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# MEASUREMENTS AND VIBRATION ANALYSIS OF A FIVE-STAGE AXIAL-FLOW MICROTURBINE OPERATING IN AN ORC CYCLE

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Abstract

This paper presents results concerning the experimental investigation of a five-stage prototypical axial-flow microturbine operating in an ORC installation. The microturbine has an electrical capacity of 3 kW at the nominal rotational speed of 12000 rpm and is supplied with a low-boiling medium's vapour HFE7100. During the test, both the full vibration frequency distribution and the overall level of the velocity vibrations RMS in the microturbine were measured. A spectral analysis of vibration velocity was conducted during the grind-in process after a modernisation of the shrouding bandage of the microturbine set's flow system took place. The article presents the operational characteristics of the ORC system as well as the characteristics showing the electric power produced as the function of rotational speed of the microturbine. On the basis of the results obtained, there was assessed the dynamic performance of the microturbine and it was ranked according to ISO 10816-1 standard. The paper also shows the measurement results of the so-called kurtosis within the frequency range of 15 to 30 kHz, which made it possible to evaluate the dynamic state of the rolling bearings by which the microturbine rotor was supported.

Keywords: five-stage axial-flow microturbine, distribution of vibration level, HFE7100, ORC

## POMIARY I ANALIZA POZIOMU DRGAŃ PIĘCIOSTOPNIOWEJ MIKROTURBINY OSIOWEJ PRACUJĄCEJ W OBIEGU ORC

Streszczenie

W artykule przedstawiono wyniki badań poziomu drgań podczas rozbiegu prototypowej 5-stopniowej osiowej mikroturbiny pracującej w obiegu Rankine'a. Mikroturbina o nominalnej mocy 3 kW i prędkości obrotowej 12000 obr/min była zasilana parą czynnika niskowrzącego HFE7100. Podczas badań zmierzono rozkłady częstotliwościowe oraz ogólny poziom prędkości drgań Vrms mikroturbiny w zależności od prędkości obrotowej. Przedstawiono analizę prędkości drgań podczas docierania i po modernizacji bandaży układu przepływowego mikroturbozespołu. Zamieszczono charakterystyki pracy obiegu ORC oraz charakterystyki mocy elektrycznej w zależności od prędkości obrotowej mikroturbiny. Na podstawie uzyskanych wyników oceniono stan dynamiczny mikroturbiny oraz zaklasyfikowano badaną maszynę przepływową wg normy ISO 10816-1. W pracy zamieszczono wyniki pomiarów tzw. kurtozy w przedziale 15 – 30 kHz na podstawie której wykonano ocenę stanu łożysk tocznych układu wirującego mikroturbiny.

Słowa kluczowe: pięciostopniowa mikroturbina osiowa, rozkład poziomu drgań, HFE7100, ORC

## 1. INTRODUCTION

Bearing in mind the guidelines laid down in Directive 2009/28/EC [1], a greater intensity has been observed in actions aimed at fighting global climate change and reducing $CO_2$ emissions on the territory of the European Union. For this reason, capital expenditures for promoting CHP systems (including ORC systems that use RES) have been increased, and some Member States introduced subsidies for the purchase of such systems. The subsidies for the purchase of modern heating systems (e.g. gas-fired boilers or solid-fuel boilers) as well as an increase in fuel and energy carrier prices are a major stimulus for innovating and modernising outdated domestic heating installations. The distributed co-generation enabling the production of electricity, heat and coolness-coupled with an on-site consumption is an opportunity for the EU and the Member States. Micro CHP systems that offer power capacities up to 10 kW can meet the demand for electricity and heat of single-family houses [2,3]. By installing such a system the owner of a single-family house is able to make savings as a result of the reduction of electrical and thermal energy transmission. Moreover, the application of ORC technologies in the small user sector (including homesteads) can contribute to an improvement in welfare and energy security on the local electricity and heat markets [4]. It follows from the data presented in [5,6], that coal use in households and in the small user sector has accounted for over 30 % of all energy use in Poland, and this has been the case over the past few years. Therefore, there are reasonable grounds to believe that the consequent placing on the market of products based on ORC technologies should be financially supported by governmental institutions. The growing interest in CHP-ORC systems inspires to look for new structural solutions that will meet the needs of European and global markets. On the basis of the literature review, we can say that there are many CHP systems equipped with positive displacement expanders (e.g. scroll, screw, vane and piston expanders) and turbines [7]. However, most of the small power ORC systems (up to 10 kW) are equipped with volumetric expanders since turbine expanders are available only to a limited extent [8]. It should also be noted that, in general, volumetric expanders have higher noise and vibration levels than microturbines [9].

