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TESTS OF ROTARY MACHINES VIBRATIONS IN STEADY AND UNSTEADY STATES ON THE BASIS OF LARGE DIAMETER CENTRIFUGAL FANS

BADANIA DRGAŃ MASZYN WIRNIKOWYCH W STANACH USTALONYCH ORAZ NIEUSTALONYCH NA PRZYKŁADZIE WENTYLATORÓW PROMIENIOWYCH DUŻYCH ŚREDNIC*

The existence of unsteady states of rotary machines is a common problem. It occurs mainly during the machine start-up and is caused by "passing through" the critical velocity, which activates large scale object vibrations. Moreover, rotary machines vibrations problems are caused by faults such as unbalance, misalignment, bearing defects and others. The paper presents the results of tests of sample centrifugal fans vibrations, both in steady and unsteady states. The recorded time traces of non-stationary vibrations were analyzed with the STFT spectrum. This method allowed to identify main parameters influencing the level of vibrations during the start-up and regular operation of the machine. The tests were performed on four machines, which enabled an additional comparison of operational parameters of the whole flow system.

***Keywords**: rotary machines, unsteady states, vibrations.*

Problem występowania stanów nieustalonych maszyn obrotowych jest powszechnie spotykany. Pojawia się on głównie podczas rozruchu i wywołany jest „przechodzeniem" przez prędkość krytyczną, która wzbudza drgania obiektu w bardzo szerokim zakresie. Ponadto problemy z drganiami maszyn obrotowych wywoływane są przez takie czynniki jak niewyważenie, niewyosiowanie, defekty łożysk i wiele innych. W pracy przedstawiono wyniki badań drgań zarówno w stanach ustalonych jak i nieustalonych przykładowych wentylatorów promieniowych. Za pomocą widm STFT przeanalizowano zarejestrowane przebiegi drgań niestacjonarnych. Dzięki temu możliwe było zidentyfikowanie głównych parametrów mających wpływ na poziom drgań podczas rozruchu oraz pracy normalnej. Badania przeprowadzono na czterech obiektach, co umożliwiło dodatkowe porównanie i wyciągnięcie wniosków odnośnie parametrów eksploatacyjnych całego układu przepływowego.

***Słowa kluczowe**: maszyny obrotowe, stany nieustalone, drgania.*

1. Introduction

Vibrations in steady and unsteady states [18] occur in almost all cases of rotary machines operation. In the first case, vibrations occur during the nominal operation of the machine and, generally, if the technical condition of the machine is good, do not constitute a significant operational problem. The other type of vibrations occurs during the start-up and braking of a supercritical machine. If the rotary machine is set in motion properly and under control, vibrations occurring during the operation are not a significant risk. It is important to smoothly and quickly "pass through" critical velocities of the rotary machine. The critical velocity matching resonance frequency might become a cause of damage of the machine, if the start-up is not performed properly. Rotation of the elements with the critical velocity provides high energy of excitation in wide spectrum. This causes resonance vibrations of the system. It has to be noted that the critical velocity of the rotor, often matching the frequency of its natural vibrations, is not the only resonance area in which the fan may operate. In the case of more complex machines, the frequency of natural vibrations of particular elements, e.g. a casing has to be taken into account. Additionally, in the case of fan rotors, it has to be remembered that the excitation frequency equals the rotary frequency of the shaft but

also the blade passing frequency, which is the product of the number of blades and a rotary velocity of the shaft [16]. Moreover, numerous flow effects cause dangerous vibrations of fan elements in other frequency ranges [7, 8, 25].

The level of excitation energy is also influenced by significant assembly and operating factors such as static and dynamic balancing of rotary elements, alignment of connections, bearings condition, electromagnetic interference and others [3, 6, 8, 9].

