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COMPARATIVE STUDIES OF REFERENCE MEASUREMENTS OF CYLINDRICAL SURFACE ROUNDNESS PROFILES OF LARGE MACHINE COMPONENTS

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Abstract

The article presents results of comparative tests performed to verify the conformity of geometric deviation measurements of a crankshaft carried out at a test bed equipped with a system of elastic support with measurements adopted as reference values. A number of simulation tests were carried out with varied shaft support conditions using the proposed measuring system. The selection criteria were established for support parameters. Meeting these criteria guarantees that shaft elastic deflections and strains are eliminated. Consequently, such strains will not affect the estimation of geometrical deviations of the measured object. The comparative evaluation measurement of roundness profiles and values of roundness deviations of main crankshaft bearing journals of a marine medium speed engine was performed using a correlation calculus. The results have revealed high conformity of both determined roundness deviation values and measured profiles compared to the reference ones.

Keywords: comparative tests, reference measurements, roundness profiles, crankshafts, correlation calculus.

1. Introduction

Basically, geometric measurements of machine components are aimed to assess the conformity of the actual part with the theoretical dimensions and shapes intended by the designer. Issues related to measurements of geometric deviations of machine components mostly refer to items of relatively small dimensions for which it is arbitrarily assumed that the impact of component strains on the total measurement error is insignificant.

Measurements of large, flexible machine components featuring sets of cylindrical surfaces such as crankshafts or camshafts, parts with high geometric complexity and high elasticity, require that specific conditions be provided. These conditions should enable the elimination of elastic strains of the measured object due to its own weight. Such strains may result from a specific method of positioning and supporting the measured object. Common methods of supporting a crankshaft where it is set up in a system of fixed V-blocks do not quite eliminate elastic strains of the shaft. Measurements of the shaft so positioned are burdened with errors of elastic strain. The errors vary in sign and value due to changes in shaft stiffness while it is rotated.

The system of shaft elastic support, developed and used at the Maritime University of Szczecin, eliminates these shortcomings and allows to measure shaft geometrical deviations (form and axis alignment deviations) correctly in terms of metrology. The idea of shaft elastic support is illustrated in Fig. 1. Figure 2, in turn, schematically shows measurement systems where the shaft elastic support is implemented.

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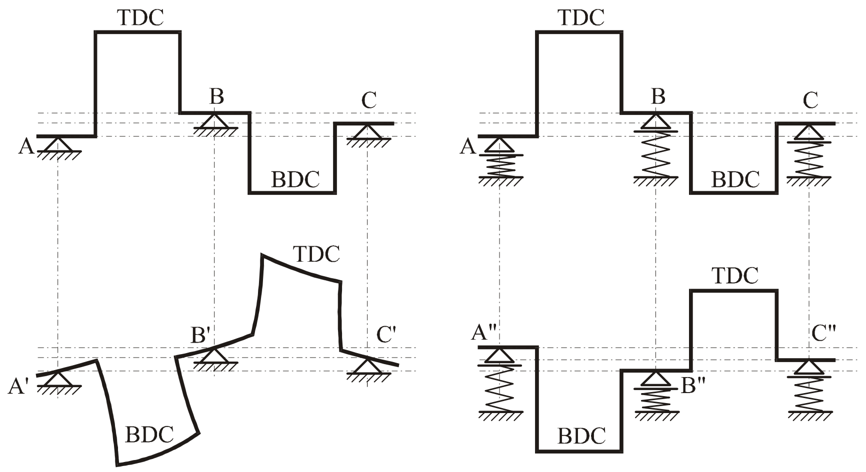

Fig. 1. The idea of an elastic support of a crankshaft; TDC – top dead center, BDC – bottom dead center.