In the interests of the health and safety of users, a micro-CHP installation [10] situated in a homestead should comply with a set of quality criteria. One of these criteria is achieving low noise and vibration levels. This is of great importance for inhabitants, in terms of their health, well-being as well as the quality of their work and work environment [11]. In accordance with Directive 2006/42/EC [12], the Ordinance of the Minister of Economy [13], recommendation included in the paper [14] and with relevant European Union requirements [15], noise and vibration level of machinery should be reduced at source to the lowest possible level taking account of technical progress and the availability of measures to reduce noise. Therefore, the continued development of methods for vibration analysis of machinery [16,17], which, for example, allow for time waveform (TWF) analyses or monitoring of machinery using Fast Fourier Transform (FFT) algorithms [18]. The wavelet entropy method is also sometimes applied for the purposes of vibration analysis. Two methods, namely the Wavelet Space Feature Spectrum Entropy (WSFSE) and the Wavelet Energy Spectrum Entropy (WESE), were used in paper [19] and they make it possible to depict instantaneous characteristics of a vibration signal and expand this signal in terms of wavelet functions which are localised in both time and frequency domains. Monitoring of vibrations became a worldwide spread practice in the energy sector since their excessive level causes faster mechanical wear and tear of machinery, which results in failures and delays in the production process [20]. In the case of vibrations of a vapour turbine, the deterioration of its nominal internal efficiency takes place, depending on the severity of the degradation of the turbine set's and flow system's constituent parts. It is, therefore, necessary to monitor the wear and tear process so as to reliably prevent or manage standstill periods [21].

For example, excessive noise and vibration levels of a 70 MW turbine were identified using a vibroacoustic analysis [22]. This was caused by an incorrectly selected bearing system that could have led to the machine damage. The aerodynamic self-excited vibrations were damped by applying four-key bearings which also contributed to shifting the stability threshold allowing for a reliable operation of the rotor within a wider range of displacements. Article [23] discusses the vibration problems which were observed in a three-stage 10 MW steam turbine and an eight-stage syngas compressor. The analysis conducted within the rotational speed range of 2000 to 12000 rpm indicates that the first and the second critical speed of the compressor occurred at 4320 rpm and at 6200 rpm, respectively. The maximum vibration level of the turbine's components was about 1.5 mm at the rotational speed of 11580 rpm, while the vibration was approximately 16 mm/s at the speed of 4000 rpm. The authors of paper [24] described a faulty operation of a turbine set equipped with a 600 kW Kaplan turbine, which operated at a hydroelectric plant. The research was carried out within the power range of 105 kW to 480 kW and on its basis, it was noticed that the permissible RMS vibration level had been exceeded. The maximum permissible values for the machines falling within the group no. 4 (in accordance with ISO 10816-1: 1995 and ISO 10816-5 standards) are $A_{P\text{-}P}$ = 30 μm, $V_{RMS}$ = 1.6 mm/s for bearings mounted on the foundation frame and $A_{P\text{-}P}$ = 65 μm, $V_{RMS}$ = 2.5 mm/s for bearings mounted off the foundation.

