Article citation info:
Żywica G., Kaniecki M., Technical diagnostics of hydropower turbines using modern measurement techniques, Diagnostyka. 2016;17(2):87-94

DIAGNOSTYKA, 2016, Vol. 17, No. 2

ISSN 1641-6414
e-ISSN 2449-5220

TECHNICAL DIAGNOSTICS OF HYDROPOWER TURBINE USING MODERN MEASUREMENT TECHNIQUES

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Summary

The article demonstrates how to take advantage of modern measurement techniques and detect defects of different types. These techniques contributed to solve the unknown operating problem of a hydropower machine. The examined hydro set comprises a water turbine (of the Francis type), elastic coupling, flywheel and electro-generator. To assess the dynamical state of the machine and detect the reasons for elevated vibration level, the following diagnostic tests were carried out: precision measurement of vibration, laser shaft alignment check and thermal imaging tests. These three widely diverse measurement techniques complemented each other, allowing the identification of the cause of increased vibrations and taking countermeasures to make further operation possible. The article provides an example of the practical application of the currently available vibrodiagnostic methods for monitoring and assessment of the technical condition of fluid-flow machines.

Keywords: technical diagnostics, vibration-based diagnostics, turbine set, water turbine

DIAGNOSTYKA TECHNICZNA TURBINY WODNEJ PRZY WYKORZYSTANIU NOWOCZESNYCH TECHNIK POMIAROWYCH

Streszczenie

W artykule przedstawiono możliwości nowych technik pomiarowych w zakresie wykrywania różnego typu defektów i problemów eksploatacyjnych na przykładzie rzeczywistej maszyny przepływowej. Badany hydrozespół składał się z turbiny wodnej typu Francisa, podatnego sprzęgła, koła zamachowego oraz generatora prądu. W celu oceny stanu dynamicznego maszyny oraz wykrycia przyczyn podwyższonego poziomu wibroaktywności, przeprowadzono precyzyjne pomiary drgań, laserową kontrolę współosiowości wałów oraz badania termowizyjne. Zastosowane metody pomiarowe doskonale się uzupełniały, pozwalając na wykrycie źródeł drgań oraz zastosowanie środków zaradczych umożliwiających dalszą eksploatację maszyny. Artykuł stanowi przykład praktycznego wykorzystania wszechstronnych możliwości nowoczesnych metod wibrodiagnostycznych do kontroli i oceny stanu technicznego maszyn przepływowych.

Słowa kluczowe: diagnostyka techniczna, wibrodiagnostyka, hydrozespół, turbina wodna

1. INTRODUCTION

The current European Union's energy policy, which aims at reducing greenhouse gas emissions, thus improving air quality (particularly in cities) promotes energy efficiency and renewable energy sources. One of the most popular and well-studied green energy technologies are hydroelectric plants that make use of local hydrological conditions. Well designed and built hydroelectric plant can function for many years whilst having as little negative impact on the environment as possible.

Hydroelectric plants are the facilities required to be highly efficient and reliable. In order to improve the economic indicators, attempts are being made to reduce investment and operating costs. In the case of hydro sets of low power capacity, the maintenance costs are very often limited to periodic inspections conducted at fairly lengthy intervals.

Currently available measurement techniques and computer aided engineering programs make it possible to carry out comprehensive experimental research and design the energy-optimized fluid-flow machines of different types. This is especially important for hydro sets which, having regard to possible changes of local hydrological conditions must provide correct operation under variable water flow rates [1]. That is why both theoretical and experimental research is being conducted in order to gain a better understanding of various fluid-flow [2-5] and electrical [6] phenomena or dynamic performance of the structural components [7]. Owing to considerable possibilities of contemporary engineering tools it has become possible to estimate the impact of different factors on the operation of a hydro set. Consideration is given to factors such as unbalance [8], supporting structure damage [9,10] and the presence of foreign bodies in a flow system

[11]. Modern diagnostic systems offer a very wide spectrum of possibilities which allow, among other things, to detect defects affecting the runner, bearings or generator, reducing operating costs [12]. The sophisticated diagnostics systems, because of its expense, are put in place only in the case of hydro sets with power capacities of more than a few megawatts [13,14]. The maintenance works for small water turbines are limited to periodic maintenance checks (checking only basic operational parameters) and further actions are undertaken in the event of operational problems [7,15].