Proper operation of fans, particularly those of huge power and large diameters, not only decreases operating costs but also increases the safety of work. Their proper operation ensures constant air exchange and suction of dangerous gases in the mine pits, which along with safety systems [14, 15] protects pit mine workers. In the paper the results of vibrations tests of centrifugal fans responsible for ventilation of the mine are presented. Flow regulation of this type of fans is based on the fan airflow-pressure curve, where areas of unstable operation are located, which is unfavorable for the object. Tests performed during the machine operation allow to obtain key information for proper operation of the machine. Such tests are significant due to general lack of information regarding the subject of this type of machine behavior in the industry and because of the fact that the previous research was performed at a smaller scale in the laboratory

[7, 25]. The analysis and interpretation of the results were granted as well. The goal of the tests was to verify correctness of the flow system operation parameters and analyze possibility to increase the system functionality and efficiency, which was related to the fact that the systems were designed in 1970's.

2. The analysis of vibrations in diagnostics of the rotary machines

Machine vibrations are an important diagnostic symptom. It concerns a large number of machines operating in different conditions and industrial, civil and military applications [4, 12]. The analysis of vibration signals allows to verify the condition of the machine and is performed in order to verify damages of elements such as toothed wheels, bearings, rotary machines parts [1, 11, 17, 22]. However, verification of turbomachines vibrations may be frequently more difficult than single machine elements, due to a larger number of factors influencing the recorded vibration signal. These can include phenomena resulting from the flow of the medium and related pressure pulsation. The problem of an influence of the flow phenomena on the flow machine vibrations and noise emitted was discussed in numerous publications, e.g. [2, 7, 13, 22, 23, 25].

Among numerous methods of vibrations analysis for diagnostic purposes [5, 10, 17, 21, 22] those based on the Fourier's analysis [10, 21, 24] are widely applied. This method allows transition from the time to frequency domain and is based on the concept according to which each signal may be presented as the sum of different amplitudes and frequencies sinusoidal signals [21, 24]. Because of the commonly applied digital record of the signal, in practice the Fourier's analysis results in application of the Discrete Fourier Transform (DFT) [21]:

$$G(k) = \frac{1}{N} \sum_{n=0}^{N-1} g(n) \exp(-j2\pi kn/N) \cdot \qquad (1)$$

The algorithm enabling quick calculation of DFT is so called Fast Fourier Transform (FFT) [21]. The trace of vibrations amplitude in the frequency function achieved through this method allows to specify characteristic frequencies occurring during the machine operation.

In the case of rotary machines in which the states of unsteady operation occur, characteristic frequencies change in time, especially during the operation in the aforementioned states. In this case application of the Fourier Transform, in which integration takes place during the whole period of time, is not sufficient to recognize frequency changes over time. Due to that fact, in the vibrations analysis, it is reasonable to apply Short Time Fourier Transform (STFT), which is based on the Fourier's analysis performed within the narrow time frame displaced according to the agreed parameters of length, resolution or overlapping of frames. This method allows to observe relation of frequency and time and recognize transition states of the machine operation. The relation defining the STFT method is presented as follows [21]:

$$S(f, \tau) = \int_{-\infty}^{\infty} x(t)\varpi(t-\tau) \exp(-j2\pi ft)dt , \qquad (2)$$

where $\varpi(t)$ corresponds to the time frame displaced during data acquisition.
Due to numerous advantages in diagnostics of the rotary machines the analysis of data presented in the paper was performed in accordance with the STFT method.

3. Tested objects

The main ventilation system of a mine, which is the subject of the test, uses centrifugal fans denoted in Poland as WPK - 5,35, which in connection with the fan stations enable the mine ventilation. In the figure 1 the WPK-5.35 fan is presented.
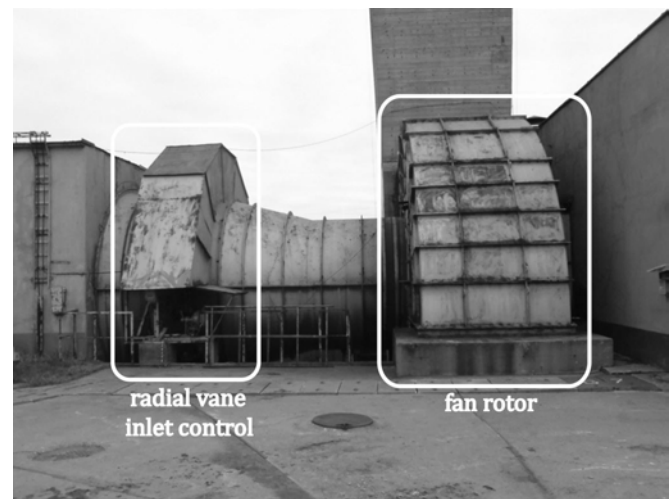

