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JÓZEF RYBCZYŃSKI[1]

Analysis of additional vibrations encountered during investigations of rotor dynamics

In the paper an effort has been made to explain unexpected rotor vibrations, which have been encountered during experiments on the research stand for the analysis of rotors and bearings. Presented have been the results of experimental investigations conducted on a research rig for the analysis of rotors and bearings developed and operated at IFFM PAS in Gdańsk. Subsequently, the rotor with its all complementary parts has been modelled and calculations were made using the NLDW computer program. Comparisons have been made between the results of experiments and corresponding calculations. These showed that the calculations reveal very good consistency with the experiments for the rotor velocities up to about $1.2 \times n_{kr}$, and absolutely are not in tune in the case of phenomena observed at higher rotational velocities. Conducted has been analysis for the possible reasons of such lack of consistency. The results from that analysis, apart from their fundamental character, are of significant practical importance, which are to be exploited in further experimental works planned in future on that rig.

1. Introduction

The research programme for investigations of rotor dynamics at IFFM PAS in Gdańsk is realised with the view to obtain diagnostic knowledge of rotating machinery. Its experimental part is conducted based on the constructed rig for the analysis of dynamics of rotors and bearings. The main objective of this part of investigations is verification of developed, at IFFM PAS, calculation models for the analysis of complex rotating machinery. Experimental investigations have been conducted on a rig equipped with the symmetrical rotor founded in three cylindrical slide bearings loaded with the discs between the bearings. Comparative calculations have been conducted using one of the versions of the NLDW computer program.

It turned out that the results of measurements and corresponding results of calculations differ significantly in some ranges of rotational velocities. In other ones, on the other hand, their consistency is satisfactory. Calculations do not

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present the existence of the increase of vibrations in the case when the rotor rotational velocity exceeds its first critical velocity, whereas these are present in the experiments. This means that either the model, on which the calculations were based, does not reflect adequately the physical properties of the object, or the experimental conditions have not been reflectede in the model with the required accuracy.

After the reconstruction of the rig further experimental works are planned. Prior to starting these experiments, however, the reasons for the increase of rotor vibrations in ranges of rotational velocity must be explained, in cases where these were not expected, and also explained must be reasons why the computer calculations do not reveal such additional increases of vibrations.

Considered have been, amongst the others, the following factors as hypothetical reasons for the found discrepancies:

- structural heterogeneity of rotor elements (for example the crack of the shaft),
- angular or radial misalignment of the shaft in the coupling,
- unknown rotor natural frequencies in the range of its possible rotational velocities,
- strong resonances of the rig's foundation frame, which vibrations propagate through the bearings onto the rotor,
- clearances in the coupling connecting two parts of the shaft,
- loosening of the discs on the rotor shaft at high revolutions (interference fastening),
- displacement of one of the bearings with respect to the geodesic line of rotor,
- twisting of the shaft,
- excitations coming from the propelling elements,
- vibrations of measurement sensors during investigations,
- errors of measurement.

In the further part of the paper presented is the analysis of these factors from the point of view of possible reasons, of experimentally explored, but unexpected increases of the vibration amplitude of the rotor. Considered have been and rejected, as little plausible, a series of hypothetical reasons for such vibrations.

2. Conditions and results of experimental investigations on a research rig

Experimental investigations realised have been on a model of the rotating machinery such as the research rig for investigations of the dynamics of rotors and bearings. The rig is so universal that it enables to install and investigate rotors of different configuration, both with respect to the distribution of mass as well as the number and kind of bearing supports. This enables simulation of various dynamical phenomena and states of motion existing in this type of machines such as: imbalance, misalignment of the line of shafts. The rig enables changes of the type of bearings as well as the stiffness of their supports. A detailed description together with the measurement capabilities of the rig can be found in [3-5]. Modelling of particular isolated dynamical phenomena, which may occur in the rotating machinery, is practically possible only on that kind experimental objects, as the complexity of the real machine constructions renders imposing of too many interacting phenomena, from which it is too difficult to separate simple symptom-defect relations, and hence to built diagnostic relations.

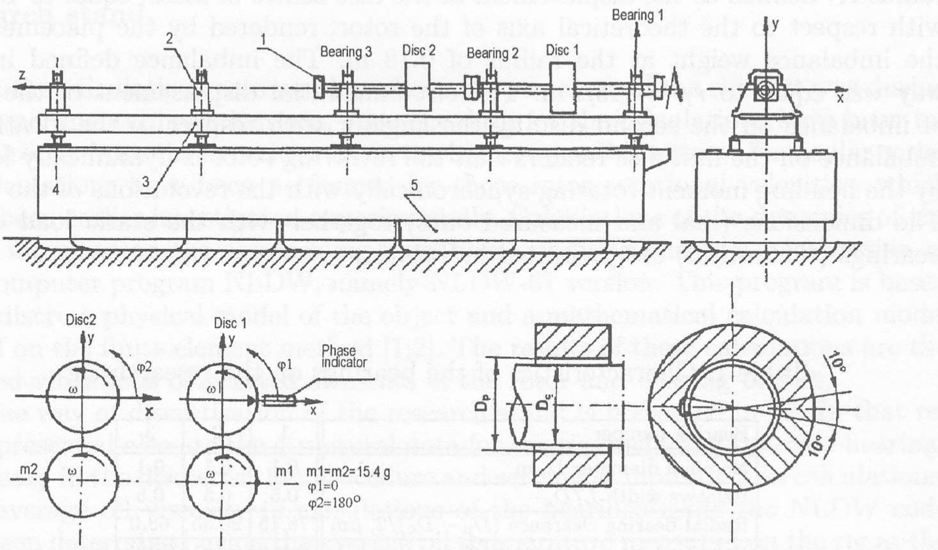

Fig. 1. A schematic of the research rig and bearings explaining the notations and orientation in the co-ordinate system; 1 – rotor, 2 – bearing stands, 3 – steel frame, 4 – foundation block, 5 – pneumatical shock absorbers; m – imbalance masses mounted on discs, φ – angle describing the location of imbalance masses.