Having analysed the information [25] provided in Polish and international standards (PN and ISO), relating to vibroacoustic diagnostics of fluid-flow machinery, we came to the conclusion that ISO 10816-3:2009 and PN-ISO 10816-1:1998 standards [20] are recommended for analysing industrial machines with nominal power above 15 kW. However, guidance for evaluating vibration severity in small machines (e.g. electric motors, pumps, generators, steam and gas turbines, turbo-compressors, turbo-pumps, fans, etc.) operating in the 10 to 200 Hz (600 to 12000 rpm) frequency range is provided in ISO 2372 standard [26]. From today's point of view, there is no existing standard that can be applied for a vibroacoustic assessment of small expansion devices (with nominal power below 15 kW) that are used in CHP systems. Moreover, a wide range of nominal speeds of expanders (turbine and volumetric ones), namely from 800 rpm [27] to 100000 rpm [28], creates additional difficulties. Therefore, an analysis based on a computation model should be undertaken already at design stage [29,30].

On the basis of the literature review, we may conclude that practically no data is available relating to experimental vibration analysis of small energy machinery (with nominal power below 5 kW). Meeting the requirements of relevant Community legislation (stated in Directive 2006/42/EC [31]) as well as satisfying the terms and conditions set down in the Ordinance of the Minister of Economy dated August 2, 2005 [32] and the Ordinance of the Minister of Labour and Social Policy dated November 29, 2002 [33], an evaluation of the dynamic performance of a 3 kW axial-flow microturbine incorporated into a CHP ORC-based installation (with a low-boiling working medium known under a trade name HFE7100) has been carried out.

## 2. TEST BENCH OF THE ORC SYSTEM

The ORC installation is composed of three basic cycles: heating cycle, cooling cycle and working fluid cycle. The scheme of the ORC installation with an axial microturbine is presented in Figure 1.

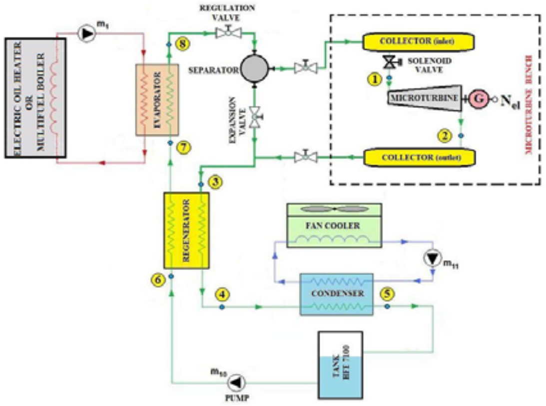


Fig. 1. Scheme of the ORC installation with an axial microturbine

The heating cycle consists of a group of oil pumps and two independent heat sources: a multi-fuel boiler and a set of two electric thermal oil heaters that can operate independently or in series/in parallel. The ORC installation can operate using an expansion valve, a microturbine or a group of expanders. The prototypical electric instantaneous heater that heats up thermal oil consists of two modules of the nominal power of 2x24 kW. The prototypical boiler enables combustion of a gaseous fuel or biomass pellets. The maximal boiler power during biomass combustion is about 30 kW. The working medium HFE7100 cycle is composed of the following components: the evaporator, regenerator, pump, condenser and measurement equipment. The third cycle in the ORC installation is the cooling system consisting of fan coolers, glycol pumps with inverters, heat exchangers and piping. The cooling system of the ORC installation performs two tasks. First, it enables cooling of the thermal oil coming to the evaporator, and thus increases the range of adjustment of oil temperature. The other important task of the cooling system is quick cooling of the HFE7100 vapour in the condenser.

### 2.1. Axial microturbine

The microturbine was created in the framework of cooperation within the national project POIG.01.01.02-00-016/08 realised by two institutions - namely the Institute of Fluid Flow Machinery, Polish Academy of Sciences and the Gdańsk University of Technology. Within the scope of this work, the following tasks were completed: analysis of the flow system [34], selection of a working medium [35] and strength computations of the microturbine components [36]. Figure 2 depicts a five-stage axial microturbine connected into the ORC system.