Fig. 1. Centrifugal fan WPK – 5.35

Vibrations of casings, channels, inlet vanes and air shutters noticed during fans operation negatively influence the working conditions. In order to precisely specify the level and cause of vibrations the tests were performed during the start-up of each fan at the station. Additionally, the tests were performed during the opening of the inlet vanes (figure 2), in order to determine the influence of different flow effects occurring during that process on the level of fan vibrations.
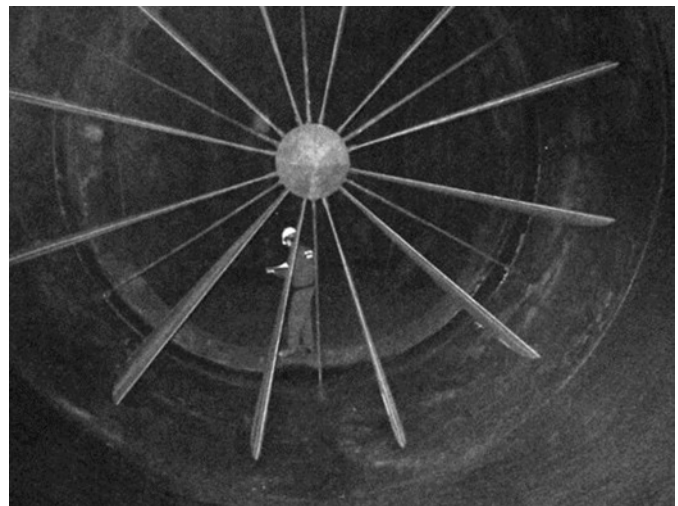

Fig. 2. Radial vane inlet control of the fan

4. Experimental tests

The goal of the tests was to record the vibrations velocity level in the ventilation systems during unsteady (start-up, opening of the inlet vanes) and steady states (regular operation). The tests were performed on the casings and intake channels of the fans and on the bearings housing of the rotors shafts. The identical arrangements of sensors, which is presented in the figures 3 and 4, were applied to all fans. Four WPK-5.35 ventilators, which cooperate within two different types of fan stations, were selected.
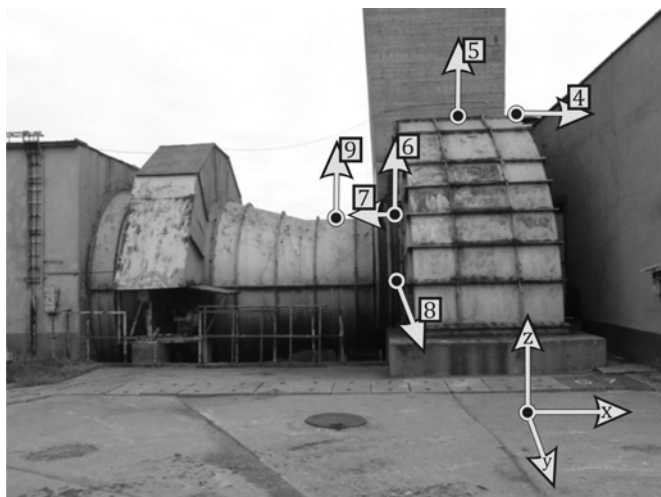
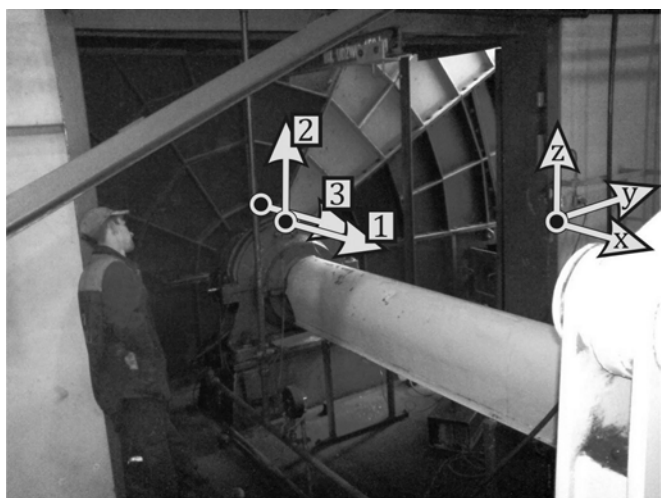*Fig. 3. Arrangement of measuring sensors on the fan casing*

