ANALYSIS OF ROTATION ACCURACY OF HIGH-SPEED SPINDLES WITH GAS-LUBRICATED BEARINGS

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The article presents a description of a test bench for analysis of rotational accuracy of a shaft supported by gas static bearings with partly porous shell walls and by those with orifice restrictors, along with the results of an experimental investigation.

High-speed machining is one of the leading directions in the development of modern manufacturing engineering. Introducing it in the metal-working industry allows to increase the productivity of labour along with the simultaneous improvement of machining accuracy and machined surface quality.

An important factor for successful realization of high-speed machining is the type of bearings used in spindle units (SU) of metal-working machines.

The work of spindle units supported by rolling bearings is accompanied, in particular, by an unstable spindle motion and thermal misalignment of bearing assemblies. Using hydrostatic bearings in high-speed spindle units leads to a spindle speed limitation (because of friction loss) and makes the design of the bearing assembly more complicated. Spindles supported by electromagnetic-bearings are not widely used so far because of their complication and the high cost of spindles and electronic control systems. Spindle units supported by gas-lubricated bearings are free of the above-listed shortcomings.

Gas bearings are reliable in operation both at high and low temperatures and humidity, the usage of them excludes pollution of the environment, reduces noise and vibration levels. Bearings of this kind are practically devoid of wear, that is why the high indices of spindle rotation accuracy remain invariable practically throughout the whole service life of machines.

Gas bearings have also some disadvantages which are in a comparatively low stiffness, bearing and damping capacities of the lubricant film. That is why bearings of this kind are used in light-loaded spindle units where the dynamic loads are light and the static loads are regulated.

The different problems of development and investigation of high-speed spindles with gas-lubricated bearings have been covered in a number of works [1–3] and others. In all presented designs of spindle units gas bearings with orifice flow restrictors were used.

At the same time, the analysis of externally pressurized gas bearings shows that it is precisely partly porous gas static bearings [4] that excel in the best operating characteristics (such as stiffness and bearing capacity of lubricant film). One of the designs of this kind of bearings – a double-row bearing with porous cylindrical inserts uniformly arranged in a pressurizing row – is shown in Figure 1.

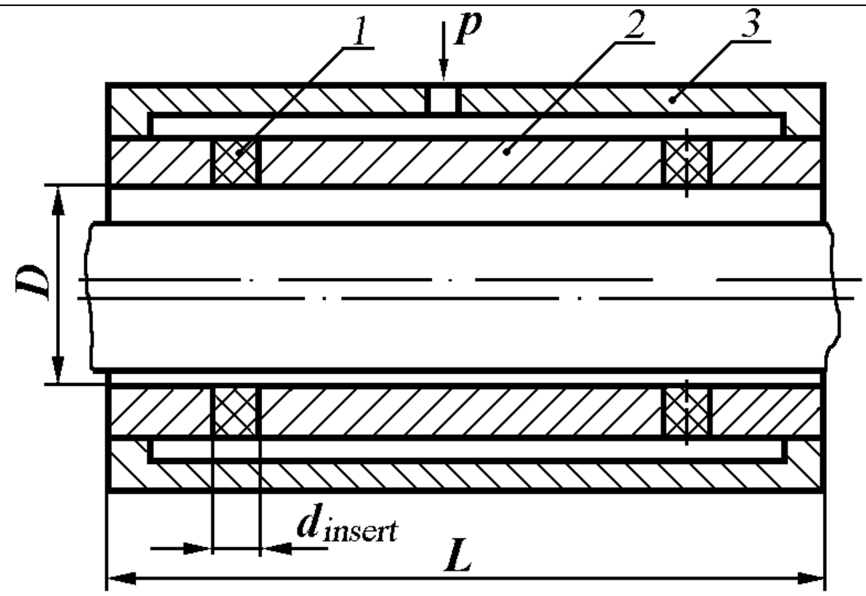

Figure 1. Partly porous gas static bearing:
1 – porous cylindrical insert, 2 – gas-tight bearing shell, 3 – bearing body

The work under review supplements the investigations [4] with experimental data on rotation accuracy of spindles supported by partly porous bearings and spindles supported by gas static bearings with orifice flow restrictors traditionally used in the metal-working-industry-used designs of high-speed spindle units.

The complex of experiments was carried out on a test bench which is represented schematically in Figure 2.