A schematic of investigated objects together with identification of the notations of elements of the rig, rotor orientation in the assumed co-ordinate system as well as the notation of the measured quantities used in the further part of the paper have been presented in Fig. 1.

The rotor and the state of the rotor load are characterised by the following quantities:

- mass of the shaft (sum of both parts with the coupling) $m_w = 215$ kg,
- mass of each of two loading discs $m_t = 185$ kg,
- total weight of the rotor $m = 585$ kg,
- distance between the neighbouring bearings $L = 1.4$ m,

- loading discs located in the middle of each part of the shaft,
- the shaft supported on three cylindrical slide bearings with two lubrication pockets placed in the horizontal plane of division (in accordance with the Fig. 1),
- imbalance of each disc (characterised by the imbalance radius r_N) is 15μm and the imbalance weights on both discs are circumferentially displaced by π with respect to each other.

Static load (theoretical) of the rotor bearings results from the total mass of the rotor and the system symmetry. The dynamical load of rotor are the centrifugal forces acting on the weights symmetrically mounted on the discs and rendering in this way the rotor imbalance. The imbalance is characterised by the imbalance radius rN defined as the displacement of the disc centre of mass, equal to 185 kg, with respect to the theoretical axis of the rotor, rendered by the placements of the imbalance weight at the radius of 0.18 m. The imbalance defined in this way was equal to $r_N = 15\mu$ m. The circumferential displacement of the mass of imbalance on the second disc by the angle π with respect to the location of imbalance on the first disc renders that the revolving rotor is dynamically loaded by the bending moment rotating synchronically with the revolutions of the rotor. The dimensions (real and measured ones) together with the static load of the bearings (theoretical) are presented in Tab. 1.

Table 1. Characteristics of the bearings on the research rig

Bearing number	1	2	3
Nominal diameter D, m	0.1	0.1	0.1
Relative width L/D, –	0.5	0.5	0.5
Radial bearing clearance $(D_p - D_c)/2$, μm	78.15	90.35	68.0
Relative bearing clearance, %	0.156	0.181	0.136
Static load (theoretical F_{st}, N	1436	2870	1436
Horizontal support stiffness C_x, N/μm	87.2	69.3	78.3
Vertical support stiffness C_y, N/μm	317.9	307.0	312.5
Horizontal damping D_x , Ns/μm	0.03	0.03	0.03
Vertical damping D_y, Ns/μm	0.03	0.03	0.03

The following quantities have been recorded during measurements

- absolute displacements of the centres of both discs in two perpendicular directions,
- relative displacements of three bearing journals in two perpendicular directions,
- absolute displacements of three bearing bushes in two perpendicular directions,

- rotor rotational velocity,
- oil temperature at the inlet and outlet from the bearings.

All displacements were measured using the addy current sensors of relative displacements. Measurement signals were synchronised with each other and with the angular location of rotor through simultaneous sampling of measuring signals in all channels and implementation of the key-phaser. This enabled the analysis of inter-correlation of the trajectory of motion of the elements of the rotor.

The results of investigation procedures on a three support stand have been presented in [6] and the results in [7].

3. Computer simulations of the experiments conducted on the research stand

In the calculations assumed have been the structure of the rig as it was during the experimental investigations. Also the data for the calculations have been taken directly from the experiments, to the highest possible extent. Particular series of calculations have been performed for those rotor rotational velocities, which have been earlier investigated experimentally. Calculations of the dynamics of the rotor mounted on the research rig have been conducted using a new version of the computer program NLDW, namely NLDW-61 version. This program is based on a discrete physical model of the object and a mathematical calculation model based on the finite element method [1,2]. The results of these calculations are the excited vibrations of selected elements of the rotor and bearing bushes.

The way of discretisation of the research stand is presented in [8]. In that report presented also are the structural data for the rig, including the slide bearings necessary in the discretisation procedure and serving as input data in calculations. The average oil viscosity in calculations of the bearings using the NLDW code has been determined using the average oil temperature measured on the rig at the inlet and outlet from the bearings. Stiffness and damping of particular bearing supports has been assumed as static, experimentally determined on the rig. These are presented in Tab. 1. Calculated have been all these quantities, which were the aim of measurements, i.e. instantaneous displacements of all discs, journals and bushes in two perpendicular directions. Calculations have been conducted using the NDLW program in the case of all experiments with the notation in the form of 1402...1419, where the stiffness support was calculated with account of the stiffness of the oil film.

4. Comparative analysis of results of experiments and calculations

The dynamical state of the rotor reflects in the shaft vibrations and the trajectories of bearing journals and discs. From the extensive experimental evidence described in [6] and [7] extracted has been a part of results, which illustrates best the discrepancies between the results of measurements and calculations. Explanation of these discrepancies has been the objective of the present work. Selected have been the experiments conducted at the rotor imbalance such as presented in Fig. 1, i.e. when the imbalance of both discs was the same with regard to the value, but the directions of the imbalance forces are displaced circumferentially by 180 degrees with respect to each other. In this way the rotor is subjected to the rotating bending moment and the rotor vibrations are relatively large.

Results of investigations are presented in the report in the form enabling direct comparisons when both the experimental and numerical results are cast onto one graph. In Fig. 2 presented are the distributions of maximal amplitudes of motions of particular elements of the rotor in the function of the shaft rotating velocity in the full range of rotating velocity. Such distributions are presented for both discs (absolute vibrations) and three bearing journals (journal vibrations with respect to the bush). The distributions have been plotted using the spline approximation of discrete values (denoted with the *) and measured ones (denoted as o).

