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ANALYSIS OF THE BMA K2400 VERTICAL CENTRIFUGE TURBINE IN TERMS OF BALANCING AND VIBRATION DIAGNOSTICS

Physical damage to a material is a diffuse defect in the form of vacancies, microcracks, micro-voids or damaged micro-volumes, which reduce the effective or load-bearing part of the material. Surface fatigue defects, such as deformation and cracks, occur in the bearing during the load transfer. Imbalance is a practical problem in the operation of many rotating machines, causing not only increased vibration of the machine, but also leading to accelerated wear of the rotor bearings. The subject of this work is the analysis of the dynamics of the BMA K2400 centrifuge in terms of the possibility of correcting the balance in the given dynamic state. The paper describes the individual stages of solving the problem of excessive machine vibrations, assuming that its bearings were replaced before the diagnostic test. As a result of the lack of effects after replacing the motor bearings and after analyzing the vibration measurement results presented in article, a decision was made to inspect the centrifuge bearings. The diagnostics was performed again, but it concerned only the bearing node No. 1 with the disassembled basket. The measurements were performed using the DIAMOND 401 AX device, equipped with Wilcoxon 780B acceleration sensors with a sensitivity of 100mV/g. The appearance of a technological defect on the outer ring of the bearing, which is a friction pair with a housing, is not a typical damage for this type of machines and was an interesting problem. The consequence of the occurrence of bearing defects may be an increase in statistical values of the vibration signal and the appearance of new amplitudes in the FFT spectra. A vicious circle is created here, where bearings in poor dynamic condition increase the transmission of vibrations through the machine, and high vibrations accelerate the degradation of the bearings. The poor condition of rolling bearings may also prevent dynamic balancing of the rotor, and thus - lead to further propagation of bearing damage caused by an increased level of the machine's own vibrations.

***Key words**: Technical Diagnostics, Vibration Diagnostics, Rotor Unbalance, Bearing, Tribology, Exploitation.*

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АНАЛІЗ ВЕРТИКАЛЬНО -ЦЕНТРИФУГОВОЇ ТУРБИНИ ВМА К2400 В УМОВАХ БАЛАНСУВАННЯ ТА ВІБРАЦІЙНОЇ ДІАГНОСТИКИ

Фізичні пошкодження матеріалу-це дифузний дефект у вигляді вакансій, мікротріщин, мікропустот або пошкоджених мікрооб'ємов, які зменшують ефективну або несучу частину матеріалу. Під час перенесення навантаження в підшипнику виникають дефекти поверхневої втоми, такі як деформація та тріщини. Дисбаланс є практичною проблемою в роботі багатьох обертових машин, що викликає не тільки підвищену вібрацію машини, але й призводить до прискореного зносу підшипників ротора. Предметом даної роботи є аналіз динаміки центрифуги ВМА К2400 з точки зору можливості виправлення балансу в даному динамічному стані. У статті описано окремі етапи вирішення проблеми надмірної вібрації машини, припускаючи, що її підшипники були замінені перед діагностичним тестом. Внаслідок відсутності ефектів після заміни підшипників двигуна та після аналізу результатів вимірювань вібрації, представлених у статті, було прийнято рішення перевірити підшипники центрифуги. Діагностика була проведена знову, але це стосувалося лише несучого вузла No1 з розібраним кошиком. Вимірювання проводили за допомогою пристрою DIAMOND 401 AX, оснащеного датчиками прискорення Wilcoxon 780B. Поява технологічного дефекту на зовнішньому кільці підшипника, що являє собою пару тертя з корпусом, не є типовим пошкодженням для машин такого типу і являє собою цікаву проблему. Наслідком виникнення дефектів підшипників може бути збільшення статистичних значень сигналу вібрації та поява нових амплітуд у спектрах БПФ. Тут створюється замкнуте коло, де підшипники в поганому динамічному стані збільшують передачу вібрацій через машину, а високі коливання прискорюють деградацію підшипників. Поганий стан підшипників кочення також може перешкоджати динамічній збалансованості ротора, а отже - призвести до подальшого поширення пошкоджень підшипників, викликаних підвищеним рівнем власних вібрацій машини.

1. Nomenclature

G- adopted balancing accuracy class [mm / s],
ω - angular velocity [rad / s],
e_per- permissible relative unbalance [g · mm / kg],
U_dop- permissible residual unbalance [g],
m - rotor mass [kg],
r- rotor radius [mm].

2. Introduction

In rotating machinery, bearings are the elements that take over a large part of dynamic loads. In order to maintain the proper dynamic condition of the machine, it is necessary to supervise the condition of the bearings, e.g. through vibration diagnostics, which, already during the raceway normalization period; enables the diagnostician to observe warning symptoms that may suggest accelerated wear of the machine bearings [1, 2].