This paper presents the research on the hydro set equipped with a 315 kW generator and Francis turbine, which were conducted in order to find the causes of elevated noise and vibration levels. The consequences of these phenomena were worrying, especially when one notices the continued deterioration of the dynamic performance of the hydro set. Apart from traditional vibration measurements, it was decided to use modern measurement techniques in order to get more information about the observed machine. The authors of this article expected that such an approach will allow to work out the most appropriate solution to the problem identified.

2. TEST OBJECT CHARACTERISTICS

The examined hydro set is the first of the twin turbomachines located on the hydroelectric plant site, supplied with water from the nearby lake. The hydro set consisted of two main components – the Francis turbine and electro-generator, which were connected by an elastic coupling (Fig. 1). Additionally, the flywheel was mounted between the coupling and generator.

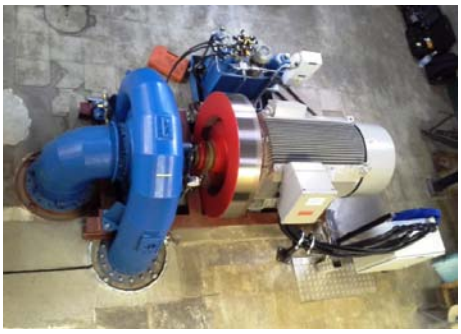

Fig. 1. Picture of the examined turbine set.

The described machine was modernized in 2013. During this time the main components were replaced, including the turbine and generator. The basic technical parameters of the hydro set are shown in Tab. 1. Both shafts (turbine's and generator's) were placed above the floor level, which facilitated the research work. The turbine shaft was supported by two rolling bearings placed in its central part. A bladed disk was at one end and a coupling sleeve at the other end of the turbine shaft. The generator shaft was equipped with its own bearing system with two rolling bearings. The flywheel was located on the generator shaft (on the side of the turbine), helping the stabilization of rotational speed and torque at transient operating conditions (runaway or emergency closing).

Tab. 1. Basic specification of the turbine set.

Water turbine	
Country of origin	Poland
Type of turbine	Francis
Number of rotor blades	11
Number of stator blades	20
Net head	67 m
Nominal flow rate	0.5 m^3/s
Rated speed	1010 rpm
Generator	
Country of origin	Germany
Type of generator	Asynchronous, 3-phase
Nominal power	315 kW
Rated speed	992 rpm
Maximum speed	2000 rpm
Mass	2000 kg

3. DIAGNOSTIC TESTS

After the performance of the modernization of the hydro set, despite a very precise balancing and alignment of the shafts, the dynamic state of the machine continued to deteriorate. This led to temporary suspension of exploitation of this hydro set. In order to find the causes of elevated noise and vibration levels the diagnostics test were carried out. The scope of these tests included vibration measurements, determination of the natural frequencies, shafts alignment check and analysis of the temperature distribution.

3.1. Dynamic state assessment

The dynamic state of the hydro set was analyzed on a basis of existing standards ISO10816-1:1995 and ISO10816-5:2000. The former standard sets out overall requirements for the conduction of measurements, and the latter contains specific recommendations for electro-hydraulic machines, including water turbines. Due to its design features, the hydro set falls in group 1, i.e. horizontal shaft machines, with a rigidly supported bearings operating at the rotational speed greater than 300 rpm. The main measured parameter was the root mean square value of the vibration velocity (V_{RMS}). The measurement points were positioned on the outer surface of the machine body as depicted in Figure 2. The measuring data were collected from three nodes in three directions, making a total of 9 parameters. The first measuring node was placed within the rear generator bearing, the second at the front generator bearing and the third in the vicinity of the bearings mounted on the turbine shaft. The vibration measurements were conducted for the directions corresponding to the axes of the coordinate system shown in Fig. 2.