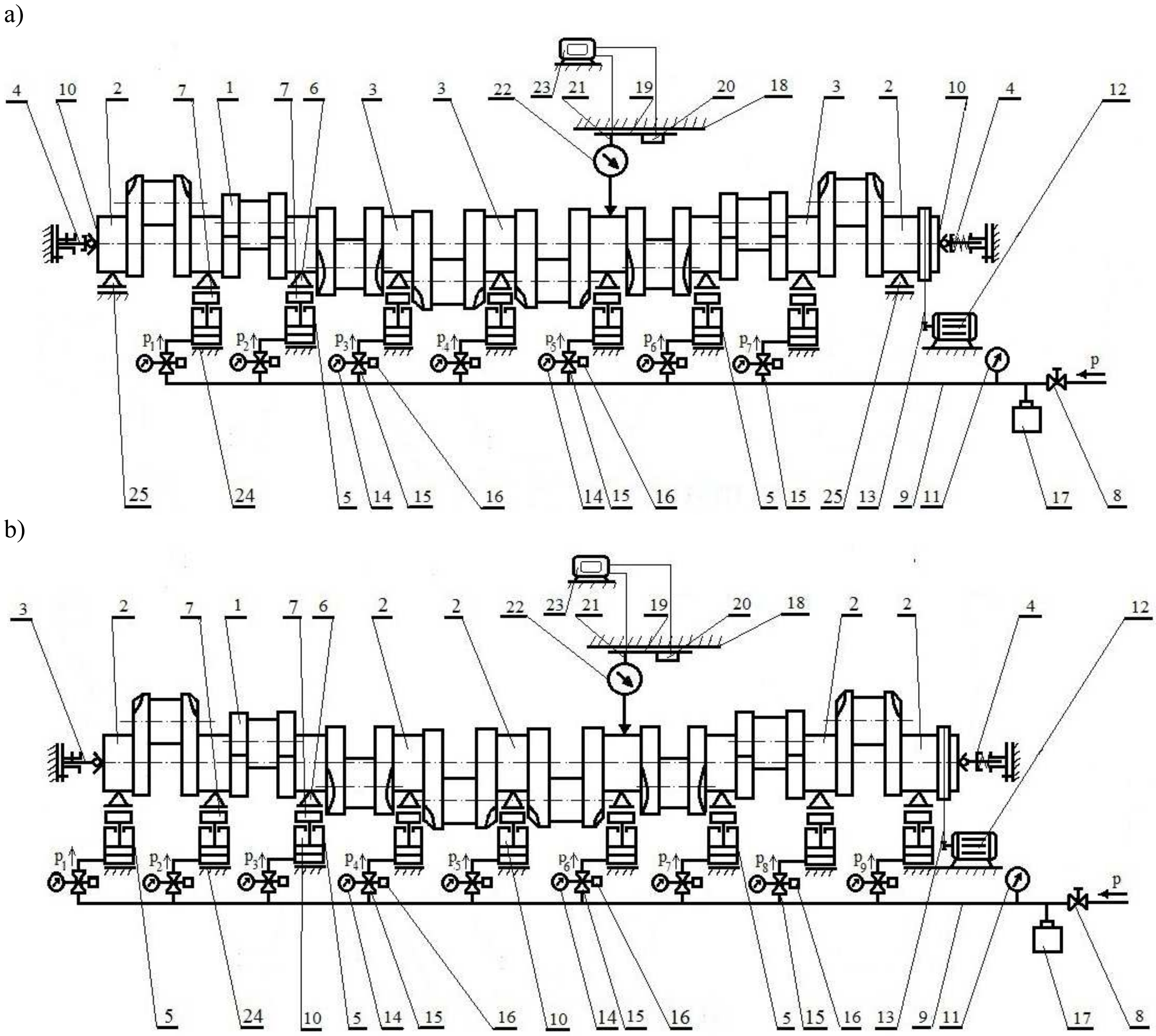

Fig. 2. A diagram of the system utilizing the concept of elastic crankshaft support with
a) the shaft set up in V-blocks, b) the shaft set into center points.

Two methods of setting up the measured object have been developed. One is used for objects without center holes and consists in placing the external cylindrical surfaces of the object in V-blocks (Fig. 2a). In the other method, the object is set into center points (Fig. 2b).

In case of V-blocks, the calculations of measurement results require a procedure of changing from the measured profile to the so called real transformed profile. The procedure is established for an object set up in two V-blocks. It should be pointed out that, as was demonstrated in [1, 2], when a measured object is positioned in V-blocks, to provide for a permanent contact of shaft journals with the V-blocks, a compromise has to be found between the values of forces applied by the lightening supports and deflections of the journals.

However, regardless of the setting up of the shaft (V-blocks or center points), elastic supports placed in the central part of the shaft act as elements compensating for possible journal deflections. These supports have specific values of lightening forces. These values are continuously controlled in the feedback circuit by precise control valves.

Proper choice of air powered cylinder pressures in elastic supports plus careful supervision of the force values applied by the supports guarantees that deflections are minimized when V-blocks are used, and eliminated completely in case of center points.

2. Selection of values and distribution of lightening forces

The existing programs for machine component strength calculations allow to model the tested object (most commonly by the finite elements method) and make strength calculations, including strains and stresses in the shaft depending on the support method. When the shaft end journals are set up in V-blocks and in its central part a number of lightening forces with looked-for values is applied, such system cannot be statically determined and the calculation of lightening forces applied is very difficult without proper software. The program used was Nastran FX 2010. The test object introduced into the program was a crankshaft of Buckau Wolf R8DV 136 engine, modeled by the finite elements method (138,303 elements). Then, having assumed the boundary conditions defining the state of deformation of the crankshaft, (deflection values at the journals), we calculated the values of lightening forces meeting the adopted constraints. Besides, a variant of shaft support was determined that satisfies the criterion of optimal support where the deflection at journals is minimum.

Research has shown that to completely eliminate shaft deflections the lightening forces applied along the shaft length through the supports have to be varied and their values adjusted depending on the angle of shaft rotation. Some results of calculated values of lightening forces and corresponding angle of shaft turn are given in Table 1.