It results from the graphs that the results of calculations using the NLDW-61 program does not reveal the rotor behaviour in the whole range of the rotor rotational velocity, for which the measurements have been conducted. Distributions of the amplitudes of vibrations, both for the journals and the discs does not reveal all the increases of amplitudes observed on the experimental distributions.

In the experimental distributions there are perceptible three local increases of vibrations, which is true for both discs and all three bearing journals. All three maximum values are appearing in the case of all five considered elements of the rotor at the same shaft rotational velocity (with the error less that 1%). This means that the vibrations of each of the elements are caused by the same phenomenon taking place in the rotating system. In Tab. 2 presented are values of the frequencies of particular increases of amplitudes of vibrations presented in the figures: separately for each rotor element and its means value. These are values taken out from the experimental distributions interpolated using the spline method. Presented are also the frequencies of rotor rotations for the cases of experiments closest to the locations of amplitude maximum (in the column "experiment").

Table 2. Table of resonance velocities of rotor

Experiment number	n rev/min	Resonance frequencies, Hz						
		experiment	Disc 1	Disc 2	Pin 1	Pin 2	Pin 3	Average
1407	2853	47.55	48.84	48.67	48.94	48.80	47.34	48.52
1411	3364	56.07	57.40	57.37	56.80	56.50		57.02
1414	3898	64.97	65.36	65.66	65.30	65.93	65.76	65.60

In Fig. 3 presented are the trajectories for both discs and three journals: measured on the rig (upper raw) and calculated ones (lower raw). Selected have

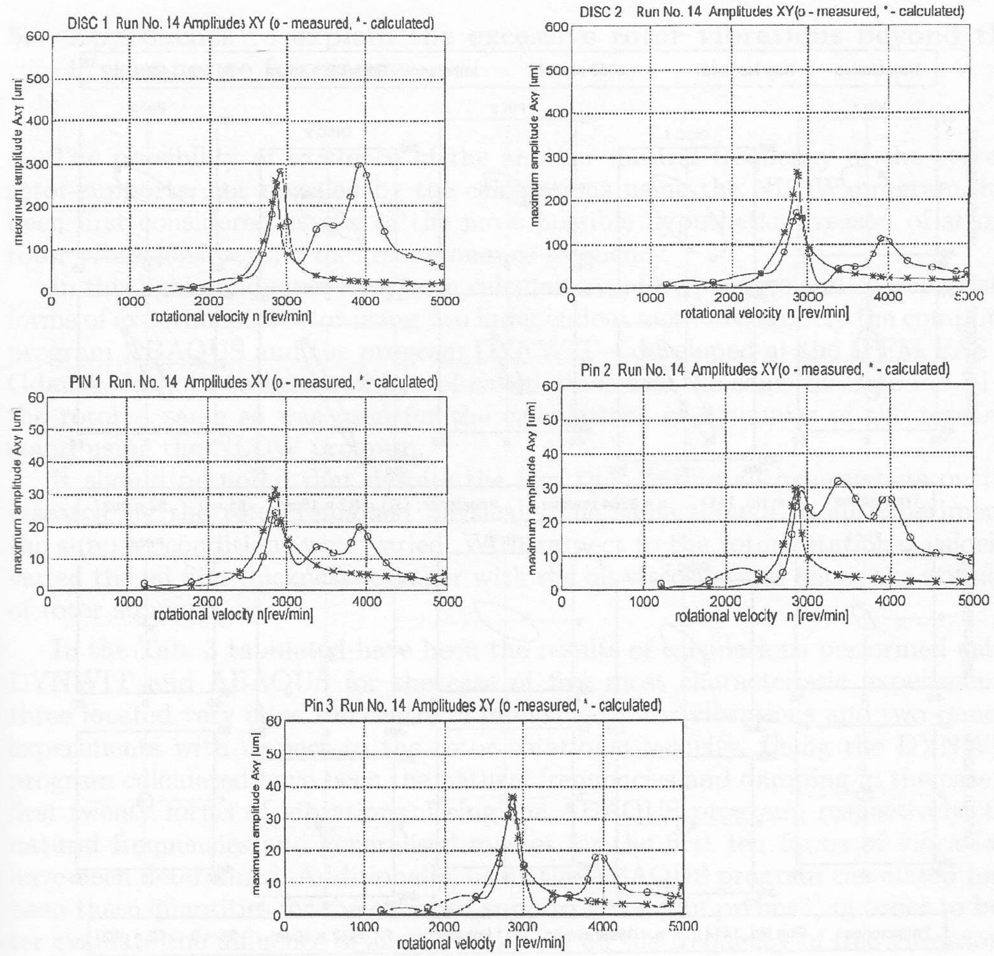

Fig. 2. Dependence of amplitude of vibrations of particular parts of rotor on the shaft rotational velocity.

been three characteristic cases – the experiments corresponding to the closest values of three found resonances. As can be seen, always the disc No. 1 exhibits the strongest vibrations. In the region of the frequency of 3900 rev/min vibrations have a direction close to the vertical one. The trajectory of disc No. 1 is a very slender ellipse. Such strong disc vibrations does not have any influence on the relative vibrations of bearing pins, where the trajectories follow the small circles. In the case of the first resonance the situation is completely different. The disc trajectories has a shape of wide ellipses and are accompanied by the strong relative vibrations of bearings bushes.

