally, subharmonic oscillations (where $i = 1/2; 1/3$), may be excited, typically occurring at the frequencies of resonant (critical) modes and usually falling outside the operating frequency range of the rotor [18, 19]. In general, the rotor vibration in steady-state conditions can be described by a model of a polyharmonic process [2]:

$$(2) \quad P(t) = \sum_{i=1/m_1}^{i=n_1} P_i \sin(i\omega_p t + \phi_{i0}),$$

where P_i and ϕ_{i0} are the amplitude and initial phase of the i -th harmonic, respectively; m_1 and n_1 are positive integers.

For the investigation, a steady-state vibrational disturbance regime with additional superharmonic components at frequencies that are multiples of the fundamental rotor harmonic frequency was considered. The vibroacoustic process was modelled for the following parameters: $\omega_p = 1570 \text{ s}^{-1}$ ($f_p = 250 \text{ Hz}$); $\phi_{i0} = 0$; $P_1 : P_2 : P_3 = 1 : 10^{-2} : 10^{-3}$.

The generation of the emitted vibroacoustic signals at the output of the model was conducted by convolving the impulse responses of the blades (1) with the input vibrational disturbance (2).

For further processing, the following vibrational signals were modelled: Signal 1 – a signal without the addition of additive noise; Signal 2 – with the addition of stationary Gaussian noise with zero mean (signal-to-noise ratio, SNR $\rho = 16 \cdot 10^2$); Signal 3 – with the addition of stationary Gaussian noise with zero mean (SNR $\rho = 16$).

Since discrepancies in the values of the natural frequencies of the blades may occur in real conditions due to the technological features of their construction [1, 2], it was proposed to consider each of the obtained signals both without and with the account of possible discrepancies in the values of the natural frequencies of the blade models (in the range of $\pm 3 \dots 10\%$ of f_*).

The modelling of vibrational signals without and with discrepancies in the values of the natural frequencies was conducted over a time interval of $t = 5$ s, with a sampling period of $\Delta T = 1 \cdot 10^{-4}$ s, and the time realizations contained $N = 5 \cdot 10^4$ samples.

As an example, Fig. 2 shows a portion ($N = 10\,000$ samples) of the modelled vibrational signal with the addition of stationary Gaussian noise with zero mean SNR $\rho = 16 \cdot 10^2$ (Signal 2) and a portion ($N = 10\,000$ samples) of the modelled vibrational signal with the addition of stationary Gaussian noise with zero mean SNR (Signal 3). Both signals are presented without considering the frequency discrepancies in the natural frequencies of the blades.

On the graph, the amplitude values are plotted on the ordinate axis in arbitrary units, while the abscissa axis represents time in increments. As can be seen, the presented portion of the time realization of the vibration signal from the working wheel exhibits a noise-like structure.

Figure 3 shows the time realizations of the complete vibration signal ($N = 50\,000$ samples) with the addition of stationary Gaussian noise with a zero mean value (signal-to-noise ratio $\rho = 16$) (signal 3), considering the deviation of the natural frequencies at levels of $\pm 3\%$ and $\pm 5\%$ from f_* .

Analysis of vibration signals. To investigate the impact of various factors on the Hurst exponent, R/S analysis was conducted in several stages. Initially, for the complete realizations of the VS obtained from the output of the working wheel model – signals 1–3 – both with and without considering the deviations of the natural frequencies of the blades, R/S analysis was performed to obtain the Hurst exponent value H . The obtained values of the Hurst exponent are presented in Table 1 (values of the Hurst exponent for complete realizations of VS). From the obtained results it is evident that the Hurst exponent lies within the presented range, indicating the antipersistent nature of the analyzed processes. The value of the Hurst exponent, determined for the VS, increases with a decrease in the signal-to-noise ratio, indicating the emergence of a trend.

As shown, with varying levels of deviation in the natural frequencies of the blades, the Hurst exponent values for the signals maintain their trend; however, with a significant difference in blade frequencies (10%), the Hurst exponent de-

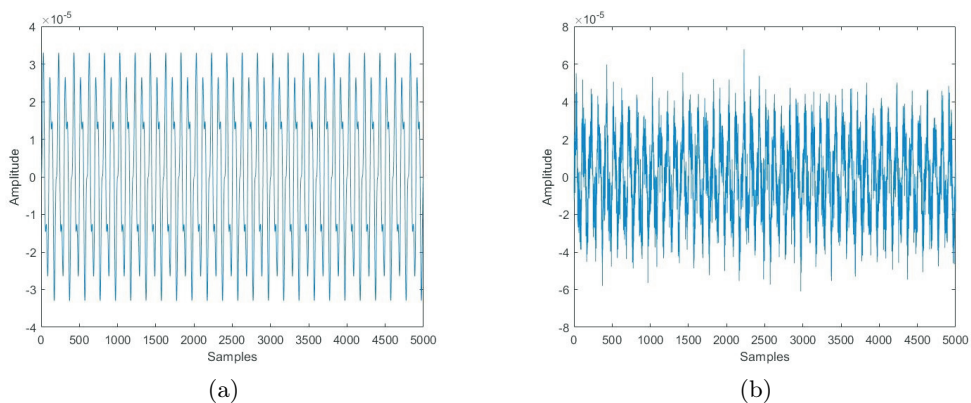


Fig. 2. Time realization of a portion of the vibrational signals: (a) Signal 2; (b) Signal 3