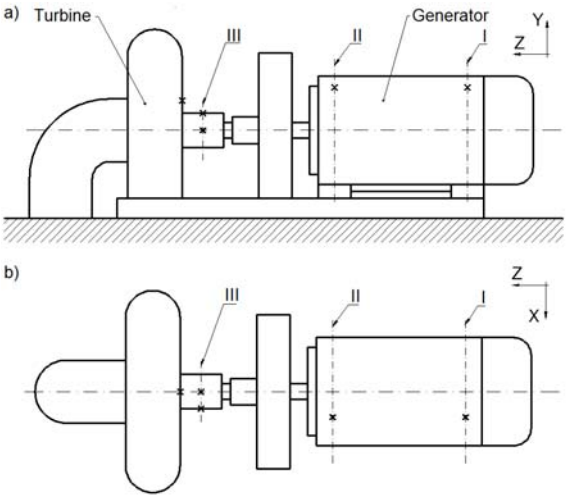

Fig. 2. Turbine set scheme with measuring points (a – side view, b – top view).

The mobile vibration analyzer (DIAMOND 401AXT) with the uniaxial accelerometer has been used as a measuring device. The measurements were carried out in the frequency range 2-1000 Hz with the averaging time equal to 3 seconds.

The dynamic performance test was conducted at almost full (95%) guide vane opening, when the water turbine was delivering a power equal to 310 kW. The measured V_{RMS} values are presented in Table 2. The values exceeding the acceptable level (i.e. 2.5 mm/s which enables long-term safe operation as stated in ISO10816-5:2000 standard) are placed on a light grey background. The vibration velocity level which may result in irreparable damage to the machine (above 4 mm/s) was shown on a dark grey background.

Tab. 2. The results of measurements at 310 kW.

Node number	Direction X	Direction Y	Direction Z
	V_{RMS} (mm/s)		
I	8.50	7.57	8.38
II	11.2	5.92	8.21
III	3.35	3.55	3.07

The results of this research proved that the machine operated in poor technical condition. The so-called "alarm vibration level" was exceeded in all measuring points placed in the vicinity of the generator. This means that the device should be switched off immediately.

In order to determine the cause of elevated vibration level, it was decided to perform the frequency analysis of the registered vibration signals. The aim was to identify characteristic frequencies and find the sources of mechanical vibrations [16]. The vibration velocity spectra were analyzed in the range from 0 to 800 Hz with resolution of 0.5 Hz, which corresponds to 1600 measuring lines. The calculated frequencies of the machine components are listed in Tab. 3.

Tab. 3. Characteristic frequencies of the turbine set.

Defect	**Frequency (Hz)**
Water turbine shaft	
Unbalance	16.8
Misalignment/bending	16.8; 33.7; 50.5
Turbine blade vibrations	185.1
Generator shaft	
Unbalance	16.8
Misalignment/bending	16.8; 33.7; 50.5
Electrical excitation	50, 100

The vibration spectra for two selected measuring points, obtained during dynamic performance test are presented in Fig. 3 and 4. The dominant frequency component of 33.5 Hz is present in both signals, which is close to the frequency value corresponding to misalignment or bending of the shafts. High vibration level was also registered at 50.5 Hz. Moreover, the elevated vibration level in the vertical direction was observed at 65 Hz. Furthermore, the frequency component of 100 Hz was clearly apparent, which corresponds to the electrical excitation frequency of the generator.

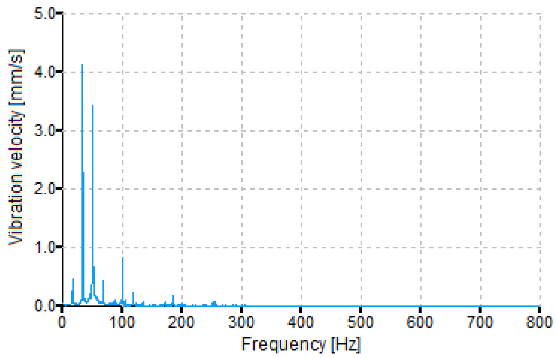

Fig. 3. Frequency spectrum of the vibration velocity at node II in the direction X.