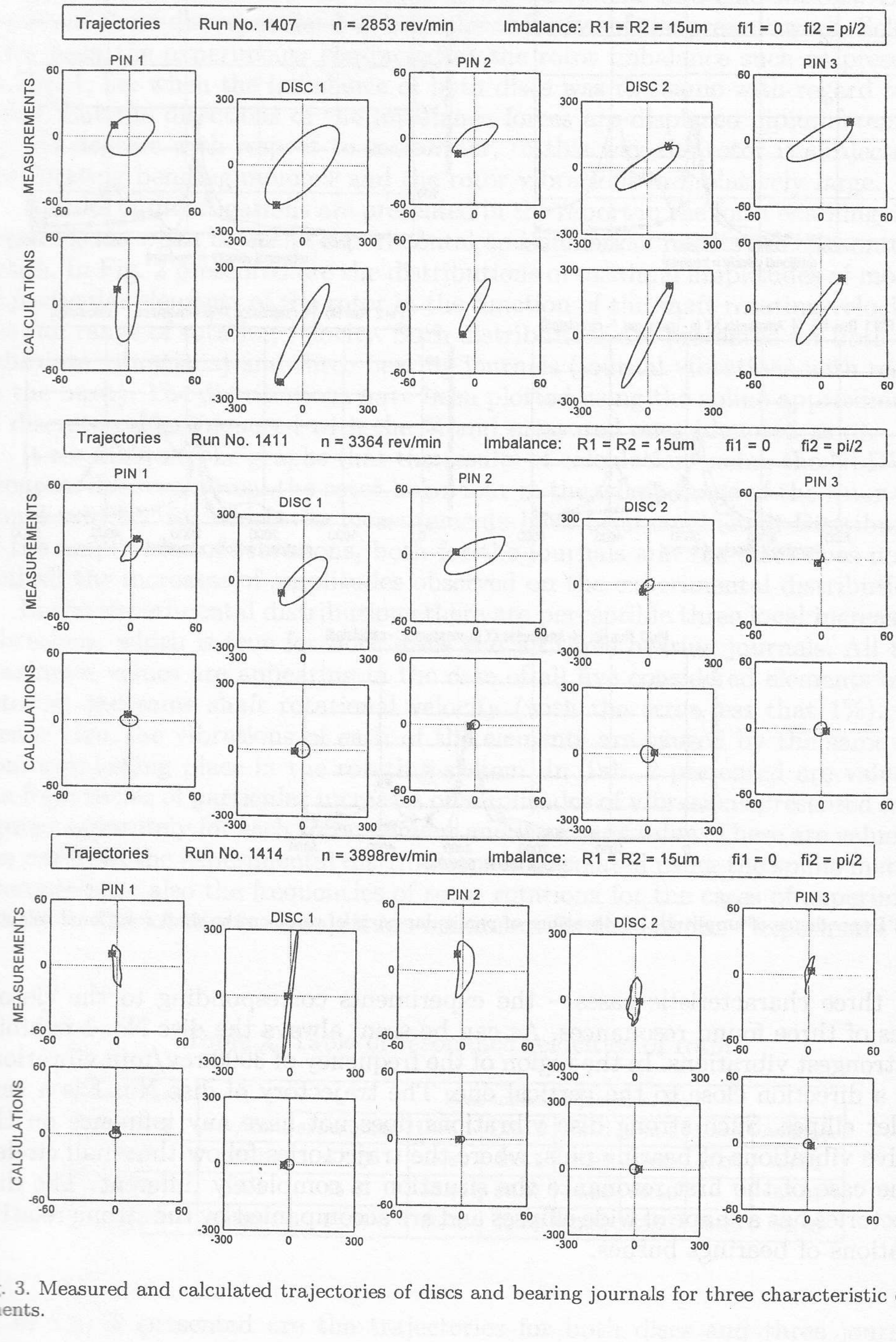

Fig. 3. Measured and calculated trajectories of discs and bearing journals for three characteristic experiments.

5. Approaches to explain the excessive rotor vibrations beyond the 1^{st} resonance frequency

The possibility of existence of the another natural frequency of the system rotor-supports, not revealed by the calculations using the NLDW program, has been first considered as one of the most possible hypothetical reason of strong rotor vibrations beyond its first resonance frequency.

In this light conducted have been calculations of frequencies and characteristic forms of experimental rotor using two independent methods, namely the computer program ABAQUS and the program DYNWIT – developed at the IFFM PAS in Gdańsk. In both cases the object of calculations was the same discrete model of the rotor – same as was used for the calculations of dynamics of the research stand using the NLDW program.

It should be noted that despite the fact that during all experiments on the research rig the rotor remained physically the same though in all experiments the support conditions were varied. With respect to the rotor rotational velocity varied the oil film thickness together with the oil viscosity and hence the stiffness of rotor support.

In the Tab. 3 tabulated have been the results of calculations performed using DYNWIT and ABAQUS for the case of five most characteristic experiments: three located very close the centre of intensified rotor vibrations and two remote experiments with respect to the rotor rotational velocity. Using the DYNWIT program calculated have been the natural frequencies and damping in the case of first twenty forms of vibrations. Using the ABAQUS program, respectively, the natural frequencies and generalised masses for the first ten forms of vibrations have been determined. Additionally, using the ABAQUS program calculated have been these quantities for the clamped support rotor ("in prisms") in order to better evaluate the influence of support stiffness on the frequency of free vibrations.

It results from the Tab. 3 that the particular natural frequencies calculated using the ABAQUS software does not stand the ones calculated using the DYNWIT program for the corresponding conditions. The differences result from the fact that the ABAQUS software takes principal stiffnesses as the rotor support stiffnesses, i.e. the reciprocal of the ratio of displacement to the acting force in the direction of displacement. The DYNWIT program can, on the other hand, account for the skewed stiffnesses, i.e. determined using the assumption that the force and the journal displacement rendered by the force are perpendicular. The specifics of the slide bearings is the fact that the skewed stiffnesses are significantly smaller than the principal stiffnesses and hence they influence stronger the system vibrations. Additionally, the principal stiffnesses are little dependent on the thickness of the oil film, and hence also the rotational velocity of the rotor. It is different in the case of skewed stiffnesses. The result of it is that the natural frequencies calculated using the ABAQUS program are almost independent from the rotor rotational velocity. Only the completely stiff support of the rotor meaningfully changes the natural frequencies corresponding to the same charac-

