APPLICATION OF A NEW TYPE OF AERODYNAMIC TILTING PAD JOURNAL BEARING IN POWER GYROSCOPE

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New type of aerodynamic tilting pad journal bearing was designed and successfully tested in several applications, one of which was power gyroscope support. Bearing design combines advantages of foil bearings, i.e. additional damping achieved by squeezing out gas film and friction of elastic elements on bearing casing surface, with qualities of classical tilting pad bearings, consisting in defined geometry of bearing gap and excellent stability. Theoretical solution of bearing characteristic calculation is shortly described, consisting in numerical solution of gas flow in narrow gap. Some computed data and results of experiments with rotors operated up to 180.000 rpm are presented too.

Keywords: *aerodynamic journal bearing, tilting pad bearing, foil bearing, rotor stability, power gyroscope, bearing damping, elastic elements*

1. Introduction

The most serious problem of using aerodynamic bearings, i.e. bearings using gas as process media, is to achieve stability of rotor movement. Tilting pad journal bearings (TPJB), which were successfully used in many applications, have excellent dynamic properties ensuring rotor stability at extremely high speeds. There are many modifications of aerodynamic TPJB, differing in complexity of manufacture and ability to adapt itself to operating conditions. Standard tilting pad supports can be replaced e.g. by flexure pivot, elastic or elastomeric support or equipped with active bearing clearance control [1, 2]. New type of aerodynamic TPJB [4], combining advantages of tilting pad and foil bearings, was designed and successfully tested in different operating conditions.

2. Aerodynamic journal bearings

Actually there are two types of aerodynamic journal bearings, which can be used in practice, namely foil bearings [3] – Fig. 1 and tilting pad bearings (TPJB) – Fig. 2.

Foil bearing according to Fig. 1 consists of 2 foils; supporting foil is corrugated and its bumps carry much thinner foil constituting bearing sliding surface. Aerodynamic pressure generated in bearing gap deforms both foils so that bearing gap can adapt itself to operating conditions. Journal vibration evokes friction of supporting foil on bearing casing surface, which results in additional damping. This type of foil bearing exhibits therefore very good dynamic properties, but due to very small thickness of bearing foil the bearing is vulnerable to mechanical damage.

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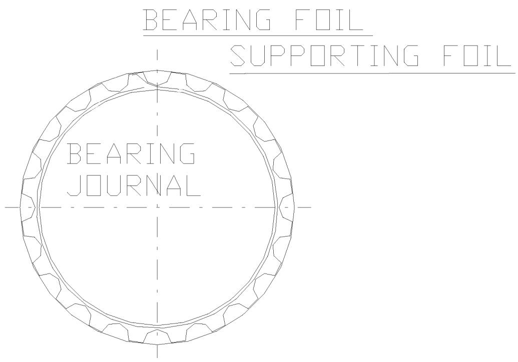


Fig.1: 'Bump' foil journal bearing

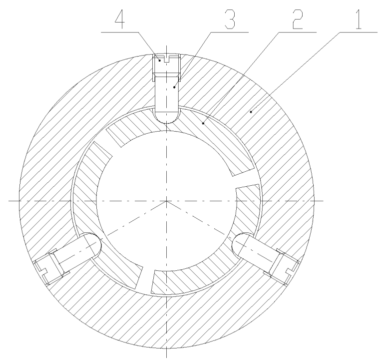


Fig.2: Tilting pad journal bearing

Classical type of TPJB according to Fig. 2 has excellent dynamic properties as regards stability due to very small cross coupling stiffness terms, but its possibility to adapt itself to changed operating conditions is very limited. Special design of TPJB, such as in Fig. 3, enables to compensate reduction of bearing gap occurring due to different thermal expansion of bearing parts. However, design of such a bearing is relatively complex and it lacks additional damping, which is present in foil bearings.