The loads applied to the rolling bearing are transferred by the rolling elements from the shaft and the inner ring to the outer ring [3]. Surface fatigue defects, such as deformation and cracks, occur in the bearing during the load transfer. Physical damage to a material is a diffuse defect in the form of vacancies, microcracks, micro-voids or damaged micro-volumes, which reduce the effective or load-bearing part of the material [2, 4]. Small damage to the bearings may directly affect the shafts on which they are mounted, as well as other structural elements, and thus lead to their damage, where the repair costs would be much higher than the cost of bearing replacement. The consequence of the occurrence of bearing defects may be an increase in statistical values of the vibration signal and the appearance of new amplitudes in the FFT spectra [1]. Here, feedback arises, where bearings in poor dynamic condition increase the transmission of vibrations by the machine, and high vibrations accelerate the degradation of the bearings [2]. The poor condition of rolling bearings may also prevent dynamic balancing of the rotor, and thus - lead to further propagation of bearing damage caused by an increased level of the machine's own vibrations.

Imbalance is a practical problem in the operation of many rotating machines, causing not only increased vibration of the machine, but also leading to accelerated wear of the rotor bearings. Majewski et al. [5] described the possibilities of automatic neutralization of unbalance with the use of mechanical elements (balls, rollers) and fluids, these systems are more and more common due to problems with manual balancing of rotors, but they remain a more expensive equivalent of classic rotors. Yamamoto et al. [6] made a rotor balancing system based on FPGA, which allows simultaneous online analysis of vibration parameters and FFT, which allows for on-going manual balancing of the machine. Le-Dinh [7] analyzed the possibilities of active balancing by applying balancing forces in opposition to corrective masses. However, in [8] it was proposed to neutralize the vibrations originating in the disc imbalance by using magnetic bearings with the AMB system by correcting the mass position.

The subject of this paper is the analysis of the dynamic state of the BMA K2400 centrifuge in terms of obtaining the appropriate balance accuracy class for the rotating element, i.e. the shaft with the basket and the centrifuge screen, as well as the diagnosis of the centrifuge in terms of the vibration level at individual bearing nodes. The article is intended for a wide range of recipients professionally related to maintenance in industrial plants, scientists interested in the operation of machines, as well as for other people interested in the subject of rational operation of rolling bearings.

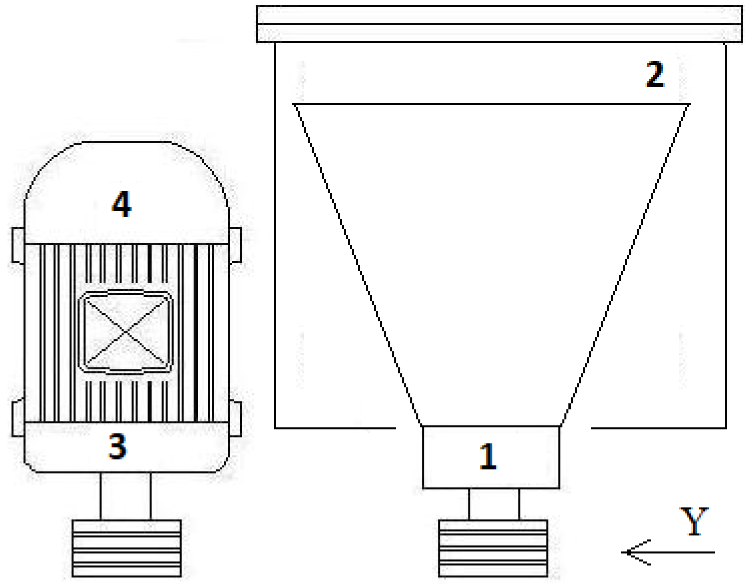

Fig. 1. Schematic sketch of the centrifuge system, measuring points are marked with numbers

3. Material and methods

After the scheduled maintenance shutdown, the NU315 ECP/C3, NU316 ECP/C3 and 7316 bearings in the BMA K2400 centrifuge were replaced. According to the numbering of measurement points in fig. 1 - the NU315 ECP/C3 bearing as a free support was at measurement point 1, while the paired bearings NU316 ECP/C3 and 7316 were in the housing inaccessible with the basket attached to the centrifuge. Consequently, the measuring point no. 2 is to show the vibrations of the entire centrifuge, and in particular of the discussed pair of bearings. The pictorial picture of the housing of the NU316 ECP/C3 and 7316 bearings is presented in fig. 2. - view of the centrifuge with the basket and top cover disassembled. The mass of the centrifuge rotor, which consists of the mass of the shaft, basket and screen, was determined at 250 kg. The radius of the centrifuge on which the correction masses are mounted was assumed as 1000 mm. The rotational speed of the centrifuge at


Fig. 2. View of the inside of the centrifuge with a disassembled basket - in the central point the housing of a pair of bearings NU316 ECP/C3 and 7316

100% efficiency (constant rotation speed during operation) is 1875 rpm.

The measurements were performed using the DIAMOND 401 AX device, equipped with Wilcoxon 780B acceleration sensors with a sensitivity of 100mV/g. The illustrative picture of the device is presented in Fig. 3. The device is characterized by typical measurement errors <1%, in extreme conditions <3%. The device is paired with the above-mentioned sensor has the following measuring ranges:

a) Velocity: 0-500 mm / s (disturbance <0.01 mm/s RMS for 10Hz - 1kHz)

b) Acceleration: 0-500 m / s2 (disturbances <0.01 mm/s^2 RMS for the frequency 10Hz - 1kHz)

The rotational speed measuring range is 60 rpm - 20,000 rpm, with a measurement error of ± 0.1%. When measuring the phase angle, the relative error is ± 1 °, while the resolution is 1 °. For run-up / coasting analysis, the measuring range is from 120 rpm to 20,000 rpm, maximum number of measuring points: 500.