The microturbine set comprises the casing with inlet and outlet channels, a set of guide vanes, the shaft with a microturbine and sealing disks, a single radial bearing, two angular bearings and the electric generator. The shaft, microturbine rotor`s packing rings and bearing casing of the axial microturbine are shown in Fig. 3. All the rotor blades and guide vanes were banded together very tightly and the packing rings - made of PTFE - acted as contact seals (with the clearance not exceeding 0.01 mm).

Fig. 2. Axial-flow microturbine connected with the ORC system [37]

The turbine shaft was supported by ceramic rolling bearings. A single bearing sold under the symbol 6004 was applied at the high-pressure side, while the bearing pair in X arrangement (angular contact ball bearings, symbol: 7204BE) was mounted at the low-pressure side [38]. The basic parameters of the axial-flow microturbine, laid down during the design stage, are shown in Table 1.

Table 1. Design parameters of the axial microturbine

| | |
|---|---|
| working fluid | HFE7100 |
| mass flow rate [kg/s] | 0.170 |
| inlet pressure [bar] | 12 |
| inlet temperature [°C] | 162 |
| rotational speed [rpm] | 0 ÷ 12000 |
| stage diameter [mm] | 100 |
| blade height [mm] | 10 |
| nominal power output [kW] | 3 |
| number of stages [-] | 5 |

Due to a high nominal pressure and a low flow rate of the HFE7100 working medium, the microturbine was constructed with partial admission. The microturbine bearings were lubricated only with the HFE7100 working medium.

Fig. 3. Photo showing the shaft, microturbine rotors, bearing casing and packing rings of the microturbine [37]

## 3. EXPERIMENTAL RESULTS

During conducting the research, the maximal absolute pressure of the working medium's vapour at the microturbine inlet was 650 kPa at the temperature of 165 °C (Fig. 4).

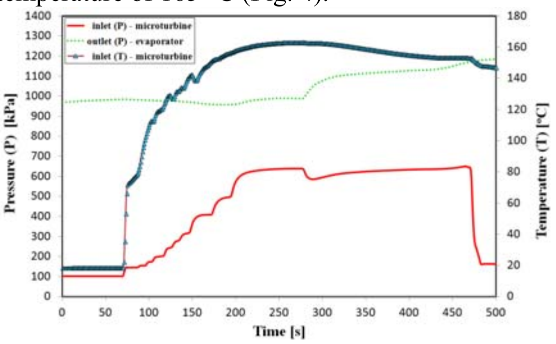


Fig. 4. Pressure and temperature of the HFE7100 vapour at the microturbine inlet vs. time

The average electrical power produced by the microturbine as the function of its rotational speed is shown in Fig. 5.

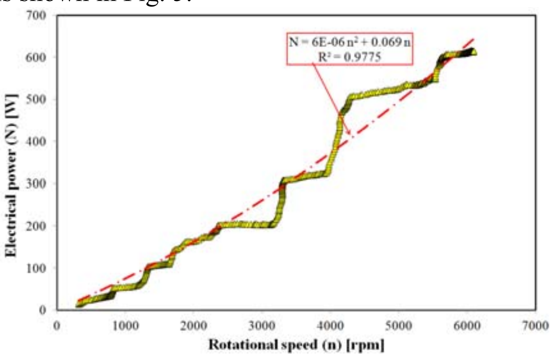


Fig. 5. Average electrical power generated by the microturbine vs. rotational speed

The electrical power produced by the microturbine increases as the rotational speed rises. The maximal electrical power was 615 W and was achieved at the speed of 6090 rpm. The electric power generated by the microturbine was much lower than its nominal power. This was mainly due to a high friction occurring between individual stages of the microturbine. The packing rings made of PTFE extended their volume (under the influence of temperature and pressure) eliminating clearances between the stages, which resulted in rubbing between the rings and surfaces of the microturbine rotors. Furthermore, it also resulted in the decrease in cross-section of the channels, which in turn caused a significant drop in supply pressure.