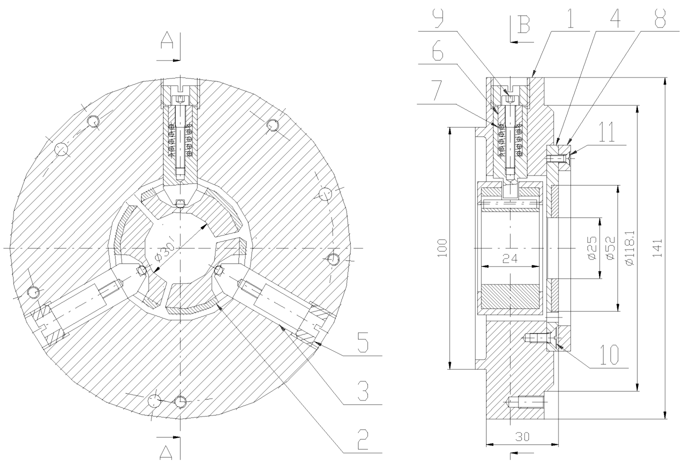


Fig.3: Aerodynamic TPJB with possibility of adaptation to changed conditions

3. New aerodynamic TPJB design

The main advantage of the new bearing (Fig. 4) is its manufacturing simplicity in comparison with design according to Fig. 3. Together with possibility of adjusting bearing clearance, this design significantly decreases requirements to accuracy of manufacture. Apart from that, the bearing exhibits excellent operating properties due to additional damping.

Bearing pads 2 are supported on elastic elements 4, which are deformed to required shape by means of bolts 3 with nuts 6. The difference between inner radius of bearing casing 1 and outer radius of the pad enables rolling of pads on elastic element inner surface, so that they can tilt in circumferential direction. The elastic elements are preloaded to such extent, that maximum pad load capacity exceeds the force necessary for elastic element deformation. In case, that bearing clearance is reduced to dangerously low value, elastic

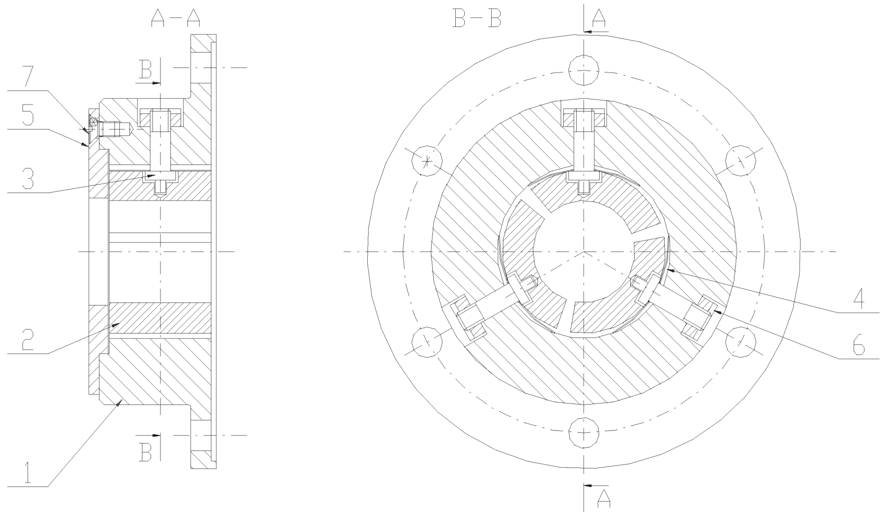


Fig.4: TPJB with pads supported on elastic elements

elements enable restoring bearing clearance to value ensuring safe operation. Basic bearing clearance is adjusted by nuts 6; the pad surfaces are not pressed to the journal and rotor run-up is easy. Unlike in foil bearings, the shape of bearing gap is given by the difference between pad and journal radii and can be therefore optimised. Friction between elastic elements and bearing body contributes to damping of gas film, similarly as in foil bearings. Moreover, overall damping is further increased by squeeze effect of gas pushed out from the gap between elastic elements and bearing body.