Table 1. Values of lightening forces at which no deflections occur at the shaft journals in relation to the shaft angle of rotation.

Rotation angle	**Journal number-forces, N**									
	1	2	3	4	5	6	7	8	9	10
0°	780.064	907.144	831.740	1108.730	798.994	1015.020	804.985	1087.280	935.406	574.830
90°	863.160	712.929	1079.040	839.009	1050.540	779.562	1054.270	835.302	1102.380	528.102
180°	723.466	978.164	809.592	1118.820	793.023	1020.910	803.532	1087.120	933.822	575.833
270°	851.830	739.756	1054.390	840.085	1083.040	733.705	1079.510	841.511	1085.690	534.062

If the values of applied forces are the same, or the support arrangement is changed, deflections impossible to eliminate will appear.

It has been theoretically assumed that the force values and the distribution of lightening supports applying those forces are strictly specified. It was, therefore, decided to apply the lightening forces to main journals of the shaft at their half lengths. The distribution of forces for one of the angular positions of the shaft, assuming the supports will be placed at half lengths of the journals, are shown in Fig. 3.

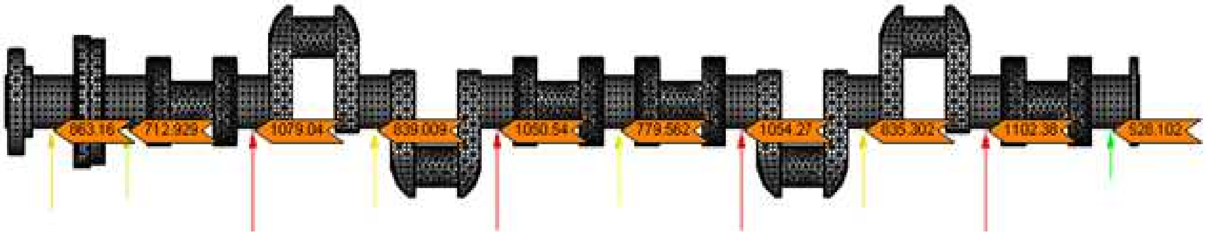

Fig. 3. Distribution of forces for a specific angular position of the shaft,
with the supports placed at half lengths of the journals.

3. Verification of the crankshaft elastic support system functioning and comparison of measured roundness profiles using correlation calculus

The system of crankshaft elastic support was verified at the existing test bed equipped with a set of load-lightening supports (Fig. 4).

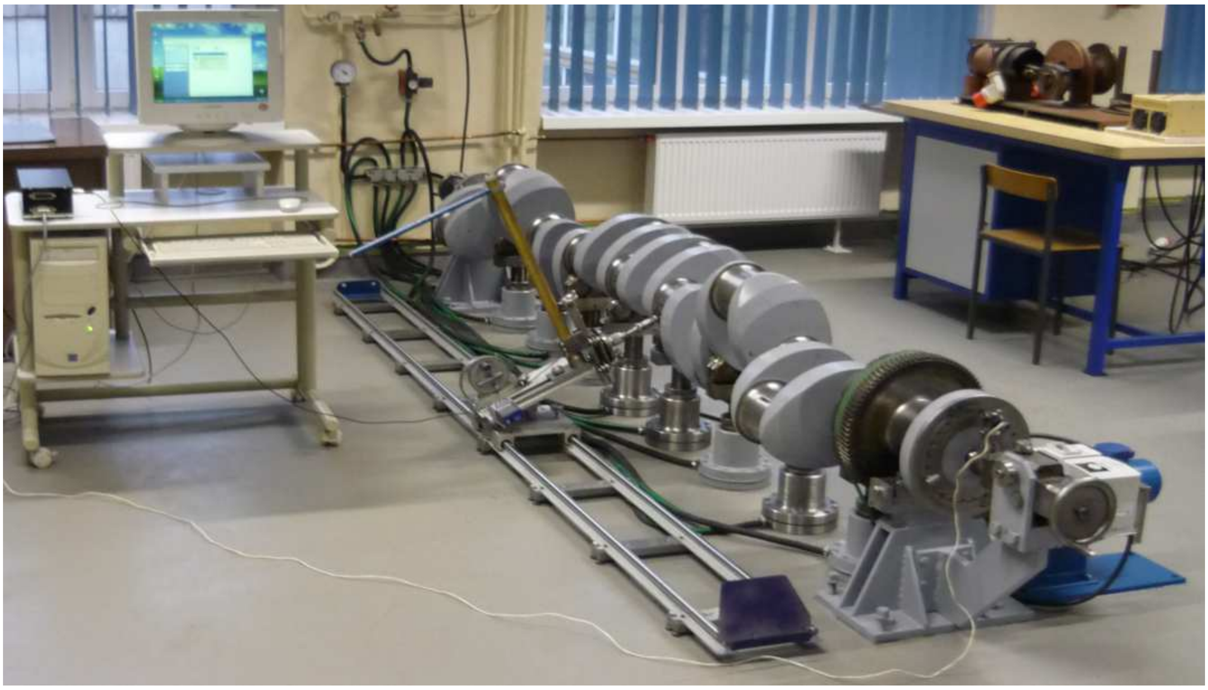

Fig. 4. A test bed for geometrical deviation measurements of crankshafts,
equipped with a system of shaft elastic support.

