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# Analiza vibracij gonilnika turbopuhala

## An Analysis of Turbocharger Impeller Vibrations

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*V prispevku so predstavljeni rezultati eksperimentalnih raziskav lastnih nihanj lopatic turbopuhala. Za vzbujanje strukture gonilnika turbopuhala (sestavljeni sistem lopatice-disk) sta bili uporabljeni impulzna motnja in akustično harmonsko vzbujanje. Odziv sistema je bil merjen z merilniki pospeškov, prilepljenimi na lopaticah gonilnika. Za določitev razmernika dušenja je bila uporabljena metoda logaritemskega zmanjševanja. Meritve so pokazale, da se lopatice na motnje odzivajo najbolj intenzivno (največje amplitude) z upogibnimi lastnimi nihanji, ki pripadajo prvi lastni obliki. V izmerjenih odzivih so bili ocenjeni razmerniki dušenja. Izkazalo se je, da je razmernik dušenja za prve lastne oblike pri obeh načinih vzbujanja enak in konstanten $\delta$ = 5E -05. Razmerniki dušenja za preostale lastne oblike nihanj so za velikostni razred večje. Analiza dinamičnih karakteristik lastnih nihanj lopatic s prvo lastno obliko nakaže možnosti numeričnega modeliranja mehanizmov dušenja, ki bi omogočilo simuliranje obratovanja v resonanci in njeni bližini.*

**(Ključne besede: turbopuhala, gonilniki, analize vibracij, dušenje vibracij)**

*The paper presents the results of experimental studies of natural vibrations of turbocharger blades. In order to excite the structure of a turbocharger impeller (composite blade-disc system), impulse disturbance and acoustic harmonic excitation were used. The response of the system was measured using accelerometers attached to the impeller blades. The logarithmic decrement method was used to determine the damping ratio. Measurements showed that the blades are most responsive (the largest amplitudes) to disturbances with bending natural vibrations, classified as the first mode. The damping ratia were estimated from the measured responses. It turned out that the damping ratio for the first free form is equal and constant for both excitation methods: $\delta$ = 5E -05. The damping ratia for other free forms of the oscillations were greater by one order of magnitude. The analysis of the dynamic characteristics of the blades' natural vibrations with the first free form indicated possibilities for numerical modeling of damping mechanisms that would enable the simulation of operation at, or near, resonance frequency.*

**(Keywords: turbocharger, impellers, vibration analysis, vibrations damping)**

### 0 UVOD

Lopatice gonilnikov modernih turbinskih strojev postajajo zaradi aerodinamičnega optimiranja čedalje tanjše, zaradi česar so izpostavljene vse večjim statičnim in še zlasti dinamičnim, mehanskim obremenitvam. Mehanske poškodbe zaradi utrujanja gradiva ali celo trenutni lom konstrukcije so najpogosteje posledica dinamičnih obremenitev, povzročenih zaradi vibracij lopatic. Te so lahko vzbujane z različnimi mehanizmi, to so: izsrednost rotorja, neustaljene aerodinamične sile, turbulenca in akustična resonanca [1]. Zaradi velikega vpliva značilnosti geometrijskih oblik turbinskega gonilnika na njegovo dinamično obnašanje so bili v dosedanjih raziskavah razviti in uporabljeni različni modeli in

### 0 INTRODUCTION

The impeller blades used in modern turbomachinery are being produced in increasingly thinner form due to the demands of aerodynamic optimisation. A consequence of this is that blades are being subjected to increasing static, and especially dynamic, mechanical loads. Mechanical damage resulting from material fatigue or even the instantaneous fracture of the structure are most frequently the result of dynamic loads caused by the blade vibrations. These can be created by various mechanisms such as rotor eccentricity, nonstationary aerodynamic forces, turbulence and acoustic resonance [1]. Due to the large influence of the geometric characteristics of turbine impellers on their dynamic behavior, various numerične metode za popis dinamičnih lastnosti konstrukcije ([2] do [4]).

V nasprotju z vitkimi, elastičnimi lopaticami v aksialnih turbinskih strojih (npr. aksialni kompresorji ali nizkotlačne stopnje parnih in plinskih turbin), so lopatice v turbopuhalih izpostavljene vibracijam z visokimi frekvencami in majhnimi amplitudami, zaradi česar je vpliv samovzbujevalnih učinkov zanemarljivo majhen [2]. Omeniti velja, da kljub majhnim amplitudam pomikov lopatic, lahko napetosti dosežejo precejšnje vrednosti.

Pri obravnavi vibracij so poleg vzbujevalnih sil pomembni tudi mehanizmi dušenja, zlasti v področju resonance in njeni okolici. Pri turbinskih strojih razlikujemo tri vrste dušenja:

- strukturno dušenje,
- sistemsko dušenje in
- aerodinamično dušenje.

Strukturno dušenje predstavlja različne disipacijske procese v materialu. Pri tem se napetostna in kinetična energija vibrirajočega sistema spreminja v toploto. Za modeliranje sestavljenih sistemov lopatice – disk je ta vrsta dušenja najpomembnejša. Analizirano in predlagano je bilo nekaj primernih metod za identifikacijo parametrov dušenja in modeliranja, ki temeljijo na poznavanju lastnih nihanj konstrukcije ([3] in [4]).

Sistemsko dušenje se pojavlja na spojih posameznih elementov konstrukcije. Določa ga disipacija energije zaradi trenja in strižnih učinkov na površinah med posameznimi sestavnimi deli konstrukcije (npr. v korenskem spoju lopate in gredi). V primerjavi z vzdolžnimi in torzijskimi vibracijami gredi turbine, je tovrstno dušenje pri vibracijah lopatic nepomembno.

Aerodinamično dušenje se pojavlja kot posledica interakcije konstrukcije (lopatice) in obtekajočega toka tekočine. Ta je tem večja, čim večje so amplitude deformacij vibrirajočih lopatic. Pri aksialnih kompresorjih in nizkotlačnih stopnjah plinskih in parnih turbin so deformacije lopatic zaradi njihove vitkosti razmeroma velike, zato je treba aerodinamično dušenje upoštevati. Nasprotno pa so lopatice na gonilniku turbopuhala kratke in zato izpostavljene vibracijam z zelo majhnimi amplitudami. V tem primeru je vpliv aerodinamičnega dušenja prav tako zanemarljivo majhen.

