Study of the vibration state (natural frequency spectrum) of the rotor of a microturbine unit with a capacity of 100 kW

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Abstract. The vibrational state of the rotor of a turbine installation, as is known, determines the level of reliability and the resource of its operation. In this article, by the vibrational state of the rotor we understand the vibrational state of all its elements, including the vibrational state of the impellers of the turbine stages. An acceptable level of vibrational state of the rotor impellers (RI) of the turbine stages is provided by various factors, among which there is the most important one - the presence of vibrational detuning between the natural frequencies of the RI and the frequencies of the disturbing flow effect, which are usually multiples of the rotor speed [1]. From what has been said, there is a substantial need for an accurate determination of the eigenfrequencies of turbine impellers. The accuracy of determining the frequencies is achieved by comparing the results of the calculations with the results of the experimental determination of the indicated natural frequencies.

1. Introduction

In the production of power plants, the issue of ensuring the reliability of the impellers is solved in several stages: Stage 1 - calculation of the natural frequencies of individual blades and impellers; Stage 2 - direct measurement of the natural frequencies of single working blades (WB); Stage 3 - direct measurement of the natural frequencies of stationary impellers with typed impellers; Stage 4 - experimental determination of the natural frequencies of the assembled impellers at an operating speed of rotation on the Campbell machine. (Stage 5 - the possible measurement of the frequency spectrum of the vibration of the supports of the assembled rotor at the operating speed of rotation during its high-speed balancing in the vacuum chamber of the Schenk machine). The main feature of the design of the rotors of microturbine plants is the use of impellers made at the same time with the blades, and also often at the same time with the installation shaft [2-10]. Considering the indicated design features, it is advisable to conduct vibrational studies of the impellers of microturbines directly on the assembled rotor, bypassing the preliminary stages of vibrational studies indicated above.

2. Methods

To perform the indicated vibrational studies of microturbine rotors, it is necessary to use highly sensitive measuring equipment with an extended frequency range up to 30 kHz and higher. An additional very
important point for microturbines is the ability to perform non-contact vibration measurements [11-15]. In this work, we consider laser scanning Doppler vibrometer from Politek PSV-400 (LDV) [16] as such equipment. The specified vibrometer meets the above requirements.

The frequency measurement in the device is based on the use of the Doppler effect. The scheme for measuring the vibration frequency using the indicated device is shown in fig. 1 and explained below.

![Figure 1. Scheme for measuring the frequency of a vibrating surface using a laser Doppler vibrometer [11].](image)

A beam from a helium-neon laser, characterized by a frequency $f_0$, passes through a system of optical prisms and falls on a vibrating surface, indicated in fig. 1 by $x(t)$. The reflected beam (in fig. 1 it is a blue beam) is characterized by some frequency shift $f_0 \pm f_D$; the amount of displacement is determined by the magnitude of the vibration velocity of the reflecting surface $v(t)$. The detector perceives a total beam, consisting of a reference (red) and reflected (blue) beam. The total beam is characterized by beating, the value of which is determined by the frequency offset of the reference and reflected beam. The detector perceives the total beam as a system of concentric light and dark spots shown in fig. 2.

![Figure 2. The optical form of the interaction of direct and reflected rays at the detector of a laser vibrometer [16].](image)

When using a PSV-500-H laser vibrometer, the maximum vibration frequency of the studied structures can be from 0 Hz to 100 kHz with surface vibration velocities from 10-5 m/s to 10 m/s [16]. The indicated parameters provide overlapping of the values of frequencies and vibration velocities of structural elements encountered in turbine construction.

Given the positive experience of Polytec in the field of frequency measurements on the rotors of small turbomachines, we attempted to measure the natural frequencies of the rotor of a gas turbine plant with a capacity of 100 kW. The overall dimensions of the rotor of the GTU 100 kW unit are shown in fig. 3. The number of blades on the impeller of the Z2 turbine is 17.

![Image of turbine](image)
3. Results and discussion

Some results of vibration studies of a rotor of GTU 100 kW, obtained using a PSV-400 scanning laser meter, are presented in fig. 4 and 5. Fig. 4 shows the visualization of the first knotless form of the rotor mounted on the mounting supports. The impellers of the compressor and turbine are involved in the oscillations in the indicated form as massive nondeformable bodies.