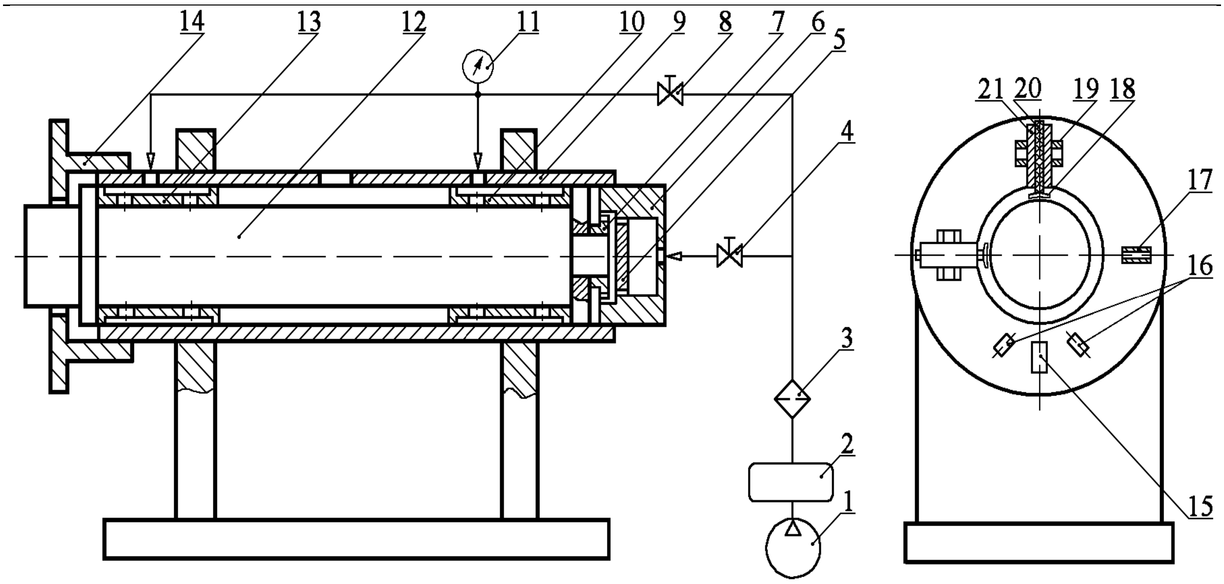

Figure 2. Diagram of the experimental test bench for the investigation of shaft rotation accuracy

The main structural members of the test bench are shaft ***12***, front ***13*** and rear ***10*** gas static bearings mounted in body ***9***, turbine wheel ***7***, nozzle diaphragm ***5*** secured in casing ***6***, shield ***14***.

The shaft was made of steel 30ХГСА (grade in accordance with GOST), its diameter was 50 mm and the length was 270 mm. The shaft surface was chromium-plated to a depth of 1 mm and hardened to the *HRC* 60-65 hardness. Once hardened, the shaft surface was finished to Class 10 with the surface finish $R_a = 0.16$ µm.

The double-row gas static bearings with porous inserts and those with supply holes were of the same geometrical dimensions (60 mm in length and 50 mm in diameter) and had an average radial clearance of 25 µm. The absolute gas boost pressure in the lubricant film was 0.3 MPa, which ensured equal values of the bearing supply parameter $\overline{m} = 7.2$. The distance between the centers of bearings was 170 mm. In each pressurizing row of the partly porous bearings there were six cylindrical porous inserts 8 mm in diameter. The air pressurization in the clearances of bearings with orifice flow restrictors was performed through two double rows of feeders. In each row there were 16 holes 0.5 µm in diameter.

The dynamic state of the shaft was determined by two capacitive transducers ***20*** with an area of 3 cm^2 each, placed adjacent to the shaft cantilever, one of them being mounted in a vertical plane and the other in a horizontal plane.

The shield was secured by screw joints at the butt-end of casing ***9*** in such a way that cover ***18*** of one transducer was arranged in a strictly vertical plane and that of the other one in a horizontal plane. The exact position of each cover with reference to the shaft center was achieved by moving slider ***21*** along guide ***19***.

The transducers working surfaces were lapped to the shaft by a fine-grain diamond grinding emery paper. The lapping was carried out at a shaft speed of 360 min^{-1}. The emery paper was of about the same thickness that the working clearance of the transducer was (0.16…0.18 mm). To avoid an influence of the shaft journal errors (under the transducers) on the results of the experiment and for reliable work of optocouple ***16***, the shaft journal has been previously lapped. The transducers position was checked by the УС 1-300 builder’s level arranged at the guide of the vertically mounted transducer.