Raziskava je osredotočena na eksperimentalno določevanje odzivov sestavljenega sistema lopatice - disk na motnje in na oceno pripadajočih razmernikov dušenja.

Gonilnik, ki je predmet raziskave, ima premer 130 mm in ima 11 lopatic (sl. 1). Izdelan je iz temperaturno obstojne nikel-kromove zlitine INCONEL 713 LC, ki ima naslednjo sestavo: Ni (76%), Cr (17%) in Fe (7%). Vroči izpušni plini vstopajo v gonilnik skozi spiralni okrov brez lopatic

models and numerical methods for the description of the dynamic properties of the structure have been developed and used in research ([2] to [4]).

As distinct from the thin, elastic blades in axial turbomachinery (e.g. axial compressors or low-pressure stages of steam and gas turbines), the blades in turbochargers are exposed to high-frequency small-amplitude vibrations, making the influence of self-excitation effects negligible [2]. It should be mentioned that in spite of the small blade-displacement amplitudes, the stresses can be very high.

In addition to the excitation forces, damping mechanisms are also important for the study of vibrations, especially at, or near, resonance frequencies. Three types of damping can be distinguished in turbomachinery:

- structural damping,
- system damping,
- aerodynamic damping.

Structural damping includes various dissipation processes within the material. The strain and kinetic energy of the vibrating system is converted to heat. This type of damping is the most important for the modeling of composite blade-disc systems. A few suitable methods based on the knowledge of free-structure vibrations were analysed and proposed for the identification of damping and modeling parameters ([3] and [4]).

System damping occurs at the joints between individual elements of a structure. It is determined by the energy dissipation due to friction and shear forces on the surfaces between individual components of the structure (e.g. at the base of the joint between the blade and the shaft). In contrast to longitudinal and torsional vibrations in turbine shafts, such damping is insignificant for blade vibrations.

Aerodynamic damping occurs as a result of the interaction between the structure (blade) and the fluid flow surrounding it. It increases in proportion to the displacement amplitudes of the vibrating blades. In axial compressors and low-pressure stages of gas and steam turbines, blade deformations are relatively large due to their thinness, therefore aerodynamic damping has to be taken into account. On the other hand, the blades of turbocharger impellers are short and are thus subjected to vibrations with very small amplitudes. In this case, the influence of aerodynamic damping is also negligible.

Our research focused on the experimental determination of the responses of a composite blade-disc system to disturbances and the assessment of the corresponding damping ratios.

The impeller which is the subject of this research has a diameter of 130 mm and 11 blades (Figure 1). It was produced from a heat-resistant nickel-chromium alloy, INCONEL 713 LC, which has the following composition: Ni (76%), Cr (17%), Fe (7%). Hot exhaust gases enter the impeller through a spiral cass krožnim prečnim prerezom. Glavni obratovalni podatki so:

- največja vstopna temperatura 1020 K,
- največji masni tok 1 kg/s,
- tlačno razmerje 3,5,
- največja krožna frekvenca 60 000 min$^{-1}$.

ing without blades and with a circular cross-section. The basic operating data are as follows:

- max. inlet temperature: 1020 K,
- max. mass flow: 1 kg/s,
- pressure ratio: 3.5,
- max. rotating speed: 60,000 rpm.

Sl. 1. *Gonilnik turbopuhala z 11 lopaticami*
Fig. 1. *Turbocharger impeller with 11 blades*

V prejšnji raziskavi [5] je bila z metodo končnih elementov izvedena numerična analiza dinamičnega obnašanja obravnavanega gonilnika. S pomočjo modalne analize so bile izračunane lastne frekvence rotorja. Poudarek je bil na določevanju tistih lastnih oblik, pri katerih so lopatice upogibno nihale. Številu lopatic ustrezno je bilo izračunanih 11 različnih lastnih frekvenc, pri katerih so lopatice nihale upogibno z vozlom v korenu lopatic. Ta nihanja so v nadaljevanju imenovana s skupnim pojmom 1. lastna oblika. Lastna nihanja s 1. lastno obliko se med seboj razlikujejo s številom t.i. vozelnih črt po premeru gonilnika (imenovanih tudi vozelni premeri), ki razmejujejo lopatice, nihajoče v protifazi.

V zgoraj omenjenem modelu dušenje ni bilo upoštevano. Pri numeričnem simuliranju resonance tak model odpove, saj se amplitude s časom zvečujejo v neskončnost. Znano pa je, da se v obratovalnem območju krožnih frekvenc radialnih turbopuhal pojavlja več resonanc.

V literaturi [6] so predstavljeni zanimivi podatki o meritvah vibracij lopatic gonilnika turbopuhala v celotnem obratovalnem območju v dejanskih obratovalnih razmerah. Pri teh raziskavah je bila uporabljena tehnika merjenja dinamičnih sil z visokotemperaturnimi uporovnimi lističi, prilepljenimi na lopatico gonilnika. Pri tem je bil signal prenesen z rotirajočega se rotorja z uporabo enokanalnega telemetrijskega sistema. Pokazano je bilo, da se v

In a previous study [5], the finite-element method was used to perform the numerical analysis of the impeller's dynamic behaviour. The rotor's free frequencies were calculated using modal analysis. Emphasis was on determining those free forms in which the blades oscillated by bending. In accordance with the number of blades, 11 different free frequencies were calculated in which the blades experienced bending vibrations with the node at the blade root. Later on, these oscillations were jointly named the "first free form". Natural vibrations with the first free form differ in the number of node lines along the impeller diameter (also called nodal diameters), which delimit the blades oscillating in counterphases.

Damping was not taken into account in the above-mentioned model. Such a model fails in the numerical simulation of resonance, since amplitudes increase to infinity over time. It is known, however, that several resonant frequencies occur within the operating range (circular frequencies) of radial turbochargers.

The literature [6] presents interesting data on measurements of turbocharger impeller-blade vibrations over the entire operating range under actual operating conditions. In this research, the technique of measuring dynamic forces using highly temperature-resistant rods attached to the impeller blade was used. The signal was transferred from a rotating rotor using a single-channel telemetry system. It was shown that several resonances occur in the turboobratovalnem območju turbopuhala pojavlja več resonanc, ki so bile vrisane v Campbellov diagram.