Table 3. Tabulation of natural frequencies of the rotor calculated using the DYNWIT and ABAQUS program for selected experiments

	DYNWIT – frequency, Hz					ABAQUS – frequency, Hz					
Exp. No.	1402	1407	1411	1414	1419	1402	1407	1411	1414	1419	fastened
n, 1/min	1212	2853	3364	3898	4891	1212	2853	3364	3898	4891	
Form 1	10.45	24.70	29.36	34.46	42.40	40.76	40.84	40.87	40.87	40.91	43.70
2	10.46	24.76	29.40	34.95	43.33	41.01	40.89	40.88	40.93	41.00	43.70
3	12.30	30.94	39.29	43.29	44.76	52.22	52.09	52.02	52.01	52.22	65.49
4	46.82	**46.59**	44.84	45.63	47.66	53.11	52.42	52.33	52.29	52.29	65.49
5	48.00	**47.85**	**46.33**	45.88	48.06	105.15	105.96	105.96	106.22	106.23	216.11
6	51.77	48.11	**47.79**	**47.74**	56.58	108.48	106.80	106.37	106.52	107.93	228.78
7	64.97	63.71	64.05	**62.34**	61.13	147.15	148.98	150.14	149.62	152.25	228.78
8	126.24	125.20	124.88	124.54	123.77	152.58	149.63	150.72	152.26	153.27	335.24
9	137.99	136.70	136.33	135.95	135.13	216.11	216.11	216.11	216.11	216.11	335.24
10	156.77	156.09	155.80	155.47	154.68	252.39	252.06	251.56	251.55	252.77	477.95
11	205.65	205.66	205.54	205.42	203.86						
12	207.17	206.32	205.95	205.52	205.14						
13	220.65	219.92	219.66	219.41	219.07						
14	234.34	233.67	233.46	233.25	232.81						
15	244.20	244.66	245.09	245.71	247.34						
16	293.77	294.63	294.93	295.24	295.92						
17	363.93	363.66	363.58	363.50	363.34						
18	381.94	383.04	383.39	383.75	384.57						
19	422.90	423.71	423.95	424.25	424.86						
20	430.07	430.62	430.83	431.04	431.57						
	DYNWIT – damping, Ns/mm					ABAQUS – generalised mass					
Exp. No.	1402	1407	1411	1414	1419	1402	1407	1411	1414	1419	fastened
n, 1/min	1212	2853	3364	3898	4891	1212	2853	3364	3898	4891	
Form 1	-27.27	-48.32	-46.76	-56.19	-3.74	433.85	432.16	429.61	431.18	428.04	455.21
2	-29.62	-44.62	-49.94	-46.77	-6.80	432.89	432.71	430.82	428.85	430.83	455.21
3	-59.92	-114.65	-140.82	-32.75	-72.32	435.79	435.03	433.41	433.66	430.15	379.08
4	-6.83	**-12.59**	-48.10	-21.51	-5.51	430.28	432.50	430.76	430.75	433.41	379.08
5	-3.93	**-5.31**	**-15.96**	-181.36	-74.57	15.60	15.58	15.58	15.58	15.57	339.76
6	-30.78	-50.55	**-5.50**	**-5.55**	-255.37	15.44	15.54	15.57	15.55	15.47	10.74
7	-17.75	-30.40	-33.18	**-34.74**	-34.73	15.16	15.16	15.16	15.17	15.08	10.74
8	-125.91	-136.85	-140.12	-143.74	-151.94	15.07	15.17	15.11	15.07	15.04	15.41
9	-135.54	-149.21	-153.27	-157.39	-166.92	339.76	339.76	339.76	339.76	339.76	15.41
10	-92.26	-91.53	-91.18	-90.75	-89.69	68.58	68.63	68.50	68.51	68.93	48.19
11	-58.08	-75.52	-80.49	-85.34	-45.86						
12	-56.57	-56.22	-54.30	-49.74	-95.04						
13	-41.87	-56.20	-63.62	-72.38	-91.82						
14	-109.42	-133.07	-140.18	-147.41	-162.24						
15	-92.05	-100.94	-101.87	-102.08	-101.62						
16	-143.26	-152.32	-155.18	-158.01	163.78						
17	-178.91	-179.83	-180.25	-180.70	-181.78						
18	-190.65	-212.17	-217.80	-223.08	-233.68						
19	-215.65	-240.40	-246.91	-253.46	-266.24						
20	-206.07	-207.21	-207.56	-207.97	-208.84						

teristic forms. In the case of calculations performed using the DYNWIT code the natural frequencies differ significantly with respect to the rotor rotational velocity, particularly in the first few natural frequencies. They differ even in the case when they are linked to the form of vibrations (the shape of deformed rotor) and not only the form number.

For more detailed analysis very useful are the values of damping presented in Tab. 3. If the given form of vibrations is accompanied by the large damping then the amplitude of vibrations of this form will be small and vice versa. This means that in the case of the experiment No. 1407 the strongest should be vibrations corresponding to 4^{th} and 5^{th} form, in the case of the experiment No. 1411 – respectively, to the form 5^{th} and 6^{th}, and in the case of No. 1414 – 6^{th} and 7^{th}. In Tab. 4 the frequencies for these three forms are marked by the frame. In Fig. 4 presented are the schematics of these six characteristic forms calculated using the DYNWIT program. These are spatial drawings of deformed rotor during vibrations and a projection of deformed rotor onto a horizontal plane.