Figure 4. The oscillation form of the rotor of GTU 100 kW, corresponding to a frequency of 388 Hz. The maximum vibration amplitude is 36 μm / s. (view of the rotor from the side of the turbine impeller).

As a result of the measurements, a spectrum of natural vibrations of the rotor of GTU 100 kW was also obtained. Fig. 5 shows the low-frequency part of this spectrum. The spectrum contains discrete components corresponding to the lower rotor form considered above (388 Hz), as well as discrete components corresponding to the lowest forms of natural vibrations of the turbine impeller (1900 … 2400 Hz).
To clarify the structure of the spectrum in the range 1900 ... 2400 Hz, it is necessary to refer to the analysis of the results of calculations of the frequencies and forms of the centrifugal turbine of the turbine under consideration in the GTU 100 kW performed in the ANSYS program [17-21]. It can be seen from the vibration diagram shown in Fig. 6 that the frequency range of the first in-phase form of the RI corresponds to the indicated frequency range; a general view of this form is shown in Fig. 7.

![Vibration spectrum of GTU 100 kW rotor.](image)

**Figure 5.** Vibration spectrum of GTU 100 kW rotor.

![Vibration diagram of the RI turbine turbine 100 kW.](image)

**Figure 6.** Vibration diagram of the RI turbine turbine 100 kW.
Let us consider the possible reasons for the presence of several discrete components close in frequency in the frequency range under consideration of 1900 ... 2400 Hz. The first reason may be the presence of several similar in frequency forms of the RI on the basis of one form of the working blade (in this case, knotless). The second reason may be the presence of some slight different frequencies of the adjacent working blades of the RI [22]. Given the high accuracy of manufacturing the RI on a CNC machine, we will consider the first reason more significant. Let's consider it in more detail. As the calculations in ANSYS showed, the number of similar types of impellers of a centripetal turbine is determined by the number of working blades \( Z_2 \). With an even number of working blades, the number of such forms will be equal to \( Z_2 / 2 \) and these forms will be distinguished by the number of nodal diameters [1]. The lowest-frequency form will be a form with one nodal diameter, the higher-frequency form will be a form with two nodal diameters, etc. The highest frequency of the indicated homogeneous forms will be the form with common-mode vibrations of all working blades (a similar form is shown in fig. 7). A visual representation of these uniform forms with a different number of nodal diameters is given by the frequency map of a single-type RI of centripetal turbine with the number of blades \( Z_2 = 18 \) (see fig. 8). Here, the shape number is plotted along the horizontal axis, and each number formally corresponds to two numbers (oscillations along the X and Y axes). The eigenfrequencies of the considered RI are plotted along the vertical axis. The frequencies shown on the map can be grouped by type of blade shape. A conventional view of the shapes of the blades is shown on the left. Each scapular form forms the basis of a group of RI forms with a different number of nodal diameters, where this number is \( m = 0, 1, 2, ..., Z_2 / 2 \). The indicated forms of the RI with a different number of nodal diameters are conventionally drawn in the middle of the figure. Thus, each RI form with in-phase oscillations of the blades corresponds to a group of lower-frequency forms that differ from the in-phase form by the presence of nodal diameters from 1 to \( Z_2 / 2 \) (with an odd number of blades this number is \( Z_2 / 2 + 1 \)).
Considering the fact that the number of rotor blades on the impeller of the considered GTU 100 kW is 17 \((Z_2 = 17)\), there should be nine corresponding fan forms of the turbine RI. As can be seen from the spectrum shown in fig. 5, the frequency range of 1900 \(\ldots\) 2400 Hz corresponds to nine discrete components. Thus, the number of predicted and available on the spectrum of discrete components coincided.

4. Conclusions
The obtained experimental and calculated results [23-26], as well as an attempt to jointly analyze them, are of scientific and methodological significance, since they realize the desire to obtain mutual confirmation of the obtained values of the natural frequencies of the RI. This approach will guarantee reliable vibrational detuning of the GTU 100 kW rotor and its main elements, and therefore ensure the reliability of the GTU. The materials presented in the article indicate that the laser scanning method allows one to determine with high accuracy the natural frequencies and shapes of the impellers of low-consumption gas turbine plants.

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