Dušenje turbinskih lopatic je bilo predmet eksperimentalnih raziskav različnih avtorjev. Velika večina prispevkov se zaradi njihove razširjenosti nanaša na lopatice aksialnih turbinskih strojev. Te so izdelane posamično in ločeno vpete v turbinsko kolo, zato eksperimentalni podatki vsebujejo informacijo tako o strukturnem kakor tudi o sistemskem dušenju. V literaturi [2] so zbrani rezultati podrobnih raziskav dušenja lopatic nizkotlačnega gonilnika aksialne, parne turbine. Predmet raziskav je bila določitev soodvisnosti dušenja, amplitude vibracij, geometrijske oblike lopatic, vzdolžne obremenitve lopatic, oblike nihanja, gradiva in tokovnega polja. Rezultati kažejo, da imajo na dušenje pretežni vpliv histerezni pojavi (strukturno dušenje), medtem ko trenje na korenskih spojih lopatic, kakor tudi tokovno polje zanemarljivo malo prispevata k celotnemu dušenju.

Glavni nameni teh raziskav so bili:

- s preskusi določiti lastnosti strukturnega dušenja nihanja lopatic pri lastnih nihanjih lopatic gonilnika (upogibna nihanja lopatic sistema lopatice - disk);
- določiti obliko lastnega nihanja, ki pripada najmanjšemu dušenju;
- z metodo logaritemskega dekrementa oceniti velikost razmernika dušenja, in sicer z različnimi načini vzbujanja lastnih nihanj.

## 1 DOLOČANJE DISIPACIJE ENERGIJE ZARADI DUŠENJA

Za določitev velikosti strukturnega dušenja smo opravili meritve odziva turbinskih lopatic na začetno motnjo. Meritve so bile opravljene na gonilniku brez okrova v atmosferskih razmerah v laboratoriju.

Merilni sistem je bil sestavljen iz štirih merilnikov pospeškov (Brüel & Kjær, frekvenčno območje do 80 kHz) in pripadajočih nabojnih predojačevalnikov. Izmerjeni signali so bili posneti na računalnik s pomočjo 4-kanalne 16-bitne merilne kartice. Ločljivost zbiranja rezultatov je bila nastavljena na 80 kHz. Merilni sistem in sistem za zbiranje podatkov sta prikazana na sliki 2. Časovni signali odzivov so bili sočasno merjeni na 4 od 11 lopatic. Slika 3 prikazuje namestitev merilnikov pospeškov. Razvidno je, da sta bila uporabljena dva para merilnikov pospeškov – prvi z maso 0,2 g in drugi z maso 2 g. Merilno območje vseh merilnikov pospeškov je za velikostni razred večje od izračunanih lastnih frekvenc sestavljenega sistema lopatice - disk [5]. Mesto namestitve merilnika pospeškov na lopatici je bilo izbrano s poprejšnjo numerično analizo lastnih nihanj, in sicer na mestu največjih deformacij (in pospeškov) pri lastnih nihanjih s 1. lastno obliko. S

charger operating range, and these were drawn on a Campbell diagram.

The damping of turbine blades has been the subject of experimental research by various authors. A large majority of the contributions refer to the blades of axial turbomachinery, because of their widespread use. These blades are produced individually and attached to the turbine wheel separately, so experimental data contains information on both structural and system damping. The literature [2] states the collected results of a detailed study of damping of the blades of low-pressure impellers in axial steam turbines. The subject of this research was the determination of the interdependence of damping, vibration amplitude, blade geometry, axial blade loads, type of oscillation, material and flow field. The results indicate that hysteresis phenomena (structural damping) have a predominant influence on damping, while friction at the blade-root joints and the flow field contribute negligibly to the total damping.

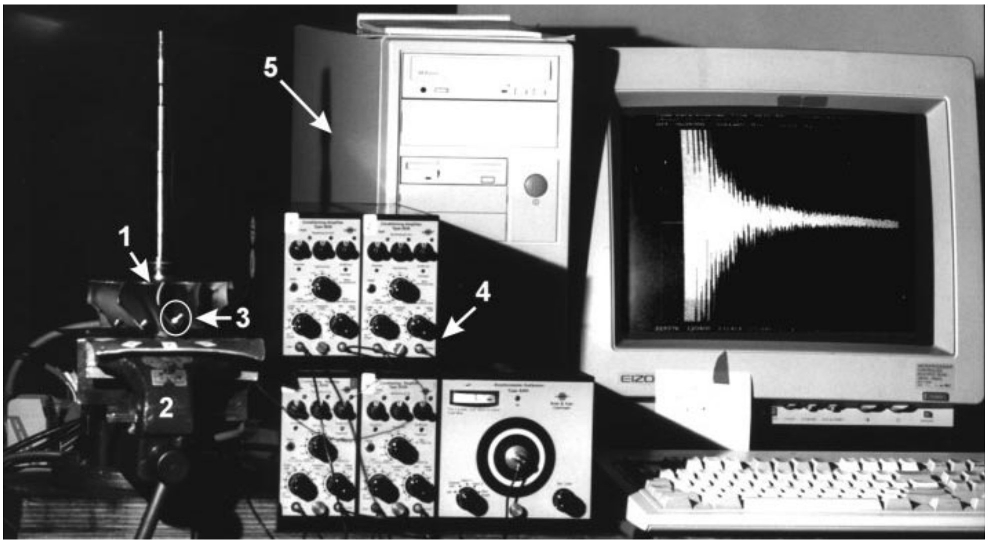
The main objectives of this study were:

- to experimentally determine the properties of structural damping of blade oscillations during free impeller-blade oscillations (bending oscillations of the blades in a blade-disc system);
- to determine the type of free oscillation which corresponds to the lowest amount of damping;
- using the logarithmic decrement method, to assess the magnitude of the damping ratio using various methods to excite natural vibrations.

## 1 DETERMINATION OF ENERGY DISSIPATION DUE TO DAMPING

In order to determine the magnitude of structural damping, we performed measurements on turbine-blade response to the initial disturbance. The measurements were performed on an impeller without a casing under atmospheric conditions in the laboratory.