Fig. 4. Arrangement of measuring sensors on the fan casing

Sensors in the measuring channel 1 and 2 were placed on the housing of the shaft bearing located closer to the rotor. The sensors on the other seven channels were placed directly on the rotors casings and the intakes channels. A detailed description of directions and arrangement of sensors on particular channels is presented in the table 1. Setting directions of the sensors are in compliance with the coordinate system presented in the figures 3 and 4.

Table 1.   Arrangement of measuring points

Channel no.	Direction	Location
1.	X	shaft bearing housing
2.	Z	shaft bearing housing
3.	X	fan casing
4.	X	fan casing
5.	Z	fan casing
6.	Z	fan casing
7.	X	fan casing
8.	Y	fan casing
9.	Z	fan casing

In the figure 5 the sensor on the channels no. 2 and 3 is presented. In the figure 6 the whole measuring system during the acquisition process is presented.
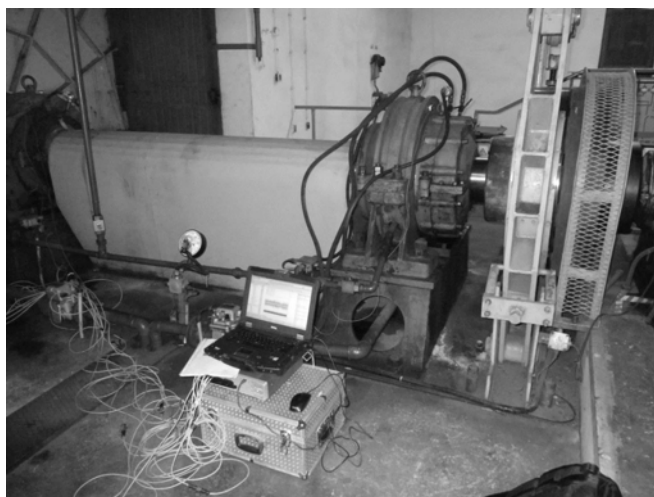

Fig. 5. Sensors on channels 2 and 3

Fig. 6. Sensors installation and data acquisition

5. Tests results

Figures 7–10 present selected STFT diagrams recorded on the selected fan. Characteristic changes of frequency in particular phases of the fan operation are presented in the figure 7.