Within the scope of the research, vibration measurements were performed using the mobile vibration analyser DIAMOND 401AXT [39] coupled with a uniaxial piezoelectric accelerometer. The vibration sensor was mounted on the microturbine casing close to the bearing, at the supplying side. The exemplary measurement results in the form of vibration velocity frequency spectra, registered at the run-up, are presented in Fig. 6 - 8.

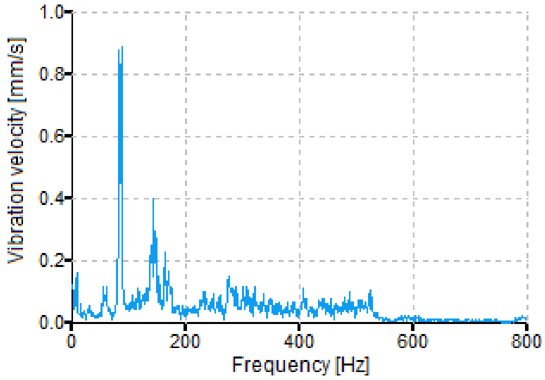


Fig. 6. Vibration velocity frequency spectrum measured on the microturbine casing at the speed of 5100 rpm

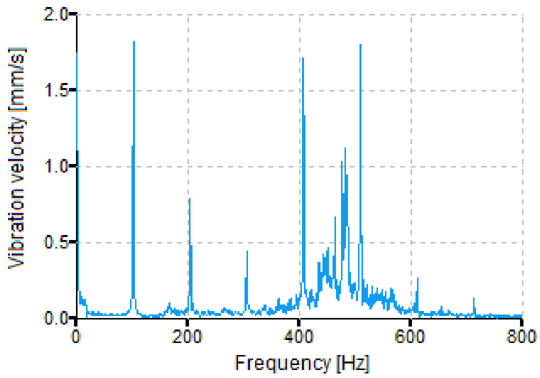


Fig. 7. Vibration velocity frequency spectrum measured on the microturbine casing at the speed of 6120 rpm

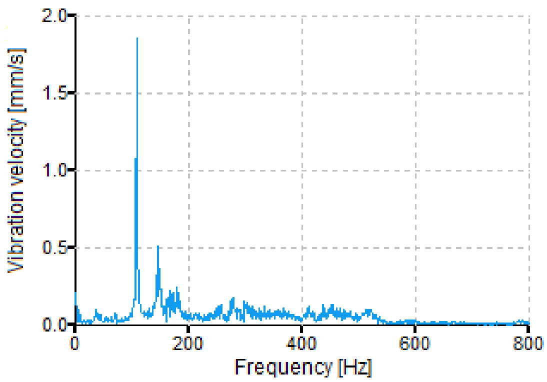


Fig. 8. Vibration velocity frequency spectrum measured on the microturbine casing at the speed of 6540 rpm

The vibration velocity spectra that are shown in Fig. 6 – 8 have synchronous components and their harmonics cover broad frequency ranges within which higher vibration levels can be observed. Both sub- and superharmonic components component are present on this plots, resulting from changes in rotational speed and properties of the rotating and supporting systems - which could have been caused by wear and tear of Teflon packing rings, microturbine disks and guide vanes. The RMS vibration velocity values as the function of rotational speed of the microturbine measured during the run-up are depicted in Fig. 9.