In the case of the experiment No. 1407, conducted in the vicinity of the first resonance the forms 4 and 5 present the transverse bending vibrations. These correspond exactly to the forms, which are induced by the system of weights (imbalances) mounted on the rotor discs. The frequencies corresponding to these forms almost exactly coincide with the measured rotor resonance frequency in run No. 1407 (compare Tab. 2). The second increase of vibrations experimentally confirmed at the frequency of 57 Hz and best reflected by the experiment No. 1411 is not mirrored in any of the calculated self frequencies of the rotor. However, the frequency of the third region of the vibration increase, equal to 65.5 Hz, lies in the vicinity of the natural frequency corresponding to the seventh characteristic form of the rotor supported as in the experiment No. 1414 with the frequency of 62.3 Hz. In this case the frequency difference is 5% and even if that difference would be qualified as a measurement and calculation error, it is hard to believe that at such rather significant damping corresponding to this form the rotor vibration with such high amplitudes were possible, which exceed even the amplitude of the rotor vibrations in the principal resonance. The way of inducing the vibrations (with the force couple from the rotor imbalance) also is not favourable to the development of such form of vibrations, which is linked with that frequency. Additionally, the maximum of vibrations for that form is located in the middle of the rotor length, which is the location of rotor support in the bearing (see Fig. 4). On the other hand, from Fig. 3 it results that the pin trajectory in the middle bearing is rather small.

It results from the above considerations that a large amplitude of vibrations of the disc No. 1 in the vicinity of the rotor rotational velocity of 3900 rev/min (65 Hz) cannot be explained as a resonance of one of the characteristic forms of rotor with the excitation due to imbalance. The reasons must be sought in another phenomenon.

Considered must be other possible reasons of discrepancies between the results of experiments and calculations, which were outlined at the beginning of the paper. Some of these hypothetical reasons can simply be excluded, whereas

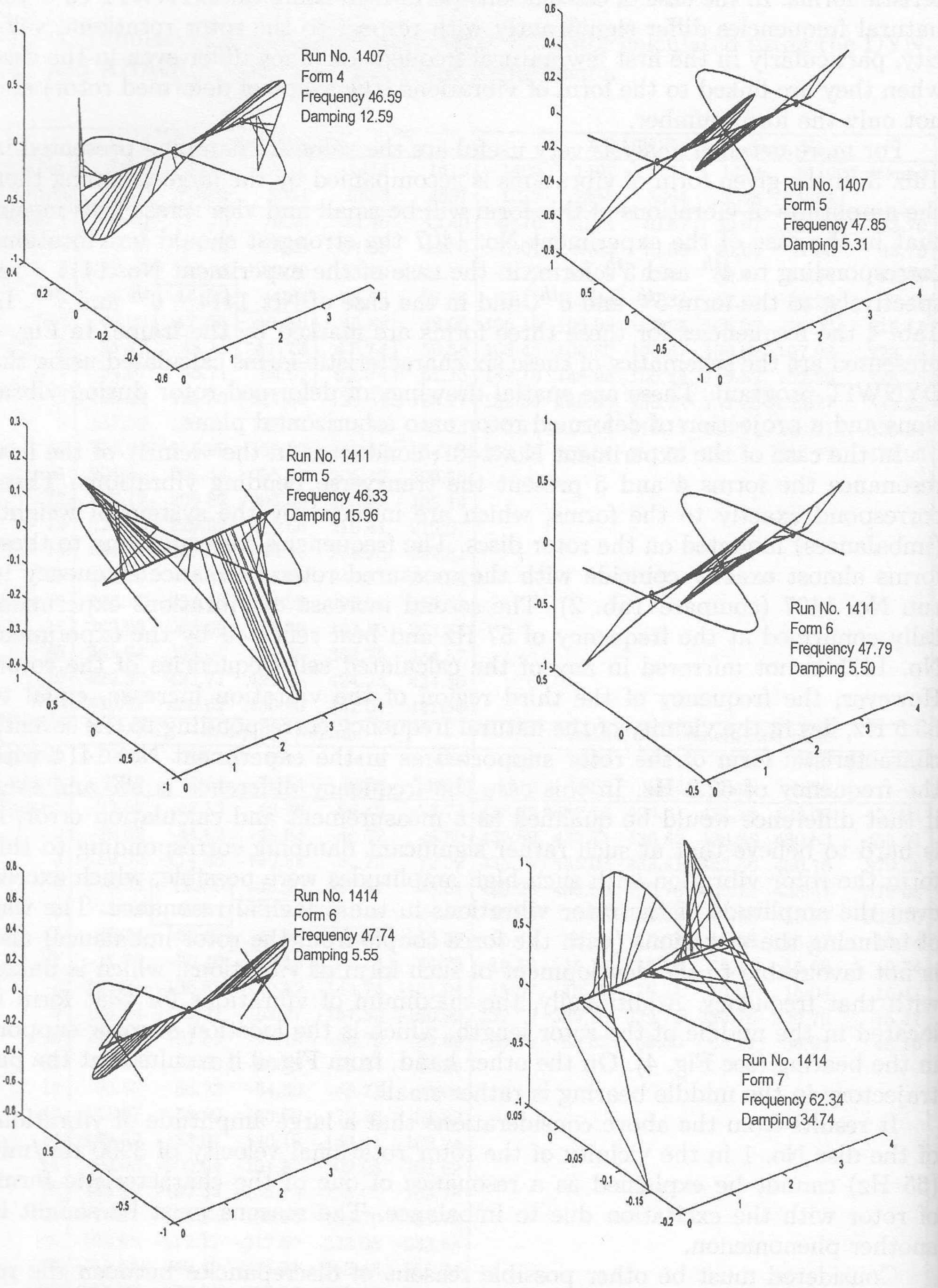

Fig. 4. Selected characteristic forms of rotor calculated using the DYNWIT code for three characteristic experiments.

some required further investigations.

Excluded could be the vibrations of measurement sensors, as it is not practically possible that all sensors would vibrate simultaneously with the same frequency, when their fixing is considered. It is also not possible that the same measurement errors would be present in various measurement series. Vibrations of the same frequency were present in all conducted measurement series.