The change of the blade passing frequency, which increases during the start-up, is easily noticed. Less clear, but also noticeable, are vibrations coming from the start-up system. Their frequency decreases to 50 Hz when the fan operates with its nominal revolutions. Particular attention should be paid to the stage in which the fan achieves the nominal revolutions (375 rpm [19]) but the inlet vanes have not yet been opened. It is clearly recognized that this is an unfavorable moment for the system as it causes strong activation of vibrations in the whole range. After opening of the inlet vanes, the stable operation of the fan and related characteristic frequencies of vibrations are recognized. In all figures presenting spectrum vibrations recorded by sensors located on the casing (figure 7-9) the aforementioned blade frequency of the rotor amounting to about 50Hz is clearly noticed. Its harmonic 100 Hz and 150 Hz are observed as well. A detailed interpretation of vibrations visible on the STFT diagrams will be possible after performing numerical, experimental or operational modal analysis. As a result, mode shapes and frequencies of natural vibra-
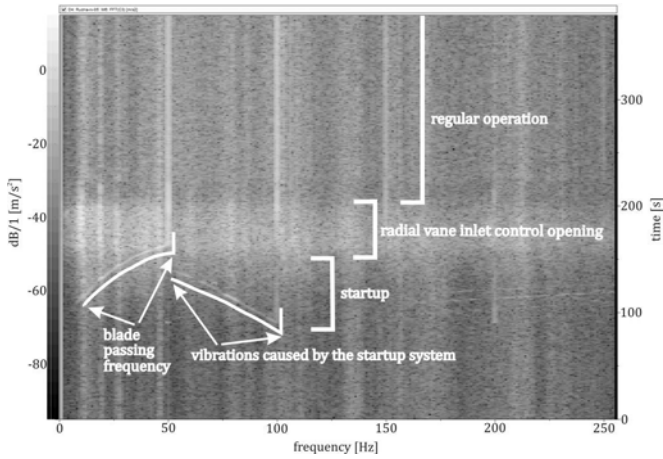
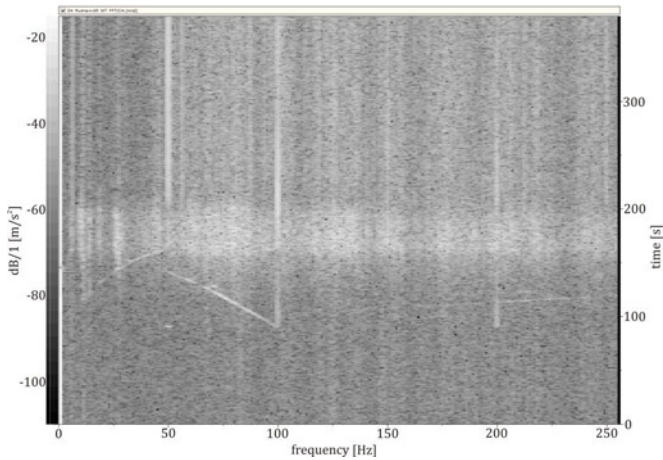*Fig. 7. W3 fan start-up – channel 3 (direction x, fan casing)*

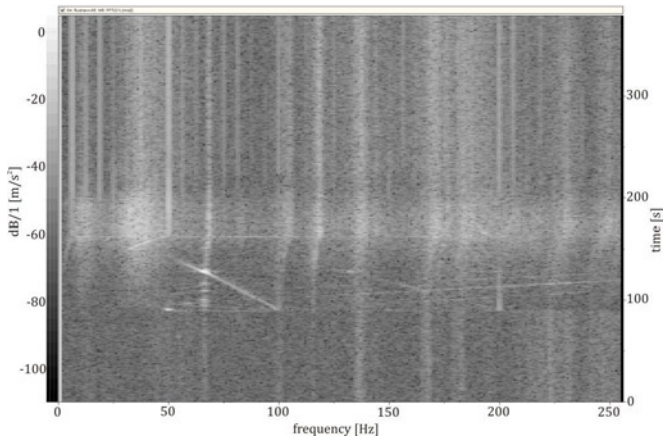*Fig. 8. W3 fan start-up – channel 4 (direction x, fan casing)*

Fig. 9. W3 fan start-up – channel 5 (direction x, fan casing)

tions will be specified and it will be possible to identify them on the presented diagrams.

The results obtained from the sensors recording bearing vibrations (in the figure 10 the spectrum of vibrations into the x direction is presented), tightly correlate with the fan casing vibrations, yet the level of acceleration is significantly lower. However, a significant difference in spectrums during the start-up and operation of the fan is noticed. During the start-up, the sensors located on the bearing do not record changes of blade velocity (vibrations are induced by the flow and influence only the fan casing). It is observable only after reaching the nominal velocity, when the flow pulsations are strong enough to influence the whole system. Also, a frequency not recorded on the fan
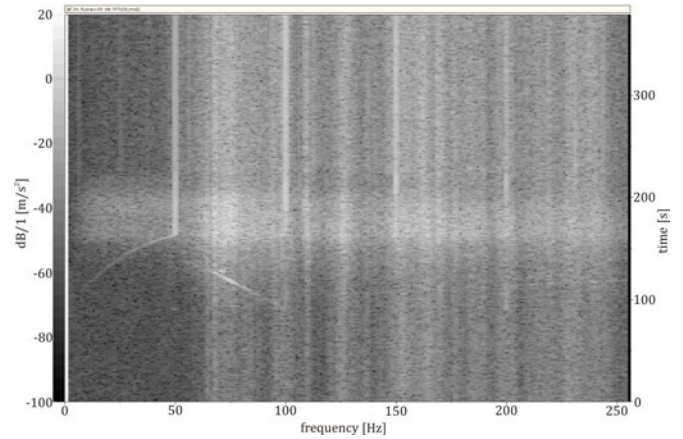