Fig. 3. Measuring device in use with a vibration sensor

The main parameters that were used to determine the dynamic state of the machine were RMS and Peak (P-K) vibration velocity in the frequency range 2Hz - 1kHz, which is suggested in ISO 10816-3: 2009 / AMD 1: 2017 [9] and ISO 20816- 1: 2016 [10], as well as AVG and Peak (P-K) of the vibration acceleration envelope in the frequency range 500 Hz - 10 kHz, which was suggested in [11, 12, 13]. The frequency of vibration measurements was selected in accordance with the recommendations of the above-mentioned norm [9, 10]. FFT spectra of velocity and vibration acceleration were also made for the analysis.

Bearing nodes no. 1-4 have been assigned to the appropriate classes in accordance with the recommendations of the withdrawn standard ISO 10816-3: 2009 / AMD 1: 2017 [9]. The limit values for individual classes are described in Table 1. node no. 1 as a flexible node was assigned to class III, nodes no. 2 and 3 as rigid nodes were assigned to class II, and node no. 4, due to high susceptibility to vibrations, was assigned to class IV.

Table. 1.

Typical limits of vibration intensity classification zones - source: [9]

V_RMS [mm/s]	Class I	Class II	Class III	Class IV
0,28	A	A	A	A
0,45	A	A	A	A
0,71	A	A	A	A
1,12	B	A	A	A
1,8	B	B	A	A
2,8	C	B	B	A
4,5	C	C	B	B
7,1	D	C	C	B
11,2	D	D	C	C
18	D	D	D	C
28	D	D	D	D
45	D	D	D	D

To define the balancing criteria, the ISO 21940-11: 2016 [14] standard was used, where the balancing accuracy class for centrifuges was G 6.3. The residual mass in the G 6.3 class was calculated using the formulas (1), (2) also taken from the above-mentioned standards.

$$e_{per} = \frac{G \cdot 10^3}{\omega} \tag{1}$$

$$U_{dop} = \frac{m \cdot e_{per}}{r} \tag{2}$$

The permissible residual unbalance was 8.03 g.

4. Result and discussion

Table. 2.

The statistical values of the time course constituting the vibration parameters - measurement no.1

Measuring point:	Bearing node No. 1			Bearing node No. 2		
Direction:	Axial	Radial X	Radial Y	Axial	Radial X	Radial Y
RMS [mm/s]		**24,99**	**18,45**	**5,17**	**2,93**	**6,00**
Peak [mm/s]		**39,01**	**30,48**	**13,40**	**5,49**	**9,60**
Envelope Peak [mm/s^2]		38,81	26,72	3,96	2,93	5,21
Envelope AVG[mm/s^2]		8,24	4,16	1,06	0,82	0,99
Measuring point:	Bearing node No. 3			Bearing node No. 4		
Direction:	Axial	Radial X	Radial Y	Axial	Radial X	Radial Y
RMS [mm/s]	**4,04**	**4,66**	**10,17**		6,78	**20,10**
Peak [mm/s]	**7,75**	**8,07**	**15,53**		11,24	**30,55**
Envelope Peak [mm/s^2]	11,25	8,46	11,44		11,26	9,14
Envelope AVG[mm/s^2]	2,14	2,23	2,48		3,27	3,09

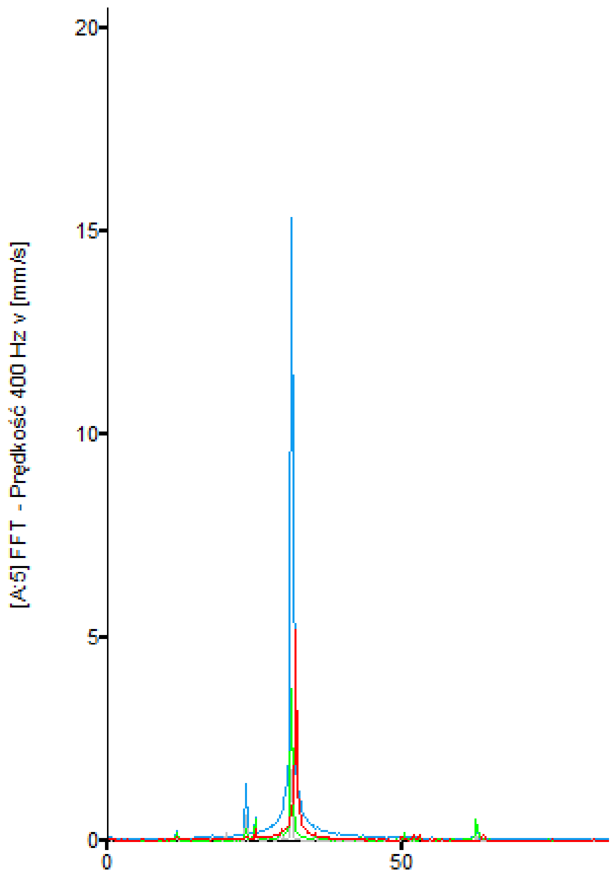

Fig. 4. FFT spectrum of the vibration velocity at the bearing node No. 1 in the radial direction X

Table 2 shows the parameter values measured in the as-is condition. The measured values from table 2 were compared with the normative values from table 1 and marked with yellow (warning states) and red (alarm states). The dynamic condition of the machine was assessed as unacceptable. FFT spectra of velocity and acceleration were also analyzed. An example of the speed spectrum in the radial direction X is presented in Fig. 4, where the amplitudes obtained in the described measurement are marked in blue, and the spectra from previous measurements on this machine are marked with other colors.