The tests were carried out for the end journals of the shaft put in V-blocks. The research included measurements of roundness deviations and profiles of the main journals for diversified shaft support values.

For selected support variants, the roundness profiles of main journals were measured at specific cross-sections along the shaft length. The profiles were in all cases compared to the profiles from a reference measurement system. Reference measurements were performed by the proven system with a MUK 25-600 head and SAJD software, developed at the Department of Manufacturing Processes and Measurements, Kielce University of Technology. The system is used for measurements of roundness profiles by the reference method. Profile measurements based on the model system were not dependent on shaft support conditions, because the MUK 25-600 head is positioned directly on the surface being measured.

Comparative assessment of roundness profile measurements was done using correlation calculus. The measure adopted for comparisons of the corresponding roundness profiles, previously filtered for harmonics range $n = 2 \div 15$, reference profile $r_1(\varphi)$ and the examined one $r_2(\varphi)$, was the value of conformity coefficient found by using a standardized intercorrelation function.

The standard intercorrelation function was defined as follows [3, 4, 5, 6]:

$$\rho(\gamma) = \frac{2\int_0^{2\pi} r_1(\varphi) r_2(\varphi+\gamma)\,\mathrm{d}\varphi}{\int_0^{2\pi} r_1(\varphi)^2\,\mathrm{d}\varphi + \int_0^{2\pi} r_2(\varphi)^2\,\mathrm{d}\varphi}. \tag{1}$$

The function was standardized so that:

$$-1 \le \rho(\gamma) \le 1. \tag{2}$$

It can be proved that $\rho(\gamma^*) = 1$ only when for a certain phase shift γ^*:

$$r_1(\varphi) = r_2(\varphi + \gamma^*). \tag{3}$$

In practice, due to the limited accuracy of measuring instruments generally the inequality $\rho(\gamma) < 1$ occurs for each γ. In this case we assume that the value γ^* for which the function $\rho(\gamma)$ assumes a maximum corresponds to the phase shift between the compared profiles, while the value of intercorrelation function $\rho(\gamma^*)$ for the determined phase shift γ^* can be regarded as a coefficient of conformity between the compared profiles.

In the proposed comparative procedure, once the value of angle γ^* is determined, corresponding to the phase shift between the examined profiles, we can visually assess the two profiles superimposed on one chart in one coordinate, polar or Cartesian system.

As a comparative criterion we adopted the values of roundness deviations and the roundness profiles of the journals. Table 2 presents the calculated values of roundness deviations and intercorrelation coefficients for the compared profiles.

Table 2. Values of roundness deviations of main bearing journals of the measured crankshaft, measured by the examined system Δ_z and reference system Δ_w, and values of intercorrelation coefficients for the compared profiles ρ.

Journal number	**Roundness deviation Δ_z µm**	**Roundness deviation Δ_w µm**	**Intercorrelation coefficient ρ**
1	26.100	23.341	0.8989
2	31.158	30.241	0.8754
3	31.394	29.158	0.9056
4	56.053	54.207	0.9165
5	30.098	28.419	0.9126
6	41.145	39.463	0.8968
7	43.671	42.038	0.9399
8	24.116	24.154	0.8893
9	35.774	32.651	0.8695
10	43.418	44.272	0.9106

The test results have shown that the values of forces applied by the supports were correct and satisfied the adopted support criteria. These criteria corresponded to the optimum support variant and ensured minimum deflections at the journals and permanent contact of main end journals with the V-blocks. The value of intercorrelation coefficient for measured journals determined for this variant ranged from 0.8695 to 0.9399, which according to J.P. Guilford's [7, 8] assessment scale of correlation indicates high or very high correlation between the compared profiles. According to this assessment scale, the degree of interrelation between the examined properties is significant or very high.

Another support variant or a change in support pressure, comparing to the optimum variant, results in a substantial increase in deflections at the journals, which significantly change the measured profile in comparison with the profile obtained from reference measurements. Consequently, the value of intercorrelation coefficient will decrease.

Figure 5 shows the distribution of deflections at the journals for the optimum support. Figure 6 shows, for this case of support, superimposed with phase shift γ^* roundness profiles (measured and reference) for journal No. 4, presented in the polar and Cartesian systems, respectively. The determined value of intercorrelation coefficient in this case (journal No. 4) is $\rho(\gamma^*) = 0.9165$.