Fig. 10. W3 fan start-up – channel 1 (direction x, bearing housing)

casing amounting to about 6.25 Hz occurs. This corresponds with the rotary frequency and vibrations are induced by an unbalance of the shaft and rotor.

In the case of other fans, vibrations spectrums are similar. What distinguishes particular fans is a difference between duration of the start-up and opening of the inlet vanes. In the first case the start-up of the fan proceeded efficiently and it is difficult to notice on the STFT spectrum changing blade and start-up frequencies, as they after a while decline in the vibrations of the whole system caused by still unopened inlet vanes. The list of the vibrations levels is presented in the table 2.

In the table the levels of maximum vibrations and root mean square (RMS) of vibrations for an unsteady state (start-up and opening of the inlet vanes) and for a steady state (operation with a nominal rotary velocity and open inlet vanes) are presented. In order to verify in which case (on which fan) the situation is the least favorable, the vibrations levels in particular channels are compared. The color font of the values indicates the maximum value of vibrations velocity or RMS in a particular channel (in the particular measuring point) recorded in an unsteady state among all tested fans. The values in the color boxes indicate respectively the maximum value of vibrations velocity or RMS in the particular channel (in the particular measuring point) recorded in a steady state among all tested fans.

6. Summary and conclusions

Tests presented in the paper were performed in the real operation conditions. These allowed to specify characteristic frequencies and levels of forced vibrations of large sizes, power and efficiency centrifugal fan elements.

The vibrations analysis performed with the use of the Short Time Fourier Transform presented changes in the level and frequency of vibrations which correspond with particular phases of fans start-up sequences. As a result strong transient states in the process of operation of tested objects were identified. By verification of the received vibration spectrum characteristics, primary excitation frequencies were distinguished: rotary frequency of the rotor, blade passing frequency and their harmonic frequencies. Excitation in the wide frequency band noticed during opening of the inlet vanes results from the operation of the fan the region of instability (located on the left side from peak pressure point on fan airflow-pressure curve).

Apart from identifying of operation conditions of the fans for its different states it was also possible to compare characteristic parameters and make conclusions on the technical conditions of the objects tested and potential correction operations.

On the basis of tests performed, a huge discrepancy of vibrations level on particular fans was noticed. The highest level both in the steady and unsteady state is related to the W1 fan. It has the most

Table 2. List of parameters measured in particular rotors for all measured channel

Fan	Channel direction	1 x	2 z	3 x	4 x	5 z	6 z	7 x	8 y	9 z
W3	Unsteady state RMS [m/s²]	0.208	0.114	1.440	0.100	1.590	**1.321**	0.632	0.341	0.622
	Unsteady state MAX[m/s²]	2.826	9.279	**13.361**	0.915	13.015	9.259	5.015	1.535	9.744
	Steady state RMS [m/s²]	0.131	0.171	0.987	0.093	1.479	2.138	0.630	0.490	0.670
	Steady state MAX[m/s²]	0.527	9.288	3.787	0.396	9.968	10.837	2.463	1.763	10.287
W4	Unsteady state RMS [m/s²]	0.155	0.120	1.323	0.192	**2.411**	0.787	1.024	0.329	**0.707**
	Unsteady state MAX [m/s²]	1.488	9.252	12.349	**1.816**	**17.564**	9.287	**7.592**	**4.190**	9.497
	Steady state RMS [m/s²]	0.099	0.152	0.848	0.175	2.181	0.789	0.894	0.356	0.768
	Steady state MAX [m/s²]	0.406	9.319	3.692	0.874	10.539	9.272	3.290	1.276	10.144
W1	Unsteady state RMS [m/s²]	**0.290**	**0.152**	1.218	**0.257**	1.801	1.298	**1.394**	**0.775**	**0.777**
	Unsteady state MAX [m/s²]	3.149	9.308	7.919	1.327	13.261	9.297	**7.410**	2.518	9.547
	Steady state RMS [m/s²]	0.184	0.196	0.752	0.228	0.989	1.505	1.056	0.886	0.774
	Steady state MAX [m/s²]	0.688	9.299	2.627	0.699	10.313	11.695	3.518	2.400	9.958
W2	Unsteady state RMS [m/s²]	0.210	0.132	**1.665**	0.141	1.910	0.346	0.829	0.296	0.441
	Unsteady state MAX [m/s²]	**4.008**	9.281	**13.425**	1.107	16.413	9.422	5.851	1.710	9.572
	Steady state RMS [m/s²]	0.141	0.281	0.566	0.078	0.954	0.401	0.505	0.155	0.473
	Steady state MAX [m/s²]	0.721	9.386	2.111	0.538	10.760	9.259	1.890	0.857	9.737