In relation to the FFT velocity spectra, can be observed a three times higher amplitude value at the rotational frequency of 31.25 Hz compared to the highest value so far measured, as described by Dwojak and Rzepiela in [1], this may suggest an unbalance of the machine. The experience with rotating machines to date suggests that in the case of an increased value of the RMS parameter of vibration velocity, with the simultaneous lack of anomalies in relation to the AVG and Peak parameters of the vibration acceleration envelope, the assumption about the machine's unbalance may be confirmed.

In relation to the above diagnosis, the test was carried out single plane balancing. Due to the nature of the

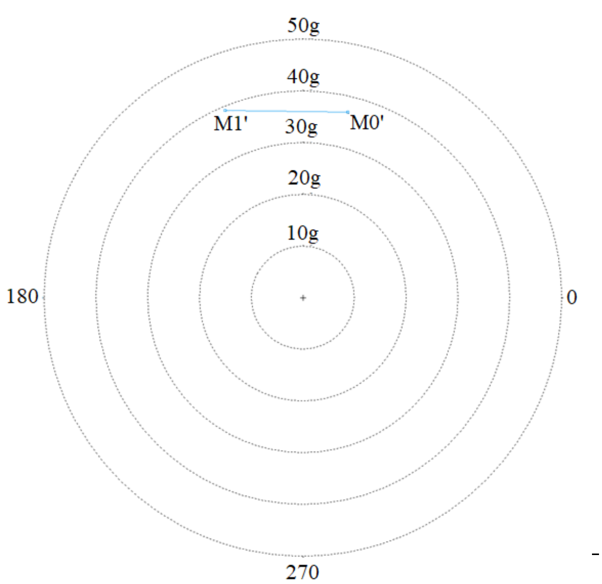

Fig. 5. Correction mass values for the first balancing

centrifuge load by the belt transmission, it was decided to perform a single-plane balancing in the radial direction Y.

Fig. 5 and Table 3 present the course of one of the centrifuge balancing tests, which turned out to be ineffective as a consequence of the absolute lack of effects when adding mass at phase angles indicated by the operated measuring device. An important fact is that the centrifuges in a given engine room, as a rule, have been balanced so far on a single plane, due to the difficulties in correcting the mass of the pulley, which could constitute the opposite plane.

As a consequence of the impossibility of balancing the centrifuge, with the vibration level assessed as unacceptable, it was recommended to inspect the centrifuge with particular emphasis on the centrifuge bearings and vibration isolators of the centrifuge.

Table. 3.

Values of balancing parameters in the Y axis - measurement no. 1

Measurment	Single-plane			
	Correction mass		Amplitude	
	Value [g]	Degree [°]	Value [g]	Degree [°]
Initial measurment	-	-	25,81	185,87
Measurement with a test weight	24,00	0	35,63	205,91
Verification measurement No. 1	42,99	122	30,04	224,78
Residual mass	50,03	161		

The first of the corrective actions implemented by the maintenance services, despite the diagnostic recommendations, was the replacement of the engine bearings (6317C3) in nodes No. 3 and 4, which, unlike the bearings of the centrifuge itself, remained unchanged during the current shutdown. For the needs of the maintenance service impact assessment, another centrifuge diagnosis was performed, the diagnostic results are presented in Table 4.

Table. 4.

The statistical values of the time course constituting the vibration parameters - measurement no. 2

Measuring point:	Bearing node No. 1			Bearing node No. 2		
Direction:	Axial	Radial X	Radial Y	Axial	Radial X	Radial Y
RMS [mm/s]		**26,31**	**17,64**	**6,15**	2,43	**8,78**
Peak [mm/s]		**41,01**	**29,66**	**9,65**	4,68	**13,51**
Envelope Peak [mm/s^2]		56,54	**119,62**	5,05	2,70	6,98
Envelope AVG[mm/s^2]		15,75	**30,60**	1,09	0,73	1,00
Measuring point:	Bearing node No. 1			Bearing node No. 2		
Direction:	Axial	Radial X	Radial Y	Axial	Radial X	Radial Y
RMS [mm/s]	**4,04**	1,83	**11,80**		**13,30**	**24,34**
Peak [mm/s]	**7,75**	3,61	**18,19**		**20,37**	**36,18**
Envelope Peak [mm/s^2]	11,25	5,30	4,93		5,41	7,21
Envelope AVG[mm/s^2]	2,14	1,28	1,16		1,82	2,36