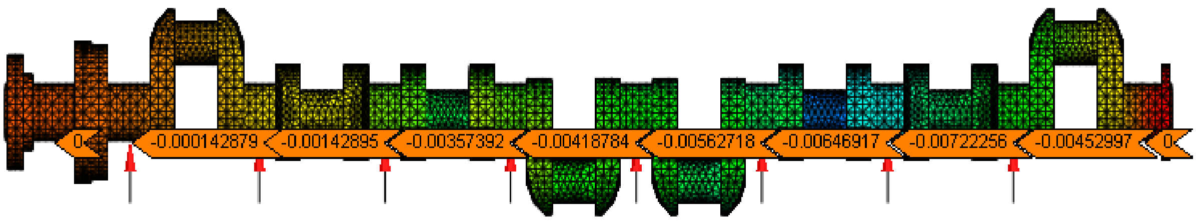

Fig. 5. Values of deflections at the journals for the optimum shaft support variant.

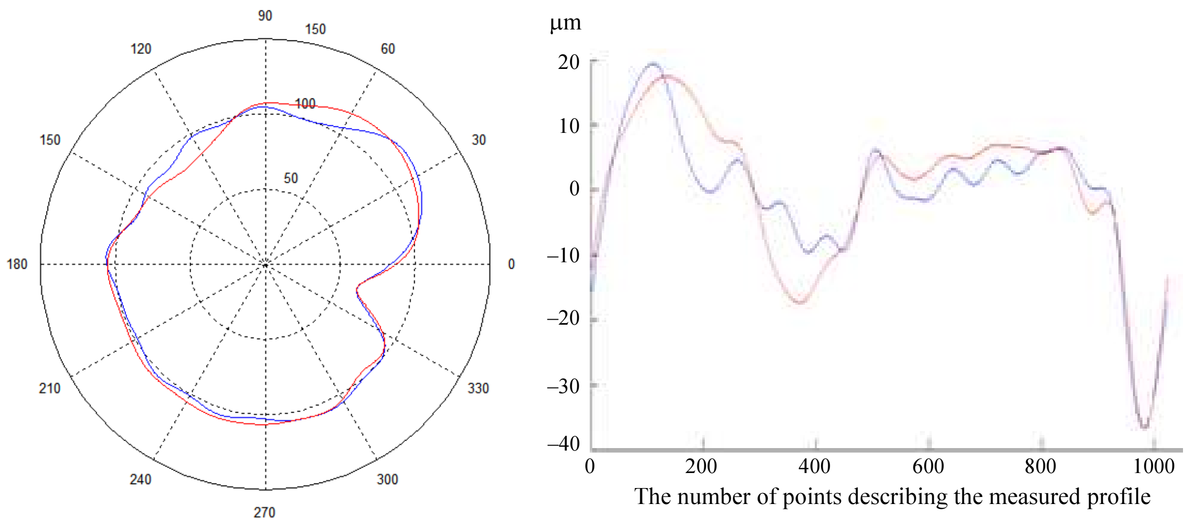

Fig. 6. The measured (blue) and reference (red) profiles of journal No. 4 in the polar and Cartesian systems for the optimum shaft support variant.

For comparison, Fig. 7 shows the distribution of deflections at the journals for the support variant other than the optimum one: the supports of journals No. 3 and 8 were removed. The determined value of the intercorrelation coefficient in this case for journal No. 4 is $\rho(\gamma^*) = 0.7987$. It is visualized in Fig. 8 by the measured and reference roundness profiles, superimposed with the phase shift γ^*, shown in the polar and Cartesian systems.

It is known that any roundness profile can be represented as a sum of Fourier trigonometric series terms, i.e. a finite cosine or sine transform. Therefore, any roundness profile can be represented as a discrete amplitude spectrum by determining the amplitudes and phase shifts of each harmonic. Such analysis allows to evaluate the influence of individual harmonics on the form of the measured profile.

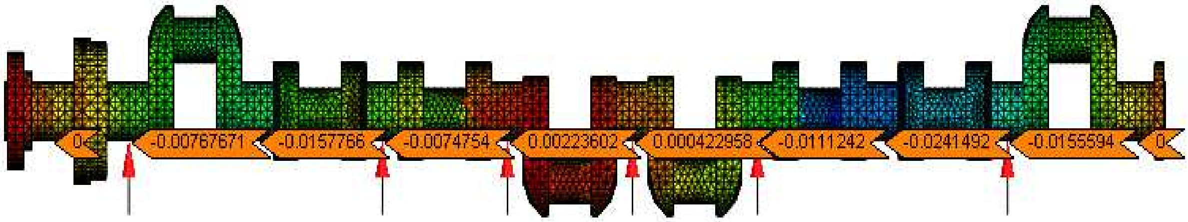

Fig. 7. Deflection values at journals for the shaft support other than the optimum one.