The measurement system consisted of four accelerometers (Brüel&Kjær, frequency range up to 80 kHz) and matching signal pre-amplifiers. The measured signals were recorded on a computer using a 4-channel 16-bit measurement card. The resolution of the data acquisition was set to 80 kHz. The measurement system and the data acquisition system are shown in Figure 2. The time signals of the responses were measured simultaneously on 4 of the 11 blades. Figure 3 shows the positions of the accelerometers. Two pairs of accelerometers were used – the first pair with a mass of 0.2 g and the second pair with a mass of 2 g. The measurement range of all the accelerometers was greater than the calculated free frequencies by one order of magnitude of the composite blade-disc system [5]. The position of the accelerometer on the blade was selected with the aid of a previous numerical analysis of the natural vibrations, i.e. at the location of the largest strains (and accelerations) in natural vibrations of the first free form.



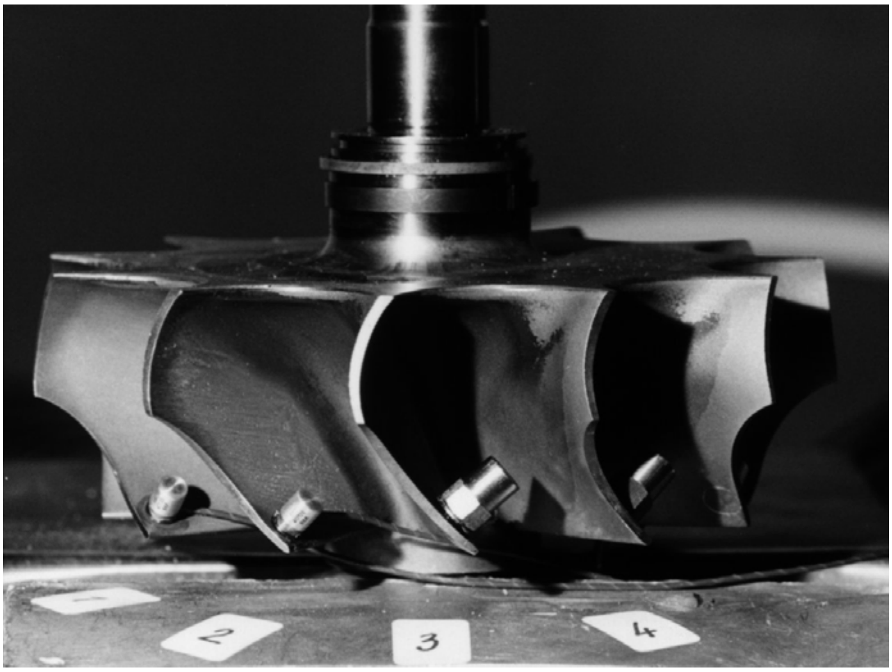

Sl. 2. *Preskuševališče in merilni sistem za merjenje odziva gonilnika na impulzno motnjo: 1 – gonilnik z gredjo; 2 – primež z vpenjalno glavo; 3 – merilnik pospeškov (eden od štirih); 4 – nabojni predojačevalniki; računalnik z merilno kartico in programsko opremo za zbiranje in obdelavo izmerjenih podatkov*
Fig. 2. *Test rig and the measurement system for measuring impeller response to impulse disturbances: 1 – impeller with shaft; 2 – clamp with clamping head; 3 – accelerometer (one of four); 4 – signal pre-amplifiers; computer with measurement card and software for acquisition and processing of measured data*

poprejšnjimi meritvami je bil preverjen vpliv mase na lastna nihanja lopatic obeh uporabljenih tipov merilnikov pospeškov. Rezultati kažejo zanemarljivo majhna odstopanja pri meritvi lastne frekvence, ki se zelo dobro ujema z izračunanimi.

Pri meritvah odzivov sta bila uporabljena dva različna načina vzbujanja:
- impulzno in
- harmonsko-akustično vzbujanje.

Preliminary measurements verified the influence of mass on the free blade oscillations in both types of accelerometer. The results show negligible deviations in the measurement of free frequency, which compares very well with the calculations.

Two types of excitation were used in the measurement of responses:
- impulse,
- harmonic–acoustic excitation.




Sl. 3. *Pritrditev pospeškomerov na lopatice gonilnika*
Fig. 3. *Attachment of accelerometers to impeller blades*

Ker vibracije lopatic povzročajo rezultirajoče dinamične sile pretežno v obodni smeri gonilnika, je treba preprečiti rotacijo osi z dovolj togim vpetjem v vpenjalno pripravo (primež s podstavkom). Prav tako so bile, v izogib kakršnimkoli vplivom interakcije dinamičnih lastnosti merilnega mesta (podstavka s primežem) z gonilnikom kot merjenim objektom, izmerjene lastne frekvence merilnega mesta samega. Rezultati so pokazali, da so le-te za dva velikostna razreda manjše od analiziranih lastnih frekvenc gonilnika.

Since blade vibrations cause dynamic forces primarily in the tangential (circumferential) direction of the impeller, axis rotation needs to be prevented by a sufficiently rigid attachment to the clamping device (vice with stand). In addition, in order to avoid any influence of the interactions between the dynamic properties of the measurement site (vice with stand) with the impeller as the measured object, the free frequencies of the measurement site itself were measured. The results show that these were smaller by two orders of magnitude than the analysed free frequencies of the impeller.

## 2 OCENA METODE ZA DOLOČITEV RAZMERNIKA DUŠENJA

## 2 ASSESSMENT OF THE METHOD FOR DETERMINING THE DAMPING RATIO

Teoretično ozadje temelji na lastnem harmonskem nihanju sistema z eno prostostno stopnjo [7]. Razmernik dušenja, ki je vpeljan v različne modele dušenja za uporabo metode končnih elementov, je določen kot razmerje med ekvivalentnim viskoznim dušenjem sistema in kritičnim dušenjem:

The theoretical background is based on free harmonic oscillation of the system with a single degree of freedom [7]. The damping ratio which was introduced into various damping models for the use of the finite-element method was determined as a ratio between equivalent viscous damping of the system and critical damping:

$$\delta = \frac{c}{c_{kr}} \qquad (1).$$

Pogost primer v tehnični praksi je, da je dušenje sistema zelo majhno, zato se frekvence lastnih nihanj dušenega in nedušenega sistema le neznatno razlikujejo. Razmernik dušenja se v tem primeru da izpeljati iz logaritemskega dekrementa:

A case frequently encountered in engineering practice is that the damping of the system is very small, therefore the frequencies of the natural vibrations of a damped and undamped system differ insignificantly. In this case it is possible to derive the damping ratio from logarithmic decrement:

$$\delta \approx \frac{\Delta}{2\pi} \qquad (2),$$

kjer je logaritemski dekrement $\Delta$ naravni logaritem razmerja sosednjih amplitud harmoničnega dela odziva sistema:

where the logarithmic decrement $\Delta$ is the natural logarithm of the ratio between adjacent amplitudes of the harmonic part of the system's response:

$$\Delta = \ln\left(\frac{x_i}{x_{i+1}}\right) \qquad (3).$$

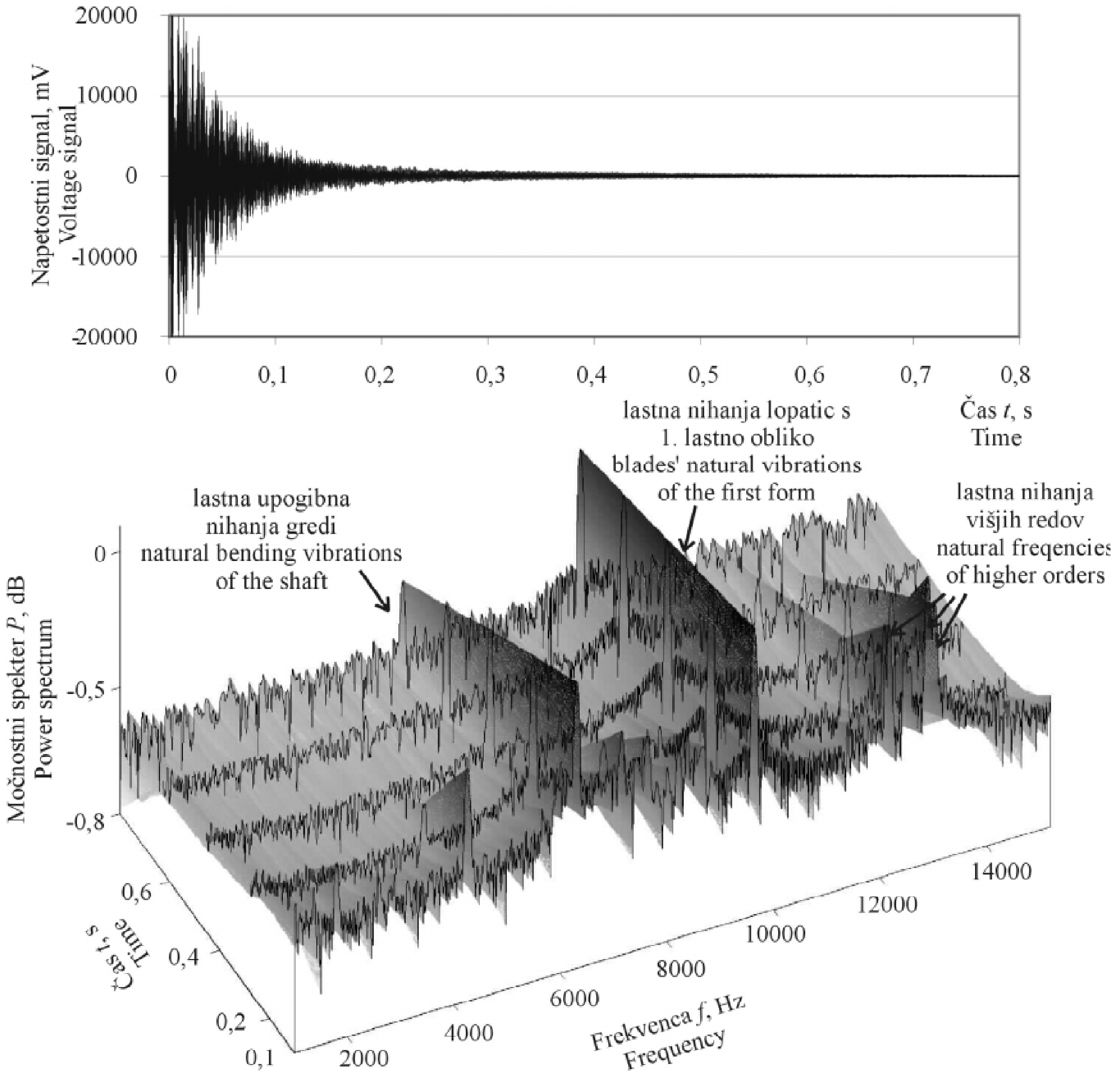
Logaritemski dekrement je bil za izmerjene signale izračunano z uporabo diskretne Fourierjeve transformacije (DFT) na ekvidistantnih intervalih signala. Kot merilo za srednjo amplitudo na intervalu je vzeta vektorska vsota koeficientov realnega in imaginarnega dela Fourierjeve vrste pri 1. lastni frekvenci (z največjo vsebnostjo v signalu). Na ta način se logaritemski dekrement izrazi iz enačbe:

For the measured signals, the logarithmic decrement was calculated using the discrete Fourier transformation (DFT) on equidistant signal intervals. As a criterion of mean amplitude on the interval we used the vector sum of the coefficients of the real and imaginary part of the Fourier series at the first free frequency (with the highest content in the signal). In this way the logarithmic decrement can be expressed by the following equation:

$$\Delta = \ln\left(\frac{a_i}{a_{i+1}}\right) \qquad (4),$$

kjer sta $a_i$ in $a_{i+1}$ srednji amplitudi dveh zaporednih intervalov merjenega signala.

where $a_i$ and $a_{i+1}$ are the mean amplitudes of two successive intervals of the measured signal.




Sl. 4. *Značilni signal odziva lopatice na impulzno motnjo (zgoraj) in pripadajoči spektrogram (spodaj)*
Fig. 4. *Characteristic signal of blade response to an impulse disturbance (above) and corresponding spectrogram (below)*

## 3 REZULTATI MERITEV

### 3.1 Impulzno vzbujanje

Frekvenčna razstavitev impulznega signala pokaže širok frekvenčni spekter, zato je ta način vzbujanja primeren za vzbujanje lastnih nihanj v širokem frekvenčnem pasu. Izvajan je bil z udarci kovinske palice po merjeni konstrukciji. Preskušeno je bilo več načinov vpetja turbinskega gonilnika v vpenjalno glavo in različna mesta izvajanja vzbujanja. Iz analize dobljenih rezultatov je bila za vse različice ugotovljena enaka frekvenčna vsebnost signalov, vendar se je izkazalo, da vpetje, kakor je prikazano na sliki 3, omogoča najboljše rezultate.