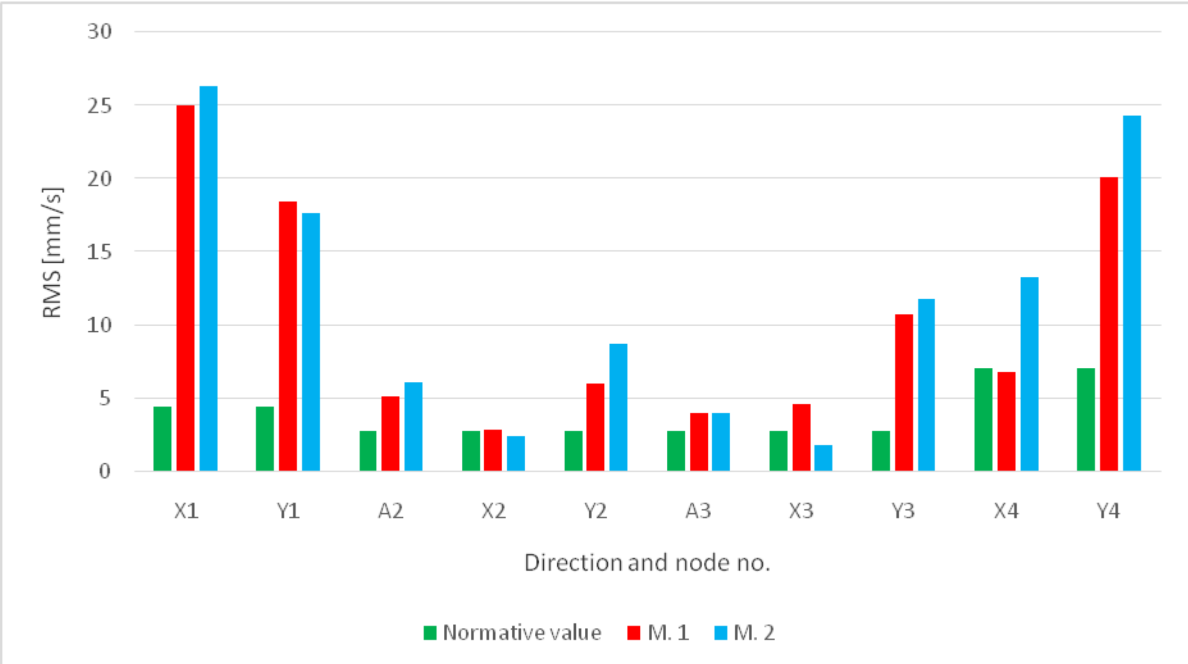

Fig. 6. Compare the normative values of RMS velocity with measured in measurements no. 1 and 2

Fig 6. shows the ratio of the RMS velocity values from Tables 2 and 4 to the normative values presented in Table 1. As can be seen, the velocity parameter values did not change significantly by replacing the motor bearings. Comparing the results from Tables 2 and 4, however, an increase in the Peak and AVG parameters of the envelope of vibration acceleration in bearing nodes 1 and 2 can be observed. suggest accelerated wear of the bearings.

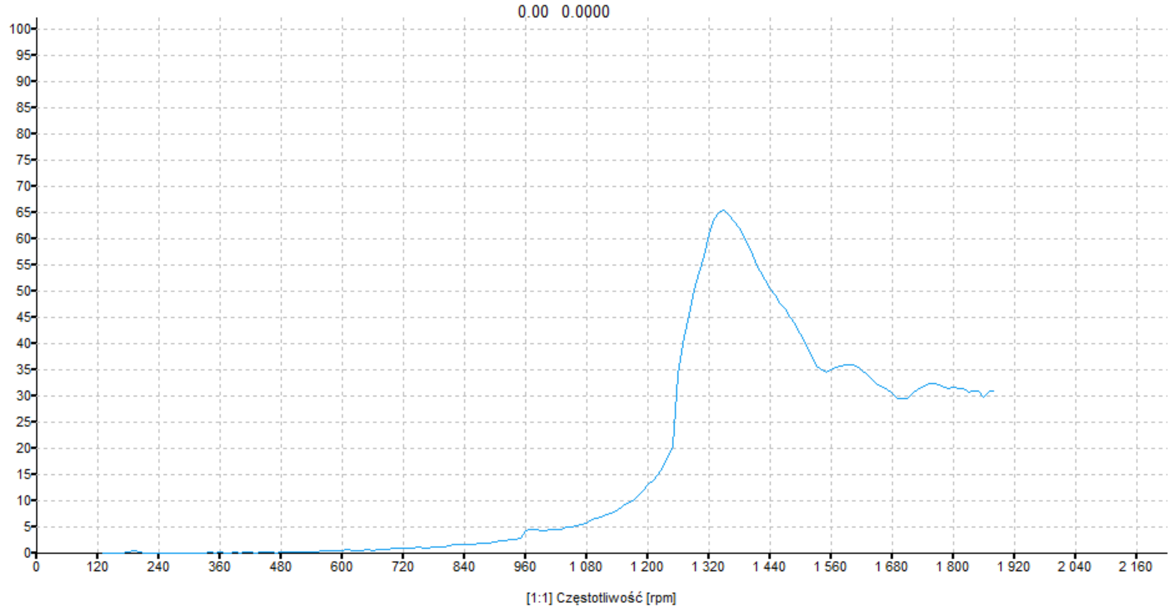

Fig. 7. Velocity values in the freewheel diagram in the Y direction

In order to rule out problems with the machine's vibration isolation, a run-up chart was made, which was suggested by Zachwieja [15]. As can be seen in Fig. 7, the centrifuge, operating at 100% efficiency, does not operate at the critical / resonant frequency.