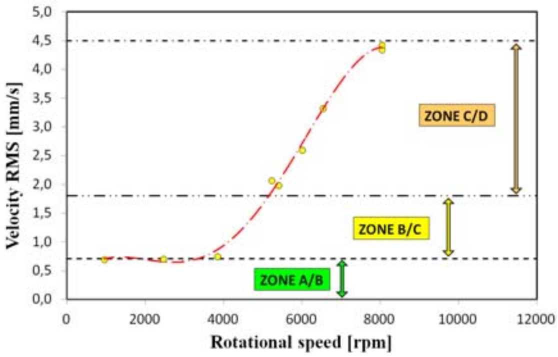


Fig. 9. RMS vibration velocity vs. rotational speed of the microturbine

It was noted that the RMS vibration velocity values increase as the rotational speed rises. Let us also note that the RMS vibration velocity values were below the value of 0.71 mm/s up to the rotational speed of 3500 rpm, i.e. within the zone A/B (in accordance with ISO 10816-1 standard). Within the speed range of 3500 to 5200 rpm, these values were within the zone B/C that corresponds to the values from 1.8 mm/s to 4.5 mm/s. Above the speed of 5200 rpm, the RMS vibration velocity values increased as the rotational speed increased (zone C/D) and this situation created a risk of damage to the machine. Therefore, the microturbine run-down was initiated immediately.

Due to a low level of electrical power generated and a unsatisfactory vibration level, it was decided to increase the clearances of the microturbine blades. During carrying out the measurements, the maximal pressure and temperature of the working medium at the microturbine inlet did not exceed the value of 860 kPa and 170 °C, respectively.

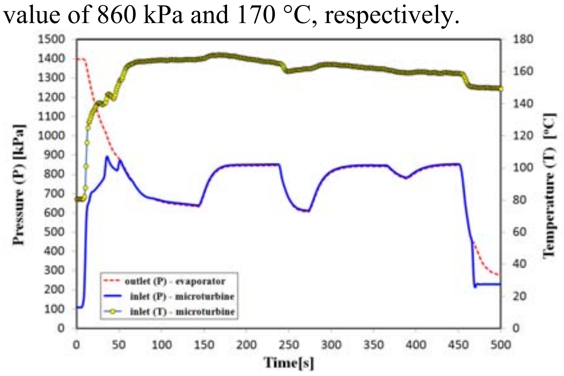


Fig. 10. Pressure and temperature of the HFE7100 vapour at the microturbine inlet vs. time

According to Fig. 10, the increase of clearances in the flow system reduced pressure losses that had been occurring at the microturbine inlet. The pressure values after the evaporator and at the pressure supply point of the machine (when the valve is opened completely, i.e. approximately 50 seconds passed) are virtually the same.

Figure 11 shows the electrical power generated during the measurement series no. 1 and no. 2 as the function of rotational speed of the microturbine.

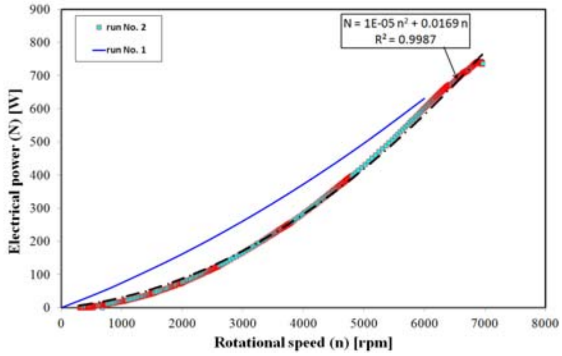


Fig. 11. Electrical power generated during the measurement series no. 1 and no. 2 vs. rotational speed of the microturbine