Loosening of the discs could be also seen to be the impossible reason. If these were to lead to the explanation of the unexplained vibrations, then such vibrations would not decay with the further increase of revolution speed, but would rather intensify. For the same reasons we can reject the excitations coming from the propelling elements as well as the material heterogeneities.

It is almost not probable that the rotor vibrations would carry the vibrations of the foundation frame staying in resonance with the excitation coming from the rotor. However, in order to exclude it complimentary measurements must be performed of the vibrating frame.

Also for this purpose the additional calculations of rotor behaviour will be performed using the NLDW program. Planned calculations will take into account variability of stiffness and dumping coefficients with respect to forced vibrations frequency. At the present moment there are performed experiments on the test rig to obtain the sets of those coefficients.

6. Computer simulation of the rotor line imperfection

Earlier works showed, that the source of increased rotor vibrations could be the imperfections of the line of shafts of the rotating machinery. There can the following imperfections considered: rotor misalignment (in this case only possible in the coupling connecting the two parts of the shafts), clearance in the clutch and also the deflection of the rotor axis from the geodesic line. In order to confirm or reject such possible reasons conducted have been additional computer simulations of the rotor crack and deflections of the rotor axis from the geodesic line.

Clearances in the fixing of the coupling bushes and the shaft ends or inadequate clamping of the contacting planes of both halves of the coupling can be utilised in modelling of the rotor crack. In this case the rotor loaded with the bending moment will "open" or "close" the crack in the way dependent from the mutual location of the plane, where the crack is located and the plane, where the shaft bending moment is suspended, which in turn results from the imbalance. Calculations have been conducted for four cases of mutual location of these planes differing by the angle $\pi/2$ ($\alpha = 0$, $\pi/2$, π, $3\pi/2$) and for several extents of the cracks p taking values from 0 to 30%. Expressed in percents values of the crack p is defined here as a ratio of the depth of the crack to the shaft diameter.

In Fig. 5 presented is a graphical representation of the dependence of the amplitudes of the discs absolute vibrations and relative vibrations of bearing journals on the rotor rotational velocity of one calculated case: for $p = 30\%$ and $\alpha = \pi/2$. As could be expected the vibrations increased significantly (see Fig. 2),

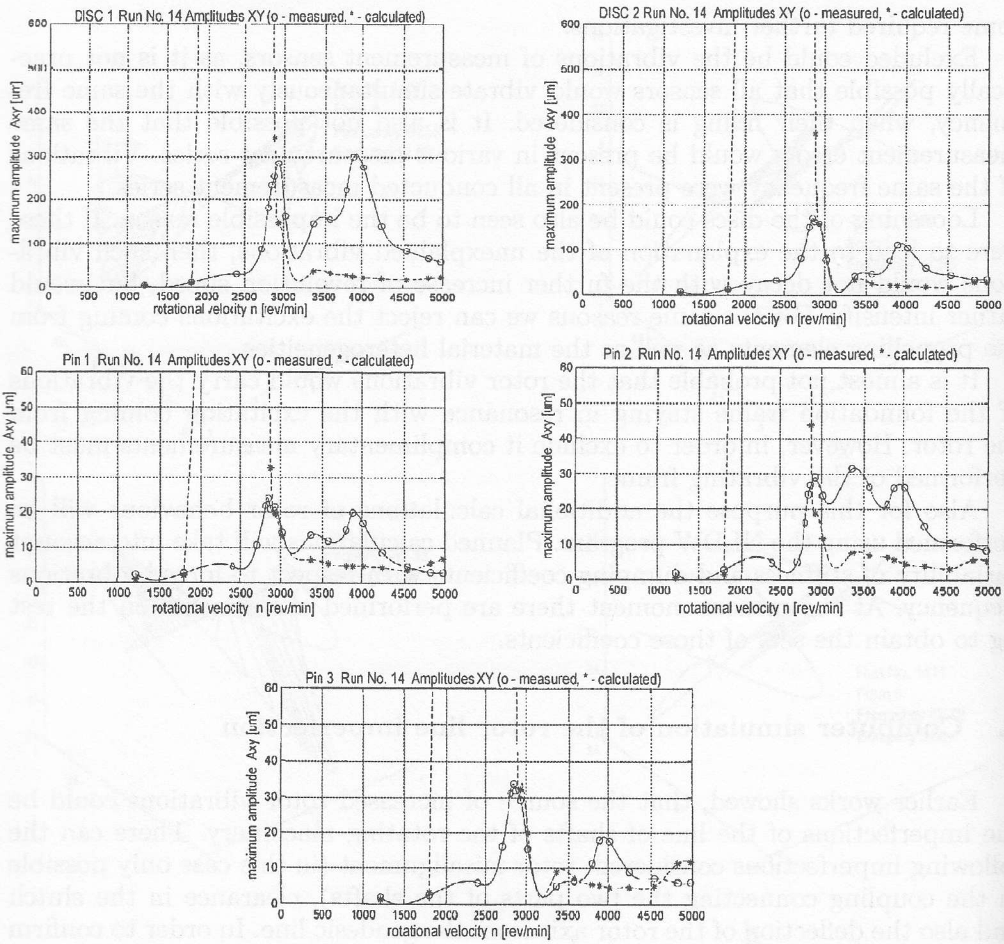

Fig. 5. Dependence of the amplitude of vibrations of particular elements of the rotor on the rotational velocity of the shaft in the case of the "rotor crack" in the coupling.

but there was not a symptom found of the increase of the amplitude of vibrations beyond the region of 1^{st} critical velocity. Not a symptom was also found in any of the other calculated cases of the crack.

Simulation of the rotor axis deflection from the geodesic line conducted has been by displacement of the middle bearing with respect to its base location in four directions: upward, downward, left and right by the increasing amounts until reaching the calculation limits of the NLDW program. At some value of the bearing displacement the program ceased calculations signalling that the minimal allowed oil film thickness has been exceeded due to the bearing overload. In practice the displacement of the one of the bearings renders increasing or decreasing of the load of this bearing and/or other bearings with respect to the direction of support displacement.