As a result of the lack of effects after replacing the motor bearings and after analyzing the vibration measurement results presented in Fig. 6, a decision was made to inspect the centrifuge bearings. The diagnostics was performed again, but it concerned only the bearing node No. 1 with the disassembled basket. The diagnostic results are presented in Table 5. This allowed for a preliminary diagnosis of the reason for the impossibility of balancing the centrifuge. The NU315 ECP / C3 bearing, despite being brand new, was damaged during the first measurement, which most likely resulted from a technological defect of the outer ring of the bearing. The pitting with dimensions of approximately 20 mm x 5 mm is presented in Fig. 8.

Fig. 8. Bearing NU315 ECP / C3 with damaged outer ring

For the purpose of assessing the dynamic condition after replacing the NU315 ECP / C3 bearing, another diagnosis was performed in the bearing node No. 1 with the cage disassembled, and the results are also presented in Table 5 for comparison.

On the basis of the measurement results presented in Table 5., can be observed a drastic, 3-fold decrease in the value of the Peak parameter of the vibration acceleration envelope and also a significant decrease in the value of the AVG parameter. On the other hand, the statistical parameters of the velocity course increased, which is definitely an unfavourable situation, no less detailed inspection of the centrifuge and the belt transmission did not show any defects, and the values of the RMS velocity parameter did not exceed the alarm values of 4.5 mm/s adopted for this class node.

Table. 5.

The statistical values of the time course constituting the vibration parameters - measurements no. 3 and 4

The condition of the bearing	Broken			New		
Direction:	Axial	Radial X	Radial Y	Axial	Radial X	Radial Y
RMS [mm/s]		1,35	0,80		2,43	2,98
Peak [mm/s]		4,40	3,24		4,65	6,26
Envelope Peak [mm/s^2]		67,34	42,61		20,94	13,67
Envelope AVG[mm/s^2]		17,26	3,45		3,45	2,82

After the basket was installed and the centrifuge was restored to its working condition, dynamic balancing was started. The visualization of the centrifuge with the basket installed during operation is presented in Fig. 9. Again, it was decided to perform a single-plane balancing in the radial direction Y in accordance with the sketch in Fig. 1. The balancing process is presented in Fig. 10 and in Table 6.