4. Calculation of bearing characteristics

Flow in bearing gap is governed by Reynolds equation, which can be written for compressible media as

$$\frac{\partial}{\partial H}\left(H^3\,\frac{\partial P^2}{\partial \vartheta}\right)+\frac{\partial}{\partial \zeta}\left(H^3\,\frac{\partial P^2}{\partial \zeta}\right)=2\,\Lambda\left[\frac{\partial (P\,H)}{\partial \vartheta}+\frac{2}{\omega}\,\frac{\partial (P\,H)}{\partial t}\right]\,, \tag{1}$$

where ϑ and $\zeta = z/L$ are dimensionless coordinates in direction of bearing periphery and bearing width, $P = p/p_{\mathrm{a}}$ is dimensionless pressure, $H = h/c$ dimensionless film thickness, L bearing width, c radial clearance, R journal radius, μ gas dynamic viscosity, ω angular velocity of the rotor, p_{a} ambient pressure and

$$\Lambda = \frac{6\,\mu\,\omega}{p_{\mathrm{a}}}\left(\frac{R}{c}\right)^2 \quad \ldots \text{ dimensionless parameter.}$$

Equation (1) is non-linear in P and can be solved by linearization and numerical methods, one of which is described in [5]. By numeric calculation we get static bearing characteristics, i.e. load capacity and friction losses. Fig. 5 presents example of pressure distribution on upper and lower pad of TPJB 20 mm in diameter for speeds in the range of 75.000 to 225.000 rpm.

Dynamic bearing characteristics, which are needed for rotor dynamic analysis, are determined by solution of Reynolds equation for small harmonic motion of journal. Bearing

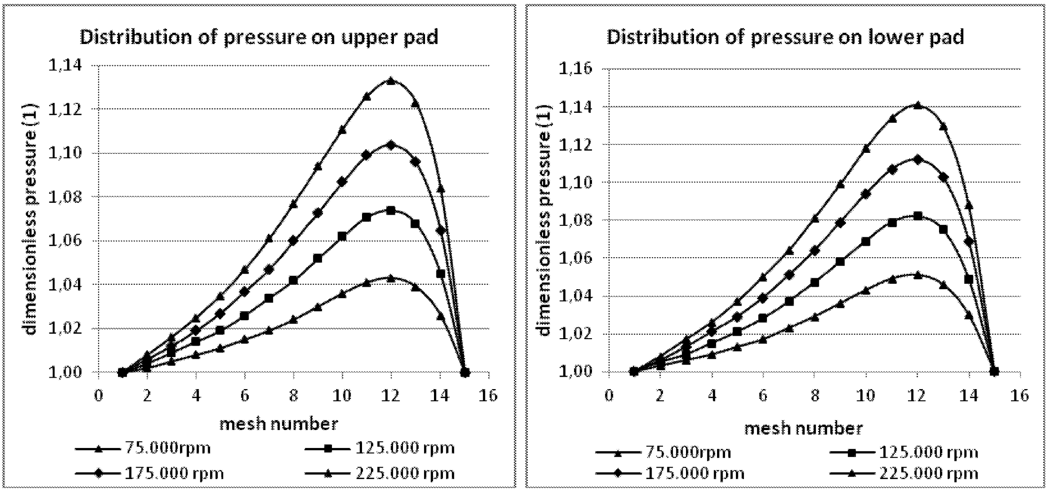


Fig.5: Calculated pressure distribution on pad of TPJB

dynamic properties can be described by matrices of stiffness and damping

$$\mathbf{K} = \begin{bmatrix} K_{\mathrm{xx}} & K_{\mathrm{xy}} \\ K_{\mathrm{yx}} & K_{\mathrm{yy}} \end{bmatrix} , \qquad \mathbf{B} = \begin{bmatrix} B_{\mathrm{xx}} & B_{\mathrm{xy}} \\ B_{\mathrm{yx}} & B_{\mathrm{yy}} \end{bmatrix} , \tag{2}$$

where the 1st index designates force direction and the 2nd index designates direction of movement.