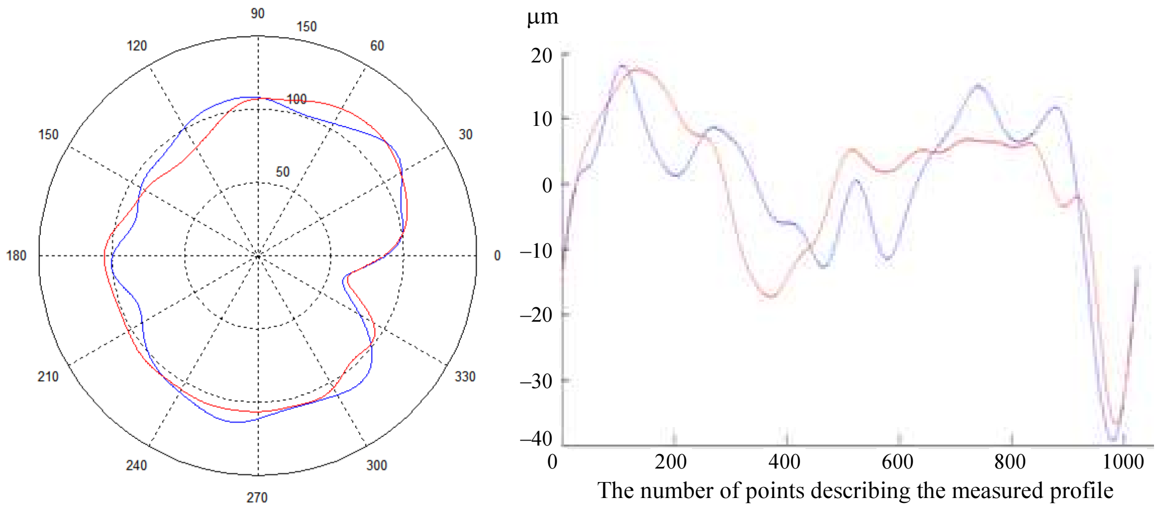

Fig. 8. The measured (blue) and reference (red) profiles of journal No. 4 in the polar and Cartesian systems for the shaft support other than the optimum one.

The harmonic components of the measured profiles were compared by using the principles of reciprocal correlation calculus. Pearson's linear correlation coefficient was a measure of correlation between the compared harmonics:

$$r = \frac{\sum_{i=1}^{n}\left(Cx_{ni} - \bar{C}x_n\right)\left(Cy_{ni} - \bar{C}y_n\right)}{\sqrt{\sum_{i=1}^{n}\left(Cx_{ni} - \bar{C}x_n\right)^2 \sum_{i=1}^{n}\left(Cy_{ni} - \bar{C}y_n\right)^2}}, \tag{4}$$

where: Cx_{ni} – value of harmonic amplitude of a measured crankshaft i-th journal profile, Cy_{ni} – value of harmonic amplitude of the reference i-th journal profile, $\bar{C}x_n$ – mean value of harmonic amplitude of a measured profile, $\bar{C}y_n$ – mean value of harmonic amplitude of the reference profile.

Correlation calculations were verified using a significance test of the correlation coefficient at the level $\alpha = 0.05$ by assuming the hypothesis: no correlation – $H_0 : r = 0$ relative to the alternative hypothesis: correlation exists – $H_1 : r \neq 0$, using for this purpose the statistics:

$$t = \frac{r}{\sqrt{1-r^2}}\sqrt{n-2}, \tag{5}$$

where: r – estimated correlation coefficient, n – sample size.

Calculated Pearson's coefficients defining the degree of correlation between the values of amplitudes and phase shifts of each harmonic of the compared crankshaft roundness profiles are given in Tables 3 and 4.

Table 3. Pearson's coefficient values for harmonics amplitudes of compared roundness profiles.

Harm.	**2.**	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.	13.	14.	15.
2.	**0.9601**	–0.0922	0.1846	0.1657	0.1925	0.1561	–0.0364	–0.0369	–0.1220	–0.1260	–0.1978	–0.2793	0.0690	–0.0419
3.	–0.0384	**0.9732**	0.0776	0.5416	0.0789	0.1659	0.5993	0.3450	0.5040	0.2800	0.3382	0.0154	–0.0313	0.0741
4.	0.1012	0.7095	**0.7359**	0.4757	0.1967	0.4578	0.4049	0.6359	0.4918	0.1048	0.2453	–0.2139	–0.1602	–0.1433
5.	0.1258	0.6390	–0.0742	**0.9734**	0.1756	0.3116	0.7176	0.2696	0.4513	0.0587	0.1122	–0.3535	0.2661	0.2492
6.	0.0747	0.1609	0.2324	0.4007	**0.8675**	0.6865	0.6614	0.3292	0.3207	0.3185	0.1020	–0.3286	–0.1185	–0.0702
7.	0.0550	0.0872	0.2033	0.2167	0.6648	**0.9136**	0.3000	0.5801	0.2591	0.1474	0.1263	–0.3083	–0.2128	–0.0245
8.	0.0124	0.1197	0.0333	0.2813	0.0057	0.0766	**0.7333**	0.2729	0.3248	–0.0147	0.0204	–0.1492	0.1191	–0.0369
9.	0.0115	0.0305	0.0212	0.0841	–0.0056	0.5569	0.0708	**0.6074**	0.2250	–0.1668	0.0223	–0.1444	0.0365	0.0703
10.	–0.0203	0.0577	0.0418	–0.0086	–0.2824	–0.1626	0.0923	0.5575	**0.6962**	–0.0448	0.2387	0.0562	–0.0014	–0.2438
11.	–0.0242	0.0678	0.0045	–0.0005	0.1779	0.0344	0.2870	0.1189	0.4733	**0.9305**	0.7286	0.4852	–0.4163	0.0306
12.	–0.0077	0.0277	0.0035	0.0028	0.1428	0.1948	0.1154	0.1169	0.2268	0.8350	**0.8489**	0.4255	–0.5390	0.2556
13.	–0.0077	–0.0061	–0.0181	–0.0264	0.0761	0.0459	0.0185	–0.0654	–0.0500	0.3725	0.3405	**0.7074**	–0.5792	0.4485
14.	0.0103	–0.0141	–0.0284	0.0298	–0.0018	0.0349	–0.0306	–0.0919	–0.1629	–0.2397	–0.2833	–0.4617	**0.5175**	0.5310
15.	0.0051	0.0093	0.0014	0.0231	0.0793	0.0846	0.0448	0.0027	0.0157	0.1609	0.1317	0.0373	–0.2697	**0.5692**