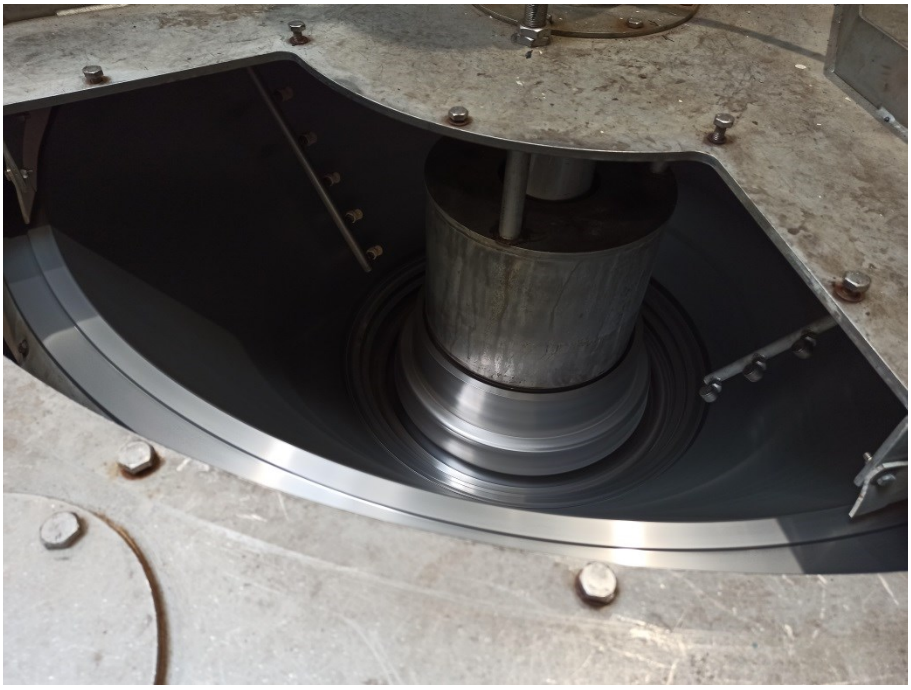

Fig. 9. Visualization of the centrifuge with mounted basket and housing

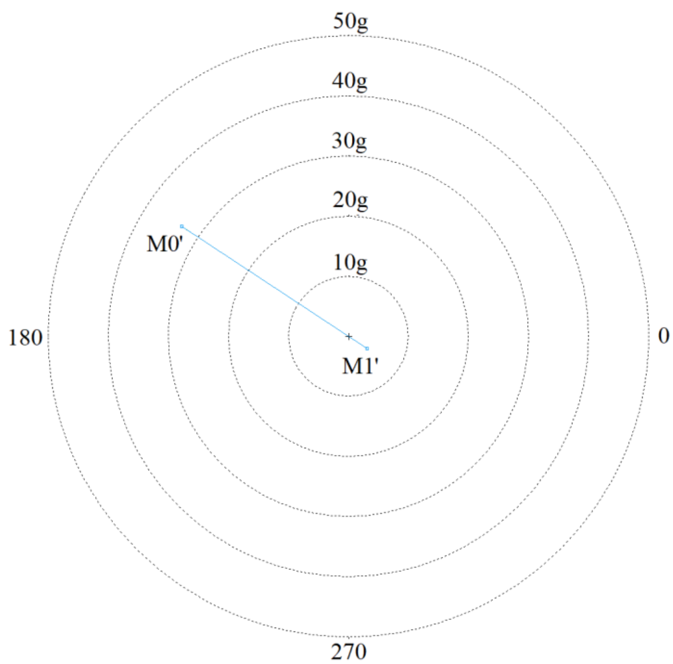

Fig. 10. Correction mass values for the second balancing

Table. 6.

Values of balancing parameters the Y axis - measurement no.2

Measurement	Single-plane			
	Correction mass		Amplitude	
	Value [g]	Degree [°]	Value [g]	Degree [°]
Initial measurment	-	-	15,04	275,94
Measurement with a test weight	5,81	0	17,30	280,74
Verification measurement No. 1	33,23	147	1,69	95,85
Residual mass	3,14	327		

The residual mass w described in Table 6. - 3.14 g meets the requirements of the permissible residual weight of 8.03 g. The results of diagnostics of the bearing arrangement No. 1 after balancing are presented in Table 7.

Table. 7.

The statistical values of the time course constituting the vibration parameters - measurement no. 5

Measuring point:	Bearing node No. 1		
Direction:	Axial	Radial X	Radial Y
RMS [mm/s]		**7,67**	1,99
Peak [mm/s]		**24,83**	3,74
Envelope Peak [mm/s^2]		22,31	11,61
Envelope AVG[mm/s^2]		5,88	3,88

As can be seen in Table 7, the dynamic condition of the machine has improved compared to the previous measurements described in Table 4, however, the criterion of the average effective speed described in ISO 10816-3: 2009 / AMD 1: 2017 [9] and ISO 20816-1: 2016 [10], and adopted as v_RMS = 7.1 mm / s has not been met, and for the purposes of the test, it is considered superior to the criterion of the balance quality class set at the level of G 6.3.

Due to the determined small correction mass given in Table 6, and at the same time that it was not possible to apply two-plane balancing, in the case in question, a single-plane balancing was carried out in the radial direction X. The results of balancing are described in Table 8.

Table. 8.

Values of balancing parameters in the X axis - measurement no. 3

Measurement	Single-plane			
	Correction mass		Amplitude	
	Value [g]	Degree [°]	Value [g]	Degree [°]
Initial measurment	-	-	4,92	170,15
Measurement with a test weight	5,81	120	0,51	155,58
Verification measurement No. 1	6,46	118	2,04	239,50
Residual mass	2,68	188		

The correction of the mass distribution in the case of the X plane, described in Table 8, was successful in meeting the appropriate criterion of the balance quality class. The suggested mass correction, however, was inconsistent with the correction mass suggested by the measuring instrument in relation to the Y-axis balancing. The above problem could be solved by two-plane balancing, which, however, due to technical reasons, could not be achieved. Table 9. supplements Table 6. with an additional measurement with a correction mass. Table 10. presents the results of the final vibration measurements on all bearing nodes of the centrifuge and the engine. Fig 11. shows the ratio of the RMS velocity values from Tables 2, 4 and 10 to the normative values presented in Table 1.

Table. 9.

Values of balancing parameters in the Y axis - measurement no. 4

Measurement	Single-plane			
	Correction mass		Amplitude	
	Value [g]	Degree [°]	Value [g]	Degree [°]
Initial measurment	-	-	15,04	275,94
Measurement with a test weight	5,81	0	17,30	280,74
Verification measurement No. 1	33,23	147	1,69	95,85
Verification measurement No. 2	3,14	327	3,88	69,57
Residual mass	8,12	300		

The obtained value of the correction mass of 8.12 g is close to the determined value of 8.03 g and oscillates within the measurement error, which allows for its conditional acceptance.

Table. 10.