For example K_{xy} means, that force acting in direction x will evoke journal movement in y direction. Dynamic forces acting on the rotor from bearings are given by

$$\begin{bmatrix} F_{\mathrm{x}} \\ F_{\mathrm{y}} \end{bmatrix} = - \begin{bmatrix} K_{\mathrm{xx}} + \mathrm{i}\,\Omega\, B_{xx} & K_{\mathrm{xy}} + \mathrm{i}\,\Omega\, B_{xy} \\ K_{\mathrm{yx}} + \mathrm{i}\,\Omega\, B_{yx} & K_{\mathrm{yy}} + \mathrm{i}\,\Omega\, B_{yy} \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} , \tag{3}$$

where Ω is circular frequency of harmonic journal vibrations.

With elastically supported pads it is necessary to introduce also stiffness and damping of elastic elements, which had to be added to stiffness and matrix of gas film.

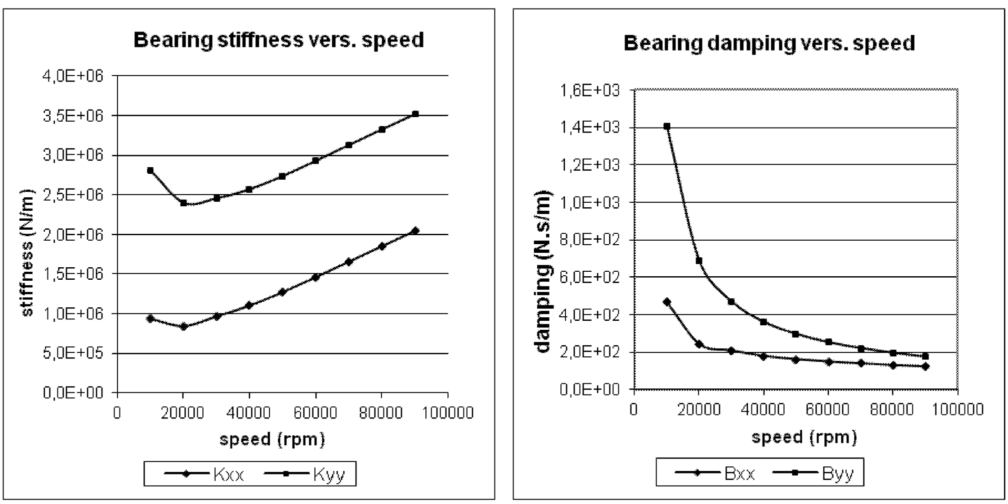
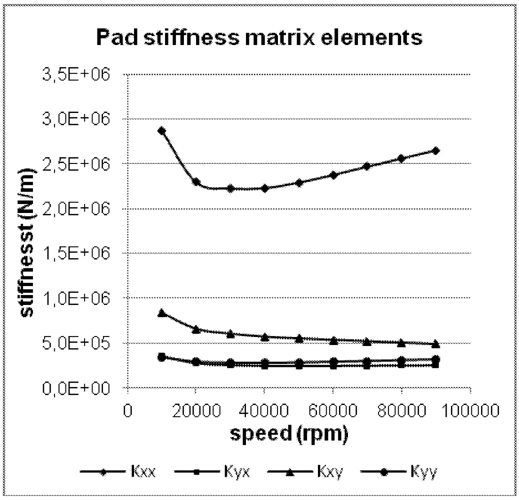


Fig.6: Principle stiffness and damping coefficients of TPJB vers. speed

Fig. 6 shows typical cases of principal stiffness and damping coefficients vers. speed for aerodynamic TPJB 30 mm in diameter.