Na sliki 4 je prikazan izmerjeni signal odziva lopatic na impulzno motnjo in njegov spektrogram. Impulzna motnja vzbudi lastne

## 3 RESULTS OF MEASUREMENTS

### 3.1 Impulse excitation

The frequency decomposition of an impulse signal shows a broad spectrum of frequencies, therefore this method of excitation is suitable for exciting natural vibrations over a wide frequency band. It was performed by hitting the measured structure with a metal bar. Several methods for clamping the turbine impeller in the clamping head and various places at which excitation was performed were tested. The analysis of the obtained results showed that all the variants had the same frequency signal content, but it turned out that the clamping method presented in Figure 3 afforded the best results.

Figure 4 shows the measured signal of the blade response to an impulse disturbance and corresponding spectrogram. Impulse disturbances excite the free fre-

frekvence celotne konstrukcije (gonilnika z gredjo) v širokem frekvenčnem območju. Po pričakovanjih je večina močneje dušenih, zato se v začetnem delu odziva zelo hitro iznihajo (sl. 5, interval A), kar se lepo vidi tudi iz spektrograma na sliki 4 spodaj. V nadaljevanju signala prevladuje samo še lastna frekvenca z najmanjšim dušenjem, ki pripada 1. lastni obliki: 9,84 kHz. Lastna nihanja višjih redov sistema lopatice - disk imajo za velikostni razred večje dušenje od nihanj v 1. lastni obliki, zato v resonanci amplitude teh nihanj lopatic niso kritične. Začetni deli signalov, ki imajo vsebnost lastnih frekvenc višjih redov, so bili zatorej izločeni iz analize.

Diagram na sliki 5 prikazuje časovno odvisnost sočasno merjenih amplitud lastnih nihanj s 1. lastno obliko za 4 sosednje lopatice (pr. sliko 3). Signal je zajet v časovnem intervalu 5 s, v katerem so izmerjeni celotni odzivi na tri zaporedne impulzne motnje. V logaritemski skali ima potek amplitud (oznaka *a*) linearno naravo (sl. 5, interval z oznako B) z enakim nagibom za vse merjene lopatice. Za opazovano 1. lastno obliko je razmernik dušenja konstanten in znaša: δ = 5 E-05. Ta ugotovitev ima dvojni pomen:

- Dušenje vibracij sestavljenega sistema lopatice - disk s 1. lastno obliko je ustrezno popisano z razmernikom viskoznega trenja, kar omogoča primerno uporabo uveljavljenih modelov dinamičnega dušenja, temelječih na sistemu z eno prostostno stopnjo.
- Signali vseh štirih merilnikov pospeškov se ujemajo (izmerjena lastna frekvenca in razmernik dušenja), kar dokazuje, da so mase merilnikov pospeškov samih dovolj majhne, da zanemarljivo malo vplivajo na dinamično obnašanje lopatic.

### 3.2 Harmonsko akustično vzbujanje

Drugi način vzbujanja je bil izveden z akustično metodo. V ta namen so bili za vzbujanje uporabljeni: frekvenčni generator kot izvor električnega signala, ojačevalnik in zvočnik s primerno frekvenčno karakteristiko (sl. 6). Frekvenca in oblika (harmonska) signala sta bili nastavljeni s frekvenčnim generatorjem. Z uporabo 4-kanalnega digitalnega osciloskopa so bile najprej zaznane lastne frekvence gonilnika. Odzivi nihanja merjenih lopatic se tako po frekvenci, kakor tudi po obliki dobro ujemajo s signalom vsiljenega nihanja (akustičnega signala). Za vsako resonanco je potekalo snemanje signala takole:

- Po ugotovitvi resonance (lokalni maksimum amplitude merjenega odziva), se je začela meritev.

quencies of the entire structure (impeller and the shaft) over a wide frequency band. As expected, most of them are strongly damped, therefore in the initial part of the response they dissipate very rapidly (see also Figure 5, interval A), which can also be seen nicely from the spectrogram in Figure 4. Later on in the signal, only the free frequency with the smallest damping predominates, which belongs to the first free form: 9.84 kHz. Natural vibrations of higher orders of the blade-disc system have greater damping, by an order of magnitude higher than the natural vibrations in the first free form, therefore at resonance frequency the amplitudes of these blade oscillations are not critical. The initial part of the signals, which contained free frequencies of higher orders, were therefore eliminated from the analysis.