The sliders` guides and racks of micrometric indicators ***15*** and ***17*** are an integrated part of the shield. Their axes of symmetry form a right angle between each other. The calibration test of recorders` readings was executed with dial indicators graduated in 1µm. The results of the calibration test indicated amplitude linearity of the capacitive transducers.

The experimental test bench works as follows. The compressed air from compressor ***1*** passes through receiver ***2*** and filter ***3*** and arrives at the front and rear gas bearings of the shaft at a time; valve ***8*** adjusts air boost pressure in the lubricant film of the bearing. This pressure is indicated by standard pressure gauge ***11.*** When the shaft “floats up”, valve ***4*** opens and the compressed air is delivered to the turbine stage. The shaft rotational speed was varied by flow action through the change of the rate of valve ***4*** opening.

The complex of experiments was carried out with the use of an automated investigating system which allows to solve the following tasks: to determine shaft speed, to measure rotating shaft motion in a lubricant film of bearings, and to build a shaft axis mechanical trajectory. The system was created on the base of a personal computer.

Electric signals from the transducer unit were fed to the original interface board for the ISA bus of the computer. The assembly program entered and processed measuring signals, stored results and displayed them in digital and graphic forms.

For shaft speed transducers we used a transistor infrared optocouple with open optical channel. The CQW58A-1 infrared diode and the Philips OP 500 phototransistor were arranged at an angle close to the shaft surface so that the light reflected from the shaft could hit the phototransistor. A black mark was made anywhere on the shaft to absorb the light flow. The interruption of the reflected light beam by the black mark during shaft rotating was used for measuring shaft speed and as a reference for determining shaft orientation from the angle of rotation.

The optocouple digital output signal came to one of three frequency inputs of the interface board. Through a program-controlled digital multiplexer the transducer signals were fed to the input of the impulse frequency meter mounted on the interface board. The frequency meter operating conditions were program-tuned, which provided an adequate accuracy of measurements of an input signal frequency in a wide range.

The measurement results were recorded in a file and then displayed on a monitor. The device-control and result-processing program set a graphic display mode. Orthogonal axes with scale marks were plotted on the monitor. The position of static shaft axis has been taken as the origin of coordinates. Thereupon reference points of the shaft motion in two coordinates were marked. The points were built at 5 degree intervals of the shaft rotation angle. The position of points between the measured points was established by interpolation with the use of Lagrange polynomial built on the data of four reference points nearest to each other. The shaft axis mechanical trajectory was built by joining adjacent points with segments.

The work of the horizontal and vertical shaft mechanical motion capacitive transducers was based on the principle described in work [4]. But there was a difference in using them which was in the way of processing signals coming from these transducers: the output signal was recorded only from one inverter input and, besides, an informational parameter in investigating the shaft rotation accuracy was not the relative rectangular pulse duration but the rectangular pulse frequency.

The mechanical motion transducers outputs were connected to two frequency inputs of the interface board. The frequency of signals coming from the outputs of the mechanical motion transducers was measured by the frequency meter. A transducer to be connected to the frequency meter was selected by a digital multiplexer.

Direct interrogation of the shaft speed transducer, without using the frequency meter, allowed the program to record the moment of passing through the black mark and to relate the shaft axis motion values being measured to this moment.

During the experiment the shaft rotational speed was changed from 12700 $\min^{-1}$ to 25400 $\min^{-1}$ which was appropriate to the variation of shaft specific speed $d' \, n$ = (0.63…1.27) $\cdot 10^6$ mm/min.

The synchronous vortex paths of the shaft operated in gas static bearings with porous inserts (1) and with supply holes (2) at a specific speed of 1.27 $\cdot 10^6$ mm/min are presented in Figure 3.

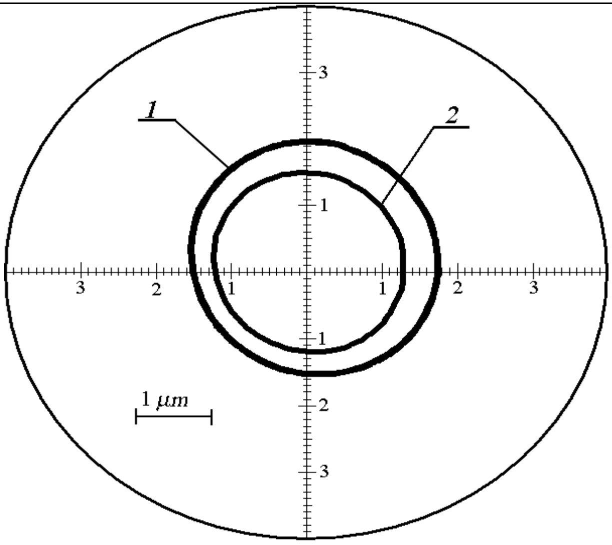

Figure 3. Synchronous vortex paths of the shaft in bearings with supply holes (1) and with porous inserts (2)

The qualitative analysis of the synchronous vortex paths discloses their elliptic shapes which can be explained by variable stiffness of the lubricant film around the bearings. The path curves are smooth and practically not blurred, that is, the shaft axis moves in a constant trajectory taking up stable state in the bearings. The quantitative estimation of the experiment results shows a distinctly lesser radial run-out of the shaft supported in bearings with porous inserts.