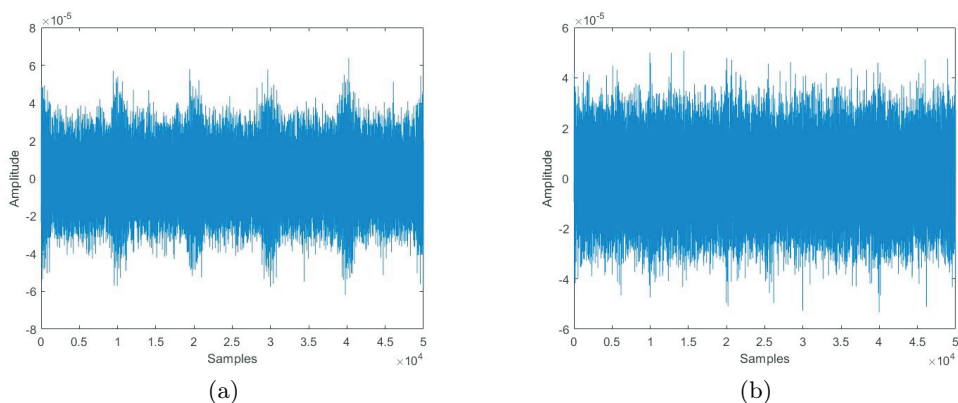


Fig. 3. Time realization of vibration signals considering the deviations of the natural frequencies of the blades for signal 3: (a) 3%; (b) 10%

creases, indicating the chaotic nature of the process. As the level of deviation in the natural frequencies of the blades increases from 0% to 6%, the Hurst exponent also increases; however, at higher levels of deviation in the natural frequencies of the blades, the Hurst exponent changes ambiguously, yet still remains within the range of values that indicate an antipersistent process. This allows us to conclude about the nature of the process regardless of the level of deviation in the natural frequencies of the blades.

In the next stage, the obtained time realizations – signals 1–3, without considering deviations in the natural frequencies of the blades – were divided into samples of varying lengths, and the Hurst exponent was calculated for each sample separately. The obtained values of the Hurst exponent are presented in Table 1 (values of the Hurst exponent for samples of varying lengths of vibration signals without considering deviations in the natural frequencies of the blades).

T a b l e 1

Values of the Hurst exponent

Values of the Hurst exponent for complete realizations of vibration signals						
Signals	Deviation level of the natural frequencies of the blades					
	0%	3%	5%	6%	10%	
Signal 1	0.1416	0.18302	0.19719	0.21857	0.18951	
Signal 2	0.25265	0.29719	0.30384	0.32172	0.299	
Signal 3	0.67114	0.6771	0.67759	0.6952	0.67744	
Values of the Hurst exponent for samples of varying lengths of vibration signals without considering deviations in the natural frequencies of the blades						
Signals	Sample length N (samples)					
	50000	30000	10000	5000	2000	1000
Signal 1	0.1416	0.14056	0.14048	0.14131	0.15479	0.1508
Signal 2	0.25339	0.25283	0.25425	0.25821	0.27834	0.28174
Signal 3	0.67059	0.67134	0.67182	0.67521	0.69439	0.70168
Values of the Hurst exponent for time realizations at different sampling frequencies						
Signals	Sampling frequency f_{∂} (kHz)					
	5	10	15	20		
Signal 1	0.14056	0.1416	0.14367	0.14367		
Signal 2	0.25496	0.25365	0.25243	0.25311		
Signal 3	0.67522	0.67134	0.67307	0.67296		

The results presented in Table 1 illustrate that for each of the signals the Hurst exponent remains nearly unchanged and retains its trend for samples ranging from 50 000 to 5000 points. From the provided data, it can be seen that with a reduction in sample volume, the value of the Hurst exponent increases, while remaining within the corresponding interval, which allows us to draw conclusions about the nature of the investigated process. According to expression (4), for sample volume $N \sim 100 N$, the time series considered (signal 1 and signal 2) are antipersistent with a reliability of 97.5% when $H \leq 0.29$. Signal 3 is persistent with a reliability of 97% when $H \geq 0.7$. Thus, it has been experimentally determined that in this case, for assessing the state of the system, it is necessary to consider a sample of length $N \geq 2000$.

The next stage of the research involved R/S analysis for signals 1–3 at different sampling frequencies. The obtained results are shown in Table 1 (values of the Hurst exponent for time realizations at different sampling frequencies).

From the obtained results, it is evident that with changes in the sampling frequency, the values of the Hurst exponent maintain their trend.

Conclusions. The analysis of simulated vibration signals from the working wheel of a gas turbine engine in a steady-state operating mode under various signal-to-noise ratios demonstrated the effectiveness of R/S analysis in determining the characteristics of the processes occurring in the studied object. The Hurst exponent retains its trend and reliably indicates the nature of the process under

any discretization parameters. It has a certain confidence interval for determining the sample size of the time series, even with a relatively small (6%) deviation of the natural frequencies of the blades.

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¹*Department of Robot Engineering, School of Mechanical Engineering, Jiangsu Ocean University, 222005 Lianyungang, China*
e-mails: taoli0401@163.com, 2424671415@qq.com

²*Department of Computer-integrated Optical and Navigation Systems, National Technical University of Ukraine “Igor Sikorsky Kyiv Polytechnic Institute”, 03056 Kyiv, Ukraine*
e-mail: olgapazdri@gmail.com

³*Department of Computerized Control Systems, Admiral Makarov National University of Shipbuilding, 54007 Mykolaiv, Ukraine*
e-mail: oleksandr.gerasin@nuos.edu.ua

⁴*State Scientific Research Institute of Armament and Military Equipment Testing and Certification, 18003 Cherkasy, Ukraine*
e-mail: Liberian2009@gmail.com

⁵*Department of Marine Technology and Ocean Engineering, Admiral Makarov National University of Shipbuilding, 54007 Mykolaiv, Ukraine*
e-mail: zvv1949@gmail.com

⁶*School of Mechanical Engineering, North China University of Science and Technology, 063210 Tangshan, China*
e-mail: 18031320645@163.com