Table 4. Pearson's coefficient values for harmonics phase shifts of compared roundness profiles.

Harm.	**2.**	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.	13.	14.	15.
2.	**0.9918**	0.5726	–0.2505	0.2948	0.2002	0.2148	0.2148	–0.4048	–0.2059	–0.1431	0.0971	–0.2551	–0.3485	0.0456
3.	0.4593	**0.9052**	0.5562	0.4321	0.5758	0.7948	0.1428	0.0912	0.3592	0.0779	0.0488	–0.3178	0.1231	0.5140
4.	0.0996	0.4184	**0.9351**	0.5324	0.6485	0.6708	–0.0372	0.2278	0.7318	0.2143	–0.0907	–0.1481	0.3065	0.2708
5.	0.2402	0.1830	0.3210	**0.8021**	0.7857	0.0819	0.2871	0.1272	0.1095	0.0907	0.0066	–0.0090	0.5958	0.0579
6.	–0.1184	0.3453	0.8076	0.7564	**0.9147**	0.3810	0.1589	0.4613	0.5692	0.3331	–0.1405	–0.0746	0.7062	0.2308
7.	0.4001	0.8482	0.6082	0.3328	0.5603	**0.8628**	0.1617	0.1668	0.4007	0.0892	0.0841	–0.2972	0.0886	0.6312
8.	0.3981	0.6632	0.0902	0.5348	0.5518	0.2770	**0.9139**	0.5461	–0.2394	0.5667	–0.0501	–0.4166	0.3948	0.5946
9.	–0.4742	–0.0262	0.5641	0.2549	0.2604	0.4158	0.3150	**0.8365**	0.4413	0.7322	–0.4621	–0.3578	0.3545	0.5591
10.	0.3391	0.3180	0.5854	0.4580	0.4590	0.6476	–0.0585	–0.0376	**0.8745**	0.2365	–0.1933	–0.3284	–0.2432	0.2196
11.	0.1722	0.2679	0.0793	0.3387	0.0970	0.4715	0.6234	0.3736	0.2146	**0.9113**	–0.3297	–0.6467	–0.1646	0.4414
12.	–0.0390	–0.0554	–0.0173	–0.1111	–0.0065	–0.1661	–0.1178	–0.0838	–0.0973	–0.2624	**0.9576**	0.4946	0.0763	–0.1594
13.	–0.1941	–0.5110	0.0123	–0.0264	–0.0348	–0.5042	–0.3188	–0.2190	–0.0246	–0.3254	0.1243	**0.7537**	0.1721	–0.8372
14.	–0.3982	–0.3319	–0.4509	–0.2741	–0.2375	–0.7672	0.1499	0.2285	–0.7981	–0.1649	0.2774	0.4618	**0.4880**	–0.2300
15.	0.0531	0.4243	0.3988	0.0446	0.1801	0.8340	0.2641	0.3861	0.3910	0.4159	–0.2184	–0.5967	–0.1518	**0.8949**

From the harmonics comparison viewpoint, essential correlation coefficient values in Tables 3 and 4 are those corresponding to diagonal elements of the correlation matrix. These elements correspond to the correlation coefficients between amplitudes and phase shifts with the same harmonic numbers.

The calculations have shown that in most cases there is high or very high correlation between amplitudes of relevant harmonics (particularly the dominating amplitudes and those crucial for the profile shape, i.e. harmonics in the range $n = 2 \div 10$). For some harmonics only, $n = 14$ and $n = 15$ the correlation is moderate. However, we may assume that the impact of these harmonics on the profile shape is slight. The determined coefficient values also show which component harmonics and to what extent affect the difference in the shape of compared profiles. This is confirmed by charts of the amplitude spectra. One selected case is presented in Fig. 9.