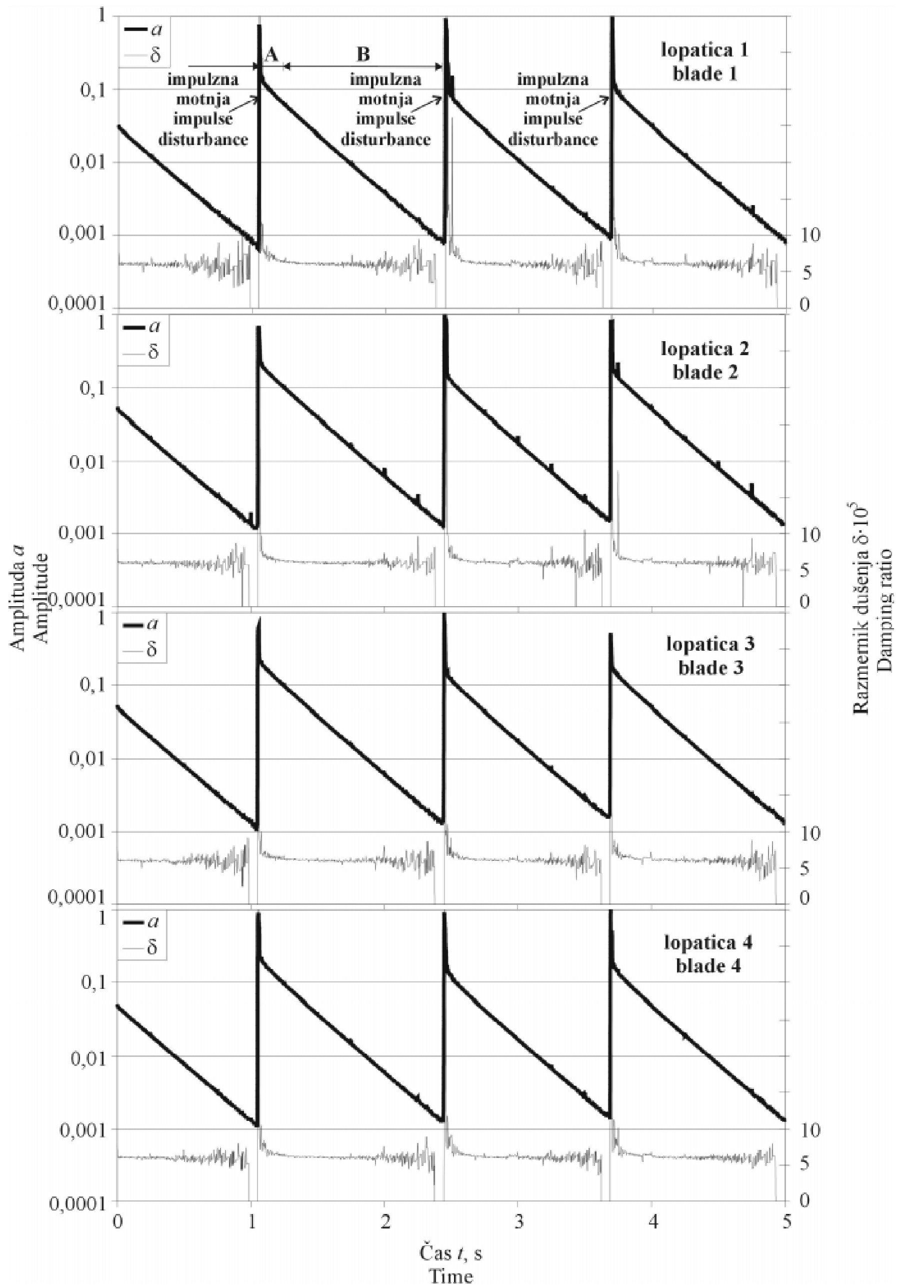
The diagram in Figure 5 shows the time dependence of simultaneously measured amplitudes of natural vibrations with the first free form for 4 adjacent blades (cf. Figure 3). The signal occurs within a time interval of 5 s, during which the entire responses to three consecutive impulse disturbances were measured. On the logarithmic scale, the course of amplitudes (*a*) is linear (Figure 5, interval designated B) with the same slope for all the measured blades. For the observed first free form, the damping ratio is constant and amounts to δ = 5 E-05. This finding has a twofold meaning:

- The damping of vibrations of a composite blade-disc system with the first free form is described satisfactorily by the viscous friction ratio, which enables the convenient use of the established models of dynamic damping based on a system with one degree of freedom.
- The signals of all four accelerometers agree (the measured free frequency and the damping ratio), which proves that the masses of the accelerometers themselves are sufficiently small so as to have a negligible effect on the dynamic behavior of the blades.

### 3.2 Harmonic acoustic excitation

The following elements were used for acoustic excitation: a frequency generator as the source for the electrical signal, an amplifier and a loudspeaker with the appropriate frequency response characteristics (Figure 6). The frequency and the shape (harmonic) of the signal were set using the frequency generator. By means of a 4-channel digital oscilloscope, the impeller's free frequencies were first detected. The responses in the form of oscillations of the measured blades agreed well, both in terms of frequency and shape, with the signal of the forced oscillation (acoustic signal). For each resonance frequency, the signal was recorded as follows:

- When resonance was established (local maximum of the measured response amplitude), measurements began.




Sl. 5. *Izmerjene vrednosti amplitud odziva štirih sosednjih lopatic na tri zaporedne impulzne motnje ter pripadajoči razmernik dušenja*

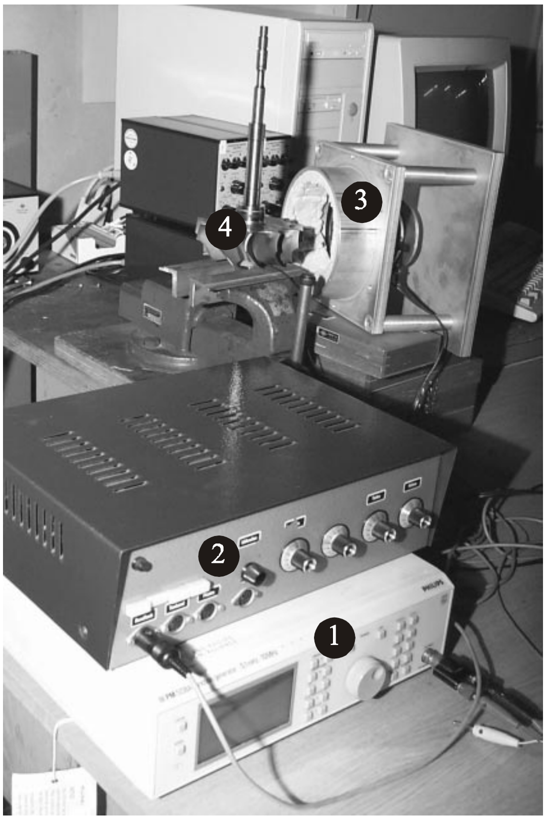
Fig. 5. *Measured values of amplitudes of the response of four adjacent blades to three consecutive impulse disturbances and the corresponding damping ratio*

- Med meritvijo je bil akustični signal prekinjen s stikalom na zvočniku. Dobljeni odziv lastnega nihanja je bil uporabljen za analizo signala.