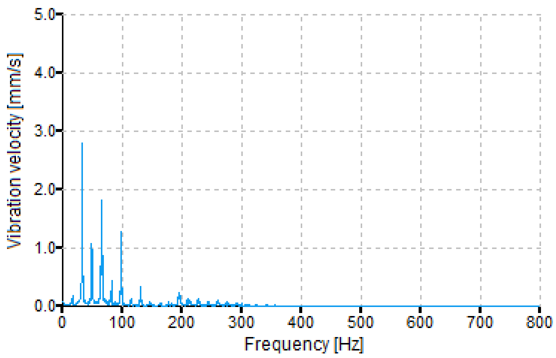

Fig. 4. Frequency spectrum of the vibration velocity at node II in the direction Y.

It is also worth noting that the synchronous frequency component (about 16,5 Hz) had very low amplitude value, and this was an indication of correct balance of the rotating system. On the basis of conducted signal analysis it can be concluded that the exceeded vibration level is caused by shaft misalignment or deflection.

In order to further examine the misalignment of shafts (turbine shaft and generator shaft), additional measurements were carried out, i.e. the vibration levels of machine housings on both sides of the coupling were determined. The two vibration signals measured in this way were in antiphase (the phase difference was 180°), thus confirming earlier assumptions. This conclusion was slightly surprising especially when taking into consideration the modernization during which the shafts had been very precisely aligned. A failure in the form of shaft deflection also seemed very unlikely because the shafts had large diameters and were not long.

3.2. Eigenfrequencies

Further tests (including modal analysis allowing for the identification of the system eigenfrequencies) were conducted because it was suspected that the misalignment of the shafts might not be the only cause of poor operational performance of the turbine set. The impact excitation method was used during which the vibration spectra of the machine were registered. At this stage of the examination the portable vibration analyzer (mentioned above) connected to an uniaxial accelerometer was used.

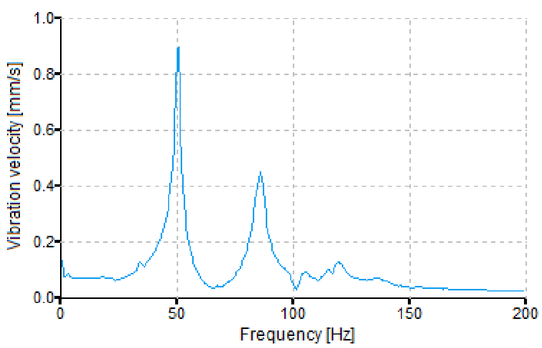

Fig. 5. Vibration spectrum after impact excitation at the node no. I (direction X).

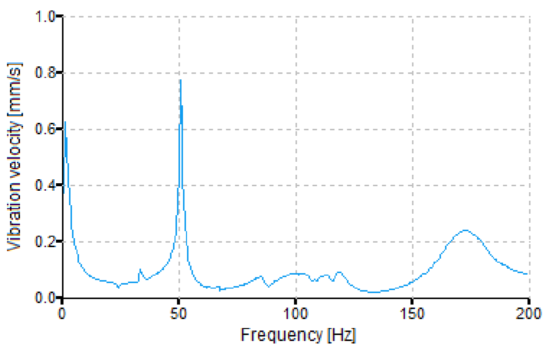

Fig. 6. Vibration spectrum after impact excitation at the node no. II (direction X).

The research was carried out in the frequency range from 0 to 200 Hz with resolution of 0.5 Hz. The Figures 5 and 6 demonstrate selected results of the modal analysis. The location of measuring points was already shown in Fig. 2.

On the basis of characteristics obtained it can be explicitly stated that the generator oscillated about its horizontal plane at the frequency of 50.5 Hz. In view of the fact that at the same frequency high vibration level was observed for two measuring nodes it has been found that these vibrations can be classified as mechanical oscillations of the entire generator. The resonance also occurred in the node I at 86 Hz, which was related to eigenvibrations of the rear part of generator (also in the horizontal plane). An attempt was also undertaken to determine other mode shapes of the hydro set but the obtained results did not allow for unequivocal identification of remaining eigenfrequencies.