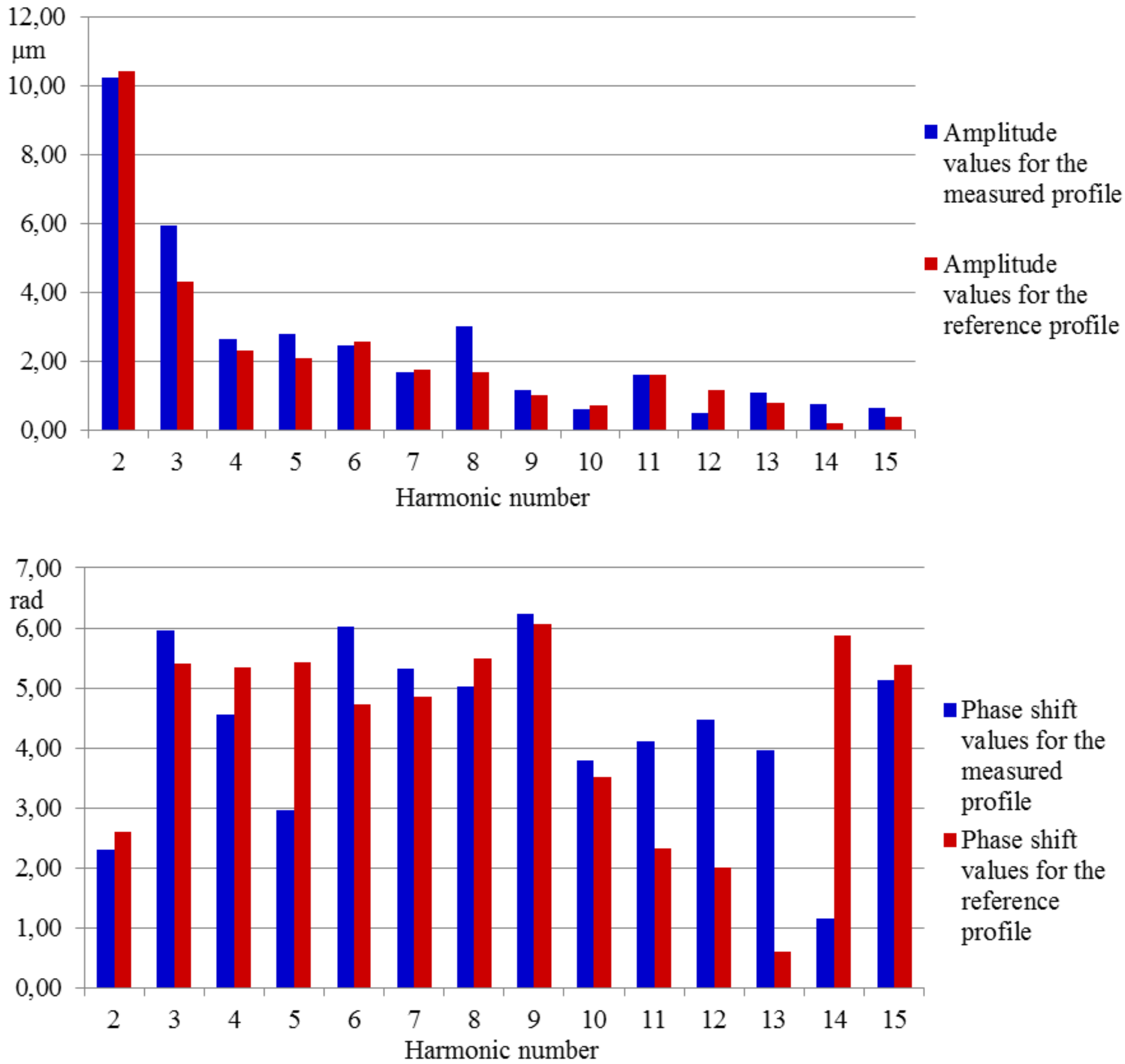

Fig. 9. Amplitude spectra charts for journal No. 2, including harmonics $n = 2 \div 15$, measurements using the shaft elastic supports system and the reference system.

5. Conclusions

The results of latest and previous tests [1, 9] allow to conclude that the system of shaft elastic support works effectively and can be used for the elimination of elastic strains of a shaft during measurements of its geometric deviation. The correlation calculations have shown that when the optimum support variant is implemented, the obtained conformity of results is high in relation to values of roundness deviations as well as the measured form of overall profile.

If we take into account the measurement conditions (the shaft was set up in V-blocks and based in a system of elastic supports in the central part) and, additionally, bear in mind that the measured roundness profiles are highly irregular, the obtained results allow to expect that the proposed system for measuring geometric deviation of objects such as a crankshaft will give results of high conformity with those obtained from reference measurement methods. It practically means that the proposed system can be used for measurements of form deviations as well as axis position of cylindrical surfaces of straight shafts and crankshafts.

Acknowledgements

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