The statistical values of the time course constituting the vibration parameters - measurement no. 6

Measuring point:	Bearing node No. 1			Bearing node No. 2		
Direction:	Axial	Radial X	Radial Y	Axial	Radial X	Radial Y
RMS [mm/s]		1,92	4,07	2,42	2,17	1,98
Peak [mm/s]		4,53	9,17	6,64	4,45	3,48
Envelope Peak [mm/s^2]		36,44	20,69	4,68	3,21	4,41
Envelope AVG[mm/s^2]		6,67	3,88	1,19	0,81	1,05
Measuring point:	Bearing node No. 3			Bearing node No. 4		
Direction:	Axial	Radial X	Radial Y	Axial	Radial X	Radial Y
RMS [mm/s]	1,13	0,62	2,50		3,80	6,64
Peak [mm/s]	2,69	1,50	4,50		6,65	10,48
Envelope Peak [mm/s^2]	4,03	6,84	12,39		7,19	7,50
Envelope AVG[mm/s^2]	1,06	1,63	1,81		2,35	2,25

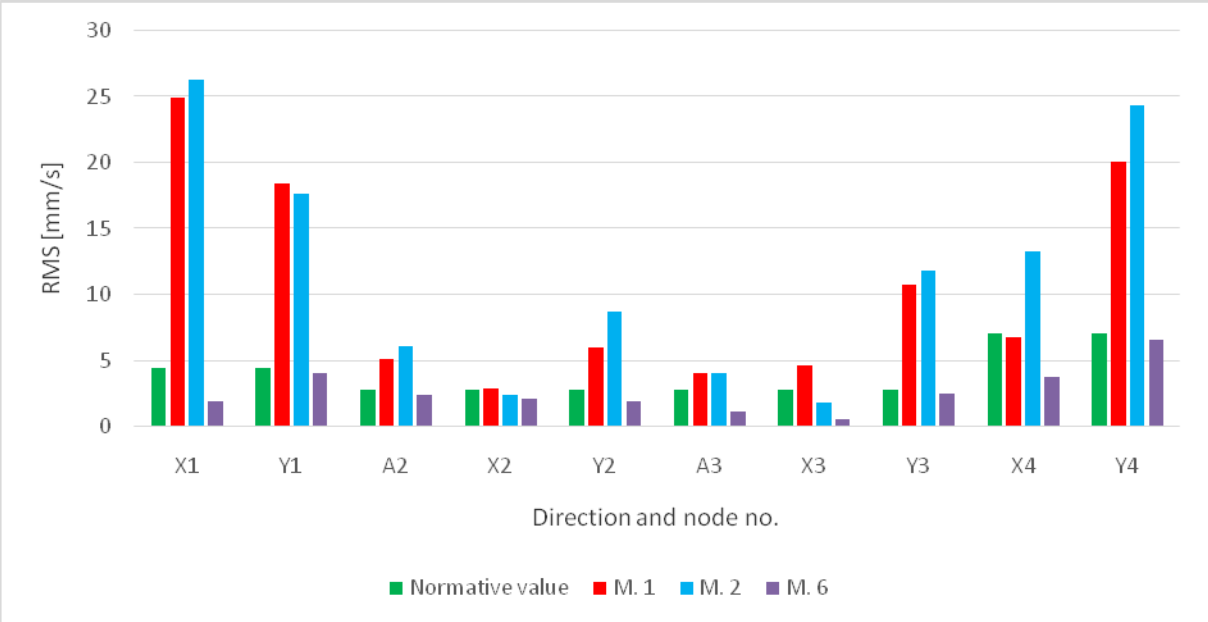

Fig. 11. Compare the normative values of RMS velocity with measured in measurements no. 1, 2 and 6

The diagnostic results described in Table 10. and Fig. 11. allow to classify the dynamic condition of the machine at a good level. As can be seen in the final results from table 9, the criterion of the balancing quality class has approximately been met.

5. Conclusions

The dynamic state of the machine during the measurements was conditioned by two factors - a technological defect of the bearing and the unbalance of the rotor. The basis for the correct balancing of the machine is the good dynamic condition of the bearings installed in the machine.

Maintenance in many plants is based primarily on the previous experience of their employees, usually not deepened by theoretical knowledge, but only supported by the experience of external companies operating in other production plants. The above-described criterion of the proper dynamic condition of bearings is often omitted when commissioning diagnostic works, and the order itself, due to the experience of in-house maintenance services, is based only on the order to balance the rotor, as in the presented case.

The values of the AVG and Peak parameters of the vibration acceleration envelope are the first criterion in the assessment of the dynamic condition of rolling bearings, which does not relate to the mass distribution on the rotor. The non-increased values of these parameters, as well as the information about the recent replacement of the centrifuge rotor bearings, led to the assumption that the problems with balancing the centrifuge result from other - described in the paper - reasons.

The appearance of a technological defect on the outer ring of the bearing, which is a friction pair with a housing, is not a typical damage for this type of machines and was an interesting problem to be described in this article. The consequence of the occurrence of bearing defects may be an increase in statistical values of the vibration signal and the appearance of new amplitudes in the FFT spectra. A vicious circle is created here, where bearings in poor dynamic condition increase the transmission of vibrations through the machine, and high vibrations accelerate the degradation of the bearings. The poor condition of rolling bearings may also prevent dynamic balancing of the rotor, and thus - lead to further propagation of bearing damage caused by an increased level of the machine's own vibrations.

The development of Industry 4.0 and Big Data technology, as well as the growing awareness of maintenance services about the need to invest in rapid response systems, prompts production plants to install more and more advanced systems for diagnosing machines on-line. However, it is worth paying attention to the fact that computer-based analyzes are not yet able to replace the work of diagnosticians who, due to their experience in many plants, have greater opportunities to analyze potential causes of damage and failure. There is also the problem of CMMS systems integration with vibration diagnostics databases, which would be crucial for computer recognition of potential sources of incorrect operation.

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