The maximal electric power, namely 750 W, was generated at the speed of 6950 rpm and was lower than the nominal speed. Due to the increase of clearances between stages of the microturbine, the machine exhibited a deterioration in operational performance, which manifested itself in such a way that similar levels of electrical power are achieved at higher rotational speeds than before. The clearances proved to be too large. The machine has much higher internal power losses and it is no longer possible to achieve nominal operating parameters (i.e. rotational speed and electric power output). In order to improve the turbine set's efficiency, a new kind of material should be selected for sealing purposes, which is the key to creating a better seal integrity. The material should have a relatively low coefficient of thermal and volumetric expansion. An alternative solution could be to apply a different type of seals. Returning to the measurement results, it may be noted that the operating temperature of the high-speed generator (around 120 °C) was much higher than the maximum value prescribed by the manufacturer and this had a considerable impact on a reduction in the generator's efficiency. This provides evidence that the passive cooling of the generator is not sufficiently efficient to keep the temperature at an acceptable level. It, therefore, seems necessary to intensify heat transfer, e.g. through the increase of the heat transfer area and/or the use of liquid/forced-convection cooling systems.

The exemplary measurement results relating to vibration velocity distributions, registered within the framework of the second measurement series (during a microturbine run-up), are presented in Fig. 12 - 14.

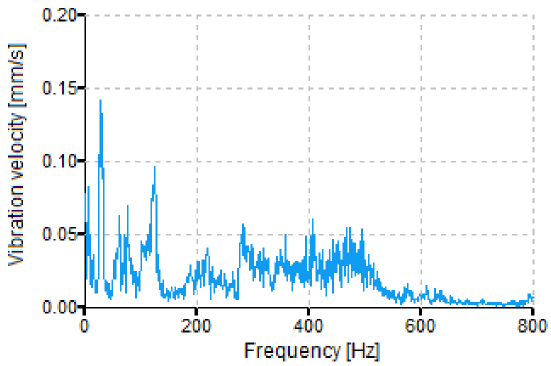


Fig. 12. Vibration velocity frequency spectrum measured on the microturbine casing during the run-up

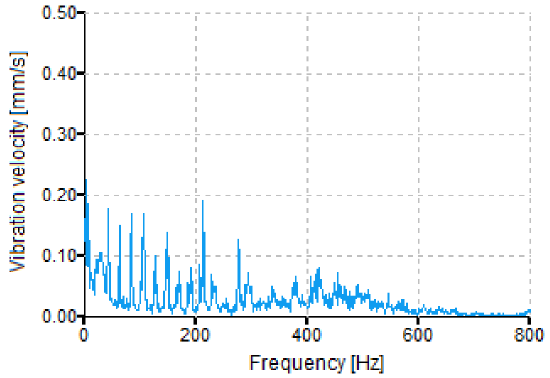


Fig. 13. Vibration velocity frequency spectrum measured on the microturbine casing during the run-up

On the basis of the vibration measurement results, one can state that the microturbine operation was accompanied by adverse phenomena of the mechanical nature. The measured vibration velocity spectra have many harmonic components; what is more, a high vibration level is present within wide frequency ranges. During vibration measurements carried out at constant rotational speeds (Fig. 14), several harmonic components were also observed. With respect to this, there is clear evidence that the rotating system's operation is far from reliable.

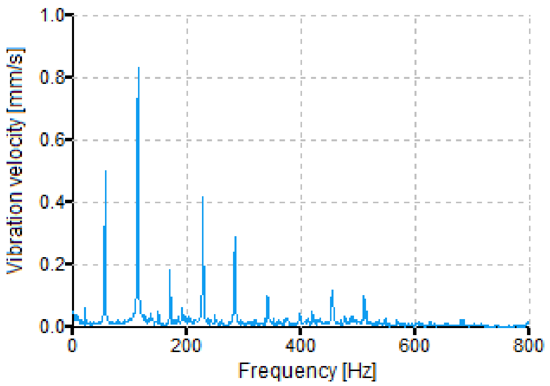


Fig. 14. Vibration velocity frequency spectrum measured on the microturbine casing at the speed of 3420 rpm

Summing up, vibration velocity spectra presented above differ substantially from spectra that are typical of properly functioning fluid-flow machines. In practice, vibration velocity spectrum of a well designed, thoroughly tested and properly maintained rotating machine, should have only one dominant synchronous component, the frequency of which corresponds to the rotational speed of machine's rotor. The occurrence of other components usually means a technical problem (e.g. a bent rotor, misalignment, failure of bearings or another defect). The presence of several harmonic components and an elevated vibration level within wide frequency ranges, which is usually accompanied by rubbing and clatter, is often a sign that a machine is nearing the end of its useful service life. Changes in vibration distribution that occur during the operation of a rotating machine are considered to be the indicator of phenomena that can cause changes in the machine itself; they can be observed during a run-in process or damage propagation.