maximum values of the RMS. This indicates a general high level of vibrations.

Similar situation occurs in the case of the W4 fan at the station. However, a significant difference is the fact that the largest number of extreme values is recognized for the maximum acceleration levels. This indicates that instantaneous accelerations of a high value occur.

Vibrations levels of the other two fans are significantly lower compared to the abovementioned.

The observed results confirm occurrence of diverse and inappropriate (too long) duration of fans start-ups. The occurrence of vibrations is also caused by the following factors: unnecessary keeping fans in motion while the radial vane inlet control is unopened, inappropriate unbalance of the rotary systems or inappropriate regulation of inlet vanes.

Also, a discrepancy between the vibration levels of particular fan casings is noticed. It may be however specified that in the channel 3 and 7 the vibrations levels the mostly deviating from the average in a given channel are observed. This proves that huge excitation of the fan casing into the axis direction in the middle area of the casing occurs. It is attributable to the fact that these are the most vulnerable areas of the casing, which vibrate with a large amplitude. In all cases it was possible to identify characteristic vibration frequencies correlated with the enforcement and the other potentially being the natural vibrations activated by operation of the rotor.

Such characteristic behaviors of fan rotor casings may prove the phenomenon of beat effect. The analysis of particular processes of acceleration allowed to find in some channels characteristic amplitude fluctuation, which confirms accuracy of this thesis. Therefore, there is probability of casings operation in the state comparable to resonance vibrations. This may cause the increase of the vibrations amplitude and generate casing damages. This fact may be finally confirmed after specifying the frequency and mode of the natural vibrations of fan casings either during the operation or with numerical or experimental methods [24]. After confirmation of this thesis it will be possible to implement construction changes in order to relocate the natural frequency of casings into the higher range.

Tests presented in the paper allow to identify operation conditions of complex mechanical-flow systems such as turbomachines. Operation of this type of machines is connected with numerous problems, which mainly result in different type of vibrations of particular elements or the whole group. The proper interpretation of reasons for their occurrence allows to make quick modifications in order to increase durability, decrease possibility of failure or allow further safe operations. In the case of huge power turbomachines, which were subject of the tests, the scale of these objects draws particular attention to the observed irregularities, whose symptoms are the increased levels of vibrations. In the case of this type of objects it is worth to consider installation of systems of periodical or constant monitoring of vibrations, recognized not only in the bearing support of the power transmission system, as it is right now, but also in the areas whose vibrations allow the early identification of potential problems. These areas are e.g. inlet vanes, fan casings etc. Additionally, it is accurate to monitor the flow parameters such as fan pressure ratio, flow machine efficiency and correlate them with the vibration signals. Such an attitude allows to fully identify phenomena occurring in different states of the machine operation and interpret them correctly. The next step should be to verify the influence of tested phenomena on the condition of the object. Numerical methods (FEM, BEM, FDM), which allow to perform simulations reflecting real (measured) operating conditions and predict its influence on the technical condition of the object, may be here applied.

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