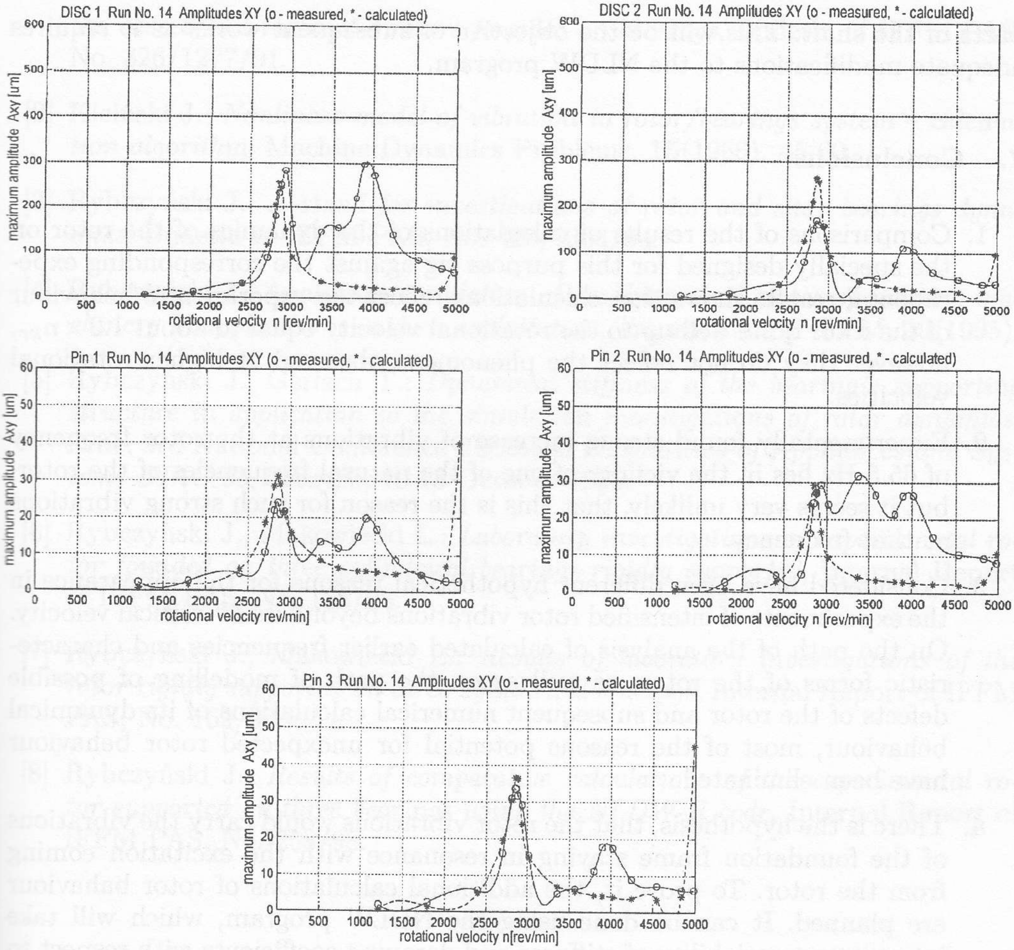

Fig. 6. Dependence of the amplitude of vibrations of particular elements of the rotor on the rotational velocity of the shaft in the case of the middle bearing displacement by 0.25 mm upward.

In Fig. 6 presented is an analogical to the Fig. 5 sample dependence of the amplitude of absolute disc vibrations and relative vibrations of the bearings journals on the rotor rotational velocity in the case of the displacement of the middle bearing by 0.25 mm upwards. Also in this case there is no increase of the amplitude of vibrations beyond the region of 1^{st} critical velocity, nor it was observed in other calculated cases.

As can be seen, neither the "rotor crack" nor the bearings displacement with respect to the geodesic line is the reason for the appearance in the experiments of additional vibrations at the rotational velocity of the rotor equal to 3364 and 3898 rev/min.

For further investigations remain the radial or angular misalignment of the shafts consisting the rotor, which is only possible in the coupling connecting two parts of the shaft. This will be the objective of subsequent works as it requires adequate modifications to the NLDW program.

7. Conclusions

1. Comparisons of the results of calculations of the dynamics of the rotor on the specially designed for this purpose rig against the corresponding experimental results shows that calculations reflect the experimental behaviour of the rotor quite well up to the rotational velocity equal to about $1.2 \times n_{kr}$, however they do not reflect the phenomena observed at higher rotational velocities.
2. Experimentally found strong increase of vibrations at the rotor frequency of 65.5 Hz lies in the vicinity of one of the natural frequencies of the rotor, but it seems very unlikely, that this is the reason for such strong vibrations at that frequency.
3. Considered have been different hypothetical reasons for the appearance in the experiments of intensified rotor vibrations beyond its 1^{st} critical velocity. On the path of the analysis of calculated earlier frequencies and characteristic forms of the rotor, as well as on the path of modelling of possible defects of the rotor and subsequent numerical calculations of its dynamical behaviour, most of the reasons potential for unexpected rotor behaviour have been eliminated.
4. There is the hypothesis, that the rotor vibrations would carry the vibrations of the foundation frame staying in resonance with the excitation coming from the rotor. To prove it, the additional calculations of rotor bahaviour are planned. It can be done using the NDLW program, which will take into account variability of stiffness and damping coefficients with respect to forced vibrations frequency.
5. Additional investigations in the form of experiments and numerical calculations are necessary in order to identify the reasons for the unexpected rotor behaviour. At present, the most probable reason remains the radial or angular misalignment of the shafts forming the rotor in the coupling connecting two parts of the shaft. This will be the topic of subsequent works.

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