3.3. Checking of shaft alignment

Precise machine shaft alignment minimizes vibration level, reduces the load on bearings and coupling, improves energy efficiency and increases the lifespan of components such as bearings, seals or couplings. Since the measurement results discussed earlier proved that the main cause of increased machine vibroactivity were vibrations taking place at the characteristic frequency related to the misalignment, it has been decided to pay more attention to this aspect.

Shaftalign OS3 high-tech laser-optical device by Pruftechnik was used for shaft alignment check. The device permits to measure parallel and angular displacements in horizontal & vertical plane with a precision of 0.01 mm. A laser shaft alignment system provides higher resolution, accuracy and speed of measurements than the traditional systems with dial indicators. In addition, the displacements of machine parts can be monitored in real-time during its correction. The picture taken during the alignment check of the hydro set shafts is presented in Fig. 7.

Fig. 7. Laser encoder with reflector used during shaft alignment.

The shaft alignment check revealed the exceedance of limit values in the vertical plane. The height difference between two coupling parts was only around 0.18 mm, meaning that a change in the heights of front and rear generator's terminals must be applied. These heights were adjusted by 0.25 mm and 0.38 mm, respectively (Fig. 8). Other values were within the recommended tolerances.

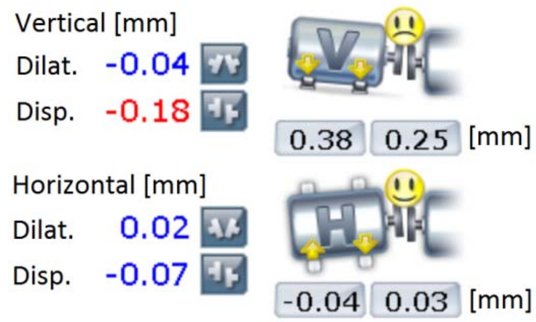

Fig. 8. Shaft alignment check results (screen from the measuring instrument).

To confirm the obtained results, the shaft alignment process was repeated several times for various angular positions of the shafts and places for fixing the measuring device. The subsequent results were very close to each other and to the results presented in Fig. 8. The generator was lowered down by removing the spacers of appropriate thickness from under its terminals.

3.4. Thermographic testing

Within the scope of diagnostics tests of the hydro set the temperature distribution near the coupling and on the outer surfaces of the generator's and turbine's body was examined. Flir E50 thermal imaging camera equipped with 45° wide-angle lens was used for this purpose. The measurement was carried out with the following camera settings: temperature range from -20 °C to +120 °C, emissivity coefficient 0.94 (appropriate for paint covering the machine bodies).

The selected results of the machine's thermographic analyses are shown in Fig. 9 - Fig. 11. In general, all temperature levels of hydro set looked good. The outer surface of the electro-generator's body reached up to 62 °C in some places and the turbine's bearing housing temperature reached a maximum of 65 °C. However, the biggest interest was aroused by the large temperature difference between the turbine and generator bodies (up to 55 °C). This can be seen in Fig. 11. In view of this information, it can be said that such a large temperature difference between bodies and bearing supports attached to the shafts should be addressed in the alignment process. Thermal expansion of steel parts, in that case, may cause the mutual displacement of the shafts in opposite directions, thus worsening operating conditions of the rotating system which leads to the occurrence of higher vibration level.

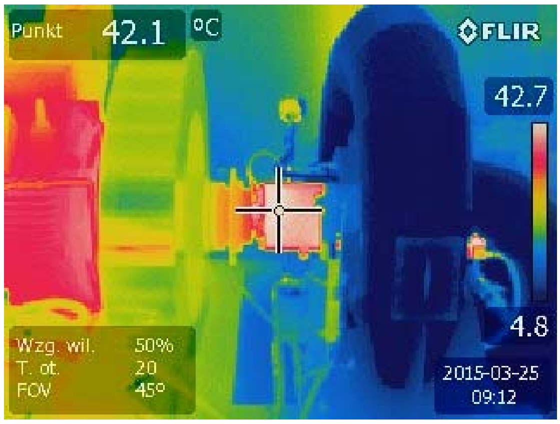

Fig. 9. Outside surface temperature of the turbine body at the point of bearing attachment.