The radial run-out considerably decreases as the shaft specific speed is reduced (Figure 4). In this case, the synchronous vortex path tends to assume the form of circle.

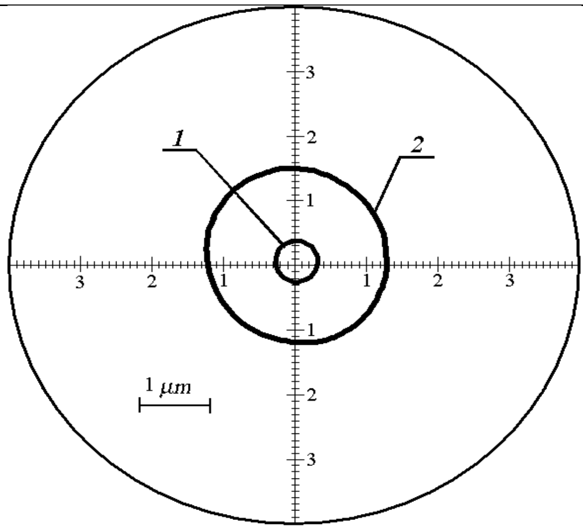

Figure 4. Synchronous vortex path of the shaft in bearings with porous inserts at a specific speed of 0.63· 10^6 mm/min (1) and 1.27· 10^6 mm/min (2)

The synchronous vortex paths presented in Figures 3 and 4 were registered in 300 revolutions of the shaft. In the course of the experiment the shaft trend, that is to say, a variation in time of an expectation value of the set of paths [5], has not been noted, which indicates that the bearing heat release was not essential.

The results of the performed complex of investigations into rotation accuracy of the shaft supported by gas static bearings with porous inserts and of the shaft supported in bearings with supply holes are shown in Figure 5.

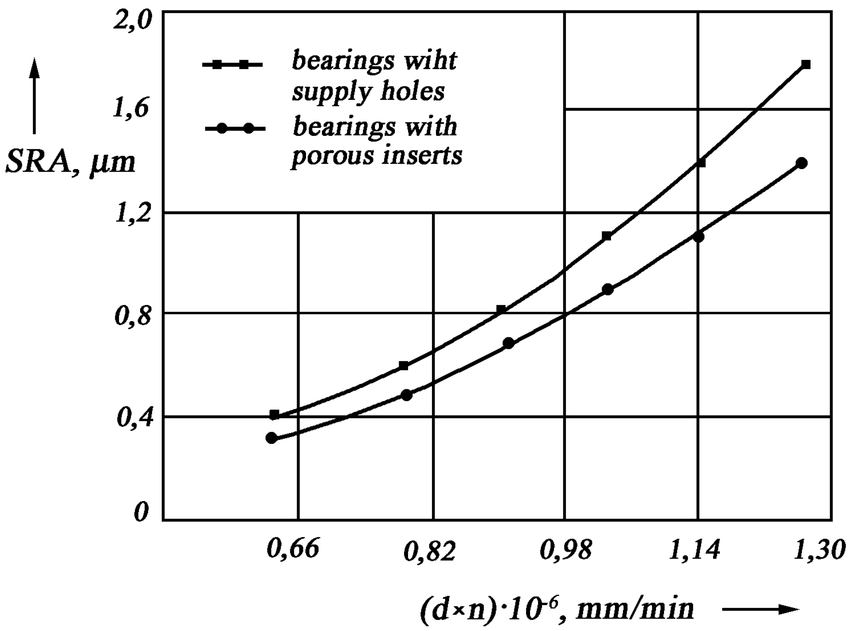

Figure 5. Dependence of the shaft rotation accuracy (SRA) upon the specific speed $d \times n$

From the dependences presented here it is clear that throughout the covered range of shaft specific speed changes the radial run-out in bearings with porous inserts is less than it is in bearings with supply holes. As calculated, the decrease of radial run-out of the shaft is 16…22%.

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