Figure 15 presents the RMS vibration velocity values registered both during the measurement series no. 1 and no. 2 as the function of rotational speed of the microturbine.

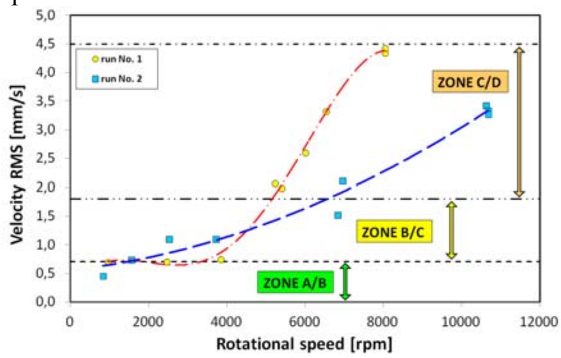


Fig. 15. RMS vibration velocity registered during the measurement series no. 1 and no. 2 vs. rotational speed of the microturbine

Let us note that the vibration velocity level increases along with increasing the rotational speed, and this is the case during both measurement series. During the second measurement series, the rotor's vibration level was within the zone A/B up to the rotational speed of 1500 rpm. Within the speed range of 1500 to 6500 rpm, the vibration velocity level was within the zone B/C that corresponds to the values from 0.71 mm/s to 1.8 mm/s. Above the speed of 6500 rpm, the vibration velocity values were within the zone C/D. It should be noted that the microturbine vibration level (up to the speed of 4500 rpm) was higher in the second measurement series than in the first one. This was due to excessive clearances in the flow system and only after the speed of 4500 rpm is exceeded, the situation is the opposite.

In the framework of the analysis of the rolling bearings, the so-called kurtosis was measured within the frequency range of 15 kHz to 30 kHz. The kurtosis, occurring in the form of a coefficient, gave us the information on high-frequency vibration signals originating from a bearing. As long as there are no stronger impulses that repeat themselves periodically within the signal, the value of the kurtosis is approximately 3. This value increases as a damage in the bearing occurs. According to the recommendations, a value of the kurtosis should be between 3 and 4 for new bearings and at least 10 or more for damaged bearings [36]. The value of the kurtosis—measured several times and at different speeds was within the range of 4 to 4.6. Since the obtained values are close to the maximum permissible value determined for new bearings, there can be no doubt that the bearings operate properly.

## 4. CONCLUSION

On the basis of the measurement results, it can be concluded that the axial-flow microturbine has not achieved its nominal operating parameters (i.e. rotational speed and electric power) in spite of providing the nominal parameters by the ORC system to the working medium—i.e. pressure, temperature and flow rate. It was observed that the RMS vibration velocity values (measured on the microturbine casing) increase along with increasing the rotational speed. During the conducted run-ups, the maximal electric power measured at the generator terminals was approximately 750 W and was achieved at the rotational speed of 6950 rpm. At the highest tested speed of the microturbine, namely 10680 rpm, the RMS vibration velocity was around 3.3 mm/s. According to ISO 10816-1 standard, this vibration level falls into the zone C/D, which means that the machine is no longer suitable for continuous operation and appropriate remedial and corrective actions should be put in place. The following improvements are found to be necessary to reach the microturbine nominal operating parameters: modernisation of the cooling system of the generator with a view of increasing its efficiency; selection of a new kind of material for sealing purposes or application of a different type of seals in the microturbine flow system.

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