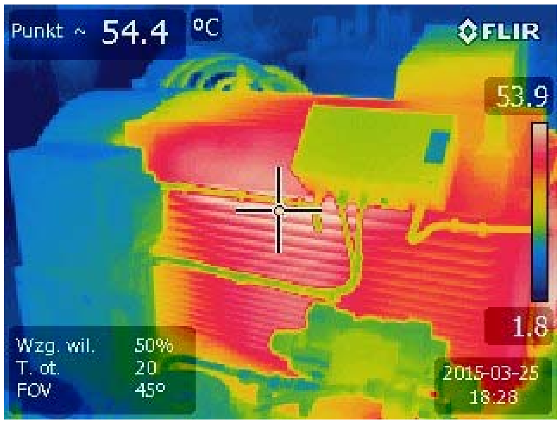

Fig. 10. The temperature distribution on the outer surface of the generator (from electrical connection side).

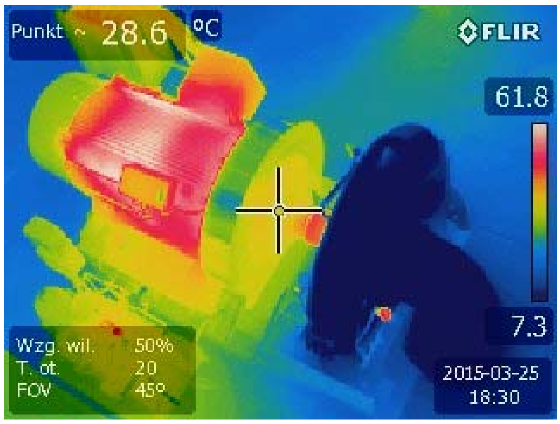

Fig. 11. Outside surface temperature of the hydro set (top view).

4. MODIFICATION OF THE MACHINE AND ITS CONTROL TESTS

The performed diagnostics tests showed that the characteristic frequencies of the turbine and generator shafts (3X) closely overlapped the eigenfrequency of the electro-generator. Both frequencies were around 50.5 Hz. Consequently, even small misalignment of the shafts caused the natural vibration of generator transferred on the entire machine.

For the above reasons, to reduce vibration level of the machine, subsequent work has therefore concentrated on making the difference between the natural frequency of the electro-generator and the characteristic frequency 3X. This objective has been achieved through increasing the stiffness of steel frame on which the generator was mounted, i.e. attaching additional elements commonly called stiffeners. Besides, in order to minimize the exciting forces causing vibrations of the system, the shaft alignment was carried out again, taking account of the displacements of shafts caused by thermal expansion of the bearing supports. To be able to assess the effect of these modifications, the appropriate control tests were conducted. The results of these tests are presented in the following sections of this chapter.

4.1. Checking of shaft alignment

The measuring results showed that in order to successfully perform shaft alignment of the hydro set the thermal expansion of bearing supports must not be ignored. This can be done in two ways: perform shaft alignment on the pre-heated machine, or take into account the estimated thermal expansion of the supports when performing shaft alignment on cold machine (by introducing "pre-misalignment"). It turns out that, according to the temperature difference, the horizontal position of the generator shaft may have changed more than 0.2 mm.

In the described case, the first way was used, i.e. the control measurements were carried out for the pre-heated machine. Regarding the initial position, the spacers with a total thickness of 0.4 mm were removed from under rear generator's terminals, and the spacer of the thickness of 0.2 mm from under front generator's terminals. The results of shaft alignment check obtained after the above changes are shown in Fig. 12.

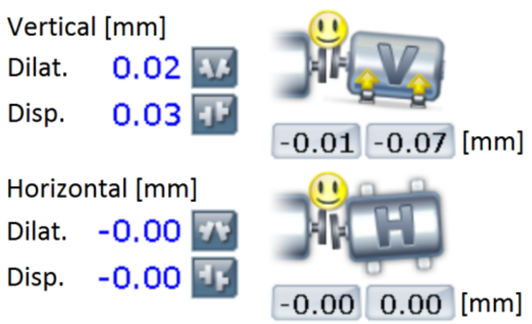

Fig. 12. The results of shaft alignment check after the adjustment of generator's position (screen of the measuring instrument).