Glede na pričakovanja iz prejšnjih numeričnih analiz [5] in frekvenčne razstavitve odzivov pri impulznem vzbujanju je bilo območje med 8,5 kHz in 11,5 kHz, kjer ležijo vse lastne frekvence, ki pripadajo 1. lastni obliki, podrobno raziskano. Med kar nekaj resonancami v tem frekvenčnem območju je zanimiva tista, pri frekvenci 9,64 kHz, saj ima od drugih za velikostni

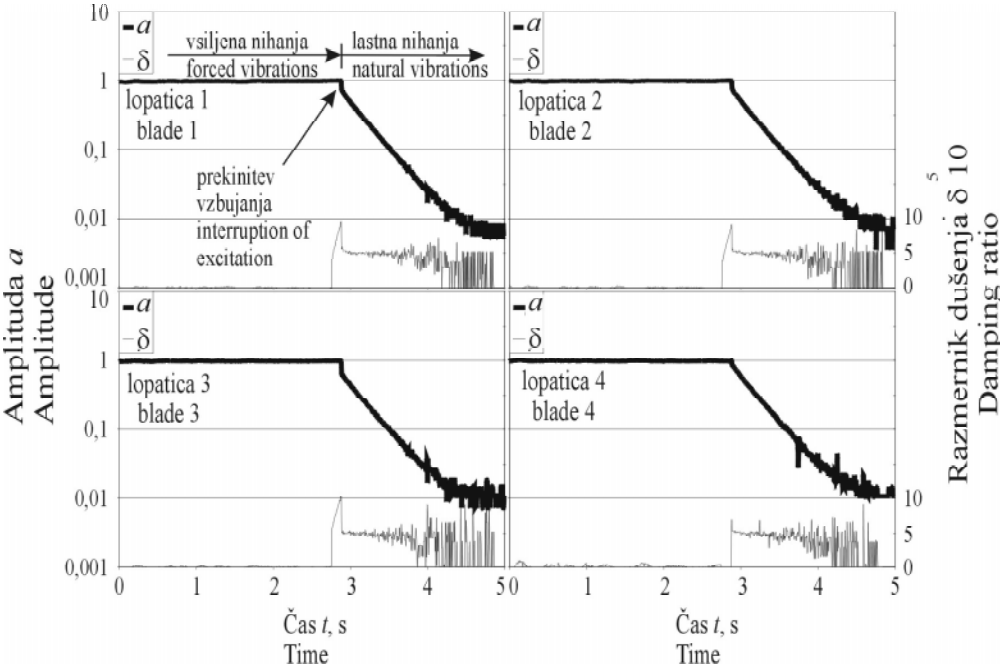
- During the measurement, the acoustic signal was interrupted using a switch on the loudspeaker. The obtained free oscillation response was used for signal analysis.

Based on our expectations from previous numerical analyses [5] and the frequency decomposition of responses in impulse excitation, the frequency range between 8.5 kHz and 11.5 kHz was studied in detail, because all the free frequencies corresponding to the first free form were located in this range. Of the many resonant frequencies in this range, the most interesting one was at 9.64 kHz, because its damping was smaller than




Sl. 6. *Sistem za akustično harmonsko vzbujanje: 1 – frekvenčni generator; 2 – ojačevalnik; 3 – zvočnik; 4 – turbinski gonilnik z gredjo*

Fig. 6. *System for acoustic harmonic excitation: 1 – frequency generator; 2 – amplifier; 3 – loudspeaker; 4 – turbine impeller with shaft*




Sl. 7. *Izmerjene vrednosti amplitud odziva štirih sosednjih lopatic na harmonsko vsiljeno nihanje ter pripadajoči razmernik dušenja*

Fig. 7. *Measured values of response amplitudes for four adjacent blades – harmonic forced oscillation and the corresponding damping ratio*

razred manjše dušenje in enako obliko, kakor pri impulznem vzbujanju. Kakor je razvidno iz diagramov na sliki 7, ki prikazujejo odzive lastnega nihanja 4 sosednjih lopatic, so nagibi amplitudnega odziva sistema, prikazanega v logaritemski lestvici enaki, kakor pri impulznem vzbujanju, zatorej je razmernik dušenja enak: $\delta = 5$ E-05.

V primerjavi z omenjeno lastno frekvenco, so preostala nihanja povezana z znatno večjim dušenjem. Razen tega ima dušenje teh lastnih frekvenc močno nelinearno karakteristiko, zato ga ni mogoče popisati z uporabljenim modelom. Zaradi velikosti dušenja je bilo vzeto, da te lastne frekvence niso kritične pri obravnavi vibracij med obratovanjem in so bile zato izvzete iz obravnave.

## 4 SKLEPI

V raziskavi so bile analizirane dinamične lastnosti vibracij gonilnika turbopuhala, ki predstavlja sestavljen sistem lopatice - disk.

Eksperimentalno je bil določen koeficient dušenja, in sicer z dvema različnima načinoma vzbujanja sistema: z impulzno motnjo in z akustičnim harmonskim vzbujanjem. Iz analize izmerjenih signalov je mogoče povzeti naslednje:

– lastna nihanja višjih redov se zaradi relativno velikega dušenja zelo hitro iznihajo;
– v signalu po iznihanju lastnih frekvenc višjih redov ostane samo še nihanje prvega reda (1. lastna oblika).

V obeh primerih vzbujanja se za lastna nihanja s 1. lastno obliko izkaže linearna narava dušenja, pri čemer znaša razmernik dušenja za obravnavano gradivo $\delta = 5$ E-05.

Eksperimentalno dobljeni podatki o dušenju so koristna informacija za oblikovanje numeričnega modela, katerega namen je simuliranje in analiza obratovanja gonilnika turbopuhala v resonanci in njeni okolici.

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that of the others by an order of magnitude, and its shape was the same as in impulse excitation. As can be seen from the diagrams in Figure 7, which show the responses of the free oscillation of 4 adjacent blades, the slopes of the amplitude response of the system shown on the logarithmic scale are the same as in impulse excitation, therefore the damping ratio is the same: $\delta = 5$ E-05.

In contrast to the above-mentioned free frequency, the other oscillations are associated with a considerably greater damping. In addition, the damping of these free frequencies has a strongly nonlinear characteristic, therefore it cannot be described using the above model. Due to the magnitude of the damping it is assumed that these free frequencies are not critical in the study of vibrations during operation and were therefore excluded from the analysis.

## 4 CONCLUSION

This study analyses the dynamic properties of the vibrations of a turbocharger impeller, representing a composite blade-disc system.

The damping coefficient was determined experimentally using two different methods of exciting the system: impulse disturbance or acoustic harmonic excitation. The analysis of the measured signals leads to the following conclusions:

– natural vibrations of higher orders dissipate very quickly due to a relatively large degree of damping;
– after free frequency oscillations of higher orders have dissipated, the signal contains only oscillations of the first order (first free form).

In both methods of excitation, natural vibrations of the first free form exhibit linear damping, whereby the damping ratio for the studied material is $\delta = 5$ E-05.

Experimentally obtained data on damping can serve as useful information for the development of a numerical model for the purposes of simulation and analysis of the operation of turbocharger impellers at or near resonance frequency.

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