The conducted measurements showed correct mutual positions of the shafts. The obtained deviation from ideal position in the vertical plane reached several dozen μm, which did not permit further correction of the position.

4.2. Eigenfrequencies

The additional elements increasing the stiffness of the generator's foundation were mounted on the steel frame to change the frequency of generator's eigenvibration. The aim was to increase the natural frequency enough to prevent overlapping it with the exciting frequency of the rotating system 3X, coming from misalignment. After mounting the frame reinforcements, the registered generator's vibration spectrum (Fig. 13 and Fig. 14) showed that the eigenfrequency increased to 55.5 Hz (was higher by 5 Hz than it had been before). This avoided the problem of operation under resonance conditions. Moreover, the vibration amplitude value of this mode shape was more than three times lower, at a similar level of exciting force.

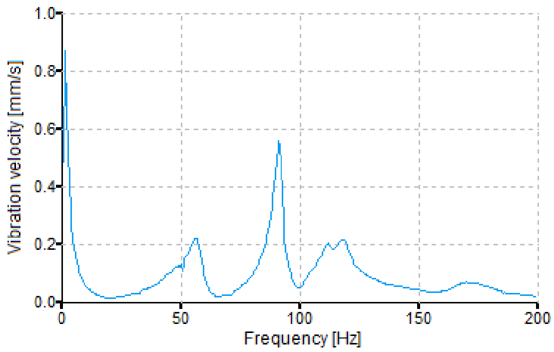

Fig. 13. Vibration spectrum after the impact excitation at node I (direction X).

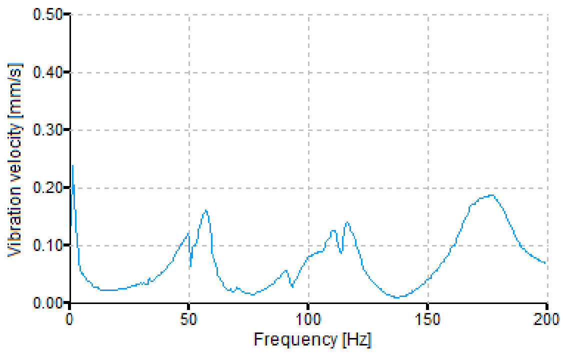

Fig. 14. Vibration spectrum after the impact excitation at node I (direction Y).

At the characteristic frequency of 50.5 Hz, the elevated vibration level continued to be observed, but it was no longer seen as a resonance.

4.3. Examination of vibration levels

The increased stiffness of the frame to which the electro-generator was attached and re-aligning the shafts, taking account of thermal expansion, led to significant reduction in noise and vibration levels. That was noticeable even without the use of any measuring devices. The measurements were re-performed in order to re-assess dynamic state of the hydro set. They were done under the following conditions: the machine was pre-heated (several hours of operation), the results were collected for three operating modes (guide vane opening at 95%, 70% and 50% which corresponded to power capacities of 310 kW, 230 kW and 125 kW, respectively). The measuring results are given in Tab. 4 - Tab. 6.

Tab. 4. The results of measurements at 310 kW.

Node number	Direction X	Direction Y	Direction Z
	V_{RMS} (mm/s)		
I	1.52	1.09	1.37
II	1.53	0.79	1.52
III	1.18	0.68	0.92

Tab. 5. The results of measurements at 230 kW.

Node number	Direction X	Direction Y	Direction Z
	V_{RMS} (mm/s)		
I	1.49	1.20	1.53
II	1.84	0.72	1.58
III	0.73	1.05	0.86

Tab. 6. The results of measurements at 125 kW.

Node number	Direction X	Direction Y	Direction Z
	V_{RMS} (mm/s)		
I	1.48	1.22	1.56
II	2.07	0.83	1.71
III	0.77	1.20	0.87

According to the evaluation criteria for electro-hydraulic machines of the group 1, the tested hydro set fulfils the requirements of long-term operation without any restrictions. The vibration level prescribed for machines that are newly commissioned (1.6 mm/s) has been exceeded only when operating with partial loads. Furthermore, it occurred for one measuring point at the power of 230 kW and for two measuring points at the power of 125 kW. Thus, it can be concluded that the introduced modifications decreased vibration level of the hydro set to the level that is expected in new hydraulic machines.

In line with a decrease in vibration level, the frequency spectra of measured signals changed (Fig. 15 and Fig. 16). Apart from a large decrease in the amplitude values of individual vibration components (compared to the results obtained during initial check, see Fig. 3 and Fig. 4), the reduction of the prevailing importance of characteristic components occurring during misalignment of the shafts could be observed. The characteristics below contain several vibration components of comparable amplitude values, which means that the overall vibration level results from the components at different frequencies, and therefore arises from different sources.

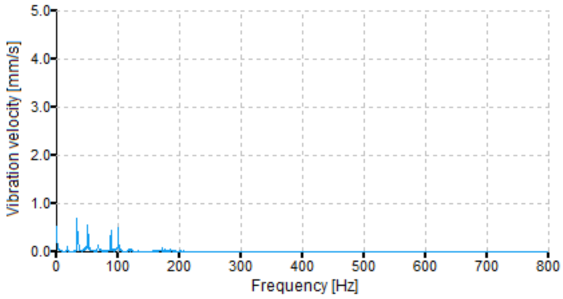

Fig. 15. Frequency spectrum of the vibration velocity at node II in the direction X

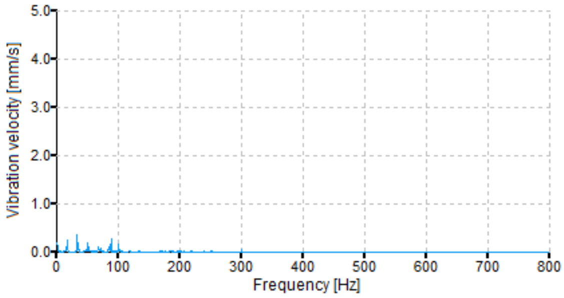

Fig. 16. Frequency spectrum of the vibration velocity at node II in the direction Y.

The correct operation of the rolling bearings was checked several times in the framework of the control tests. For this purpose the shock pulse measurement was used. The envelope spectrum of acceleration and the coefficient of kurtosis were also analyzed. The machine was operating with acceptable vibration level and there were no symptoms usually associated with bearing failure or bearing fatigue.

5. SUMMARY AND CONCLUSIONS

The article discusses the experimental research carried out on a 315 kW hydro set in which operating problems occurred during its exploitation. In the course of diagnostic tests modern measuring techniques were used such as: precise vibration measurements accompanied by frequency analyses and identification of system eigenfrequencies, laser shaft alignment and thermography. The utilization of these mutually complementary techniques made it possible to clearly indicate the reason of incorrect machine operation.

The experimental research conducted on the hydro set and the results obtained make it possible to draw the following conclusions:

- The initial assessment of the machine showed that the measured vibrations exceeded acceptable level at all measuring points. The highest vibration level was observed for the generator.
- Performing frequency analysis of vibration data and examining amplitude and phase components

indicated that the problem is caused by shaft misalignment. The identification of natural frequencies, however, demonstrated that the characteristic runner frequency equal to 50.5 Hz coexisted with the generator's eigenfrequency (lateral vibration).

- The obtained results showed that the hydro set requires very precise shaft alignment taking into account thermal expansion of the bearing supports. Within the framework of machine modernization, the stiffness of generator base has been increased so as to alter natural vibration frequencies.
- Dynamical state assessment performed after aligning the shafts and bracing of the generator base proved that the machine operation was stable and measured vibration level permits its unrestricted long-term operation.
- Due to relatively high sensitivity of the machine to shaft misalignment which could entail the occurrence of resonance phenomenon, more frequent checks on dynamical performance are strongly recommended.

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Received 2016-03-20
Accepted 2016-05-25
Available online: 2016-06-04

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