If pad mass and moment of inertia could be neglected, which holds in most of practical cases, cross coupling terms of stiffness and damping matrix ($K_{xy}, K_{yx}, B_{xy}, B_{yx}$) are at least two orders lower than principal ones ($K_{xx}, K_{yy}, B_{xx}, B_{yy}$). That is why only principle stiffness and damping coefficients are presented in Fig. 6. On the other hand, for dynamics of single pads it is necessary to consider all stiffness and damping coefficients, because they are of the same order, as show diagrams in Fig. 7.



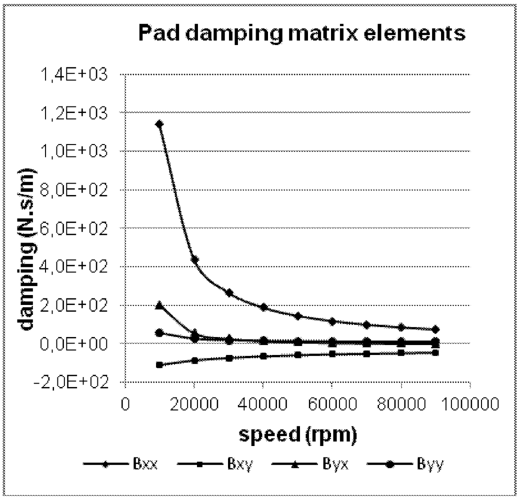


Fig.7: Pad stiffness and damping coefficients vers. speed

Stiffness and damping coefficients are used for rotor dynamic analysis. Due to low damping it is not possible to pass through bending critical speed of the rotor. It is highly recommended, that the 1^{st} bending critical speed is about 80 % above operating speed, otherwise increasing of vibration amplitude can occur at speeds exceeding 60 % of the 1^{st} bending critical speed.

5. Application of the new aerodynamic TPJB

The 1^{st} application of the bearing was at test expansion turbine with aerodynamic bearing support (see Fig. 8) [5].

The rotor 1 with mass of about 0.17 kg, driven by turbine wheel, was supported in two TPJB 2 20 mm in diameter. Rotor vibrations were measured by a pair of relative sensors 10 (Micro-epsilon S05), rotor speed was observed by optical sensor 13. As is documented by vibration signals in time domain as well as by frequency spectra in Fig. 9, the rotor operation was quite stable up to 150.000 rpm with vibration amplitudes lower than 5 μm.

Another successful application of a new TPJB was in a test turbocharger with aerodynamic support of the rotor [6]. Rotor with mass about 0.3 kg, supported in journal bearings 20 mm in diameter, could be operated up to 180.000 rpm with quite acceptable amplitudes of vibration. The rotor in test stand was exposed also to external impulse excitation. As can be seen from Fig. 10, after external shock, evoking rotor excursion almost 100 μm (due

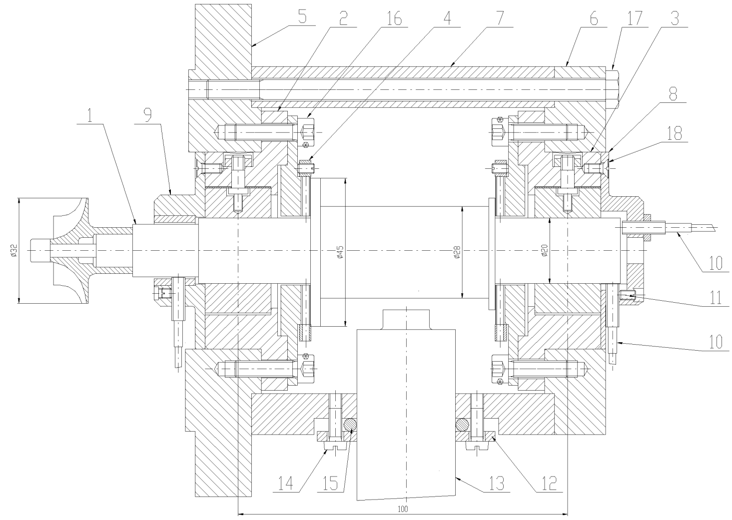


Fig. 8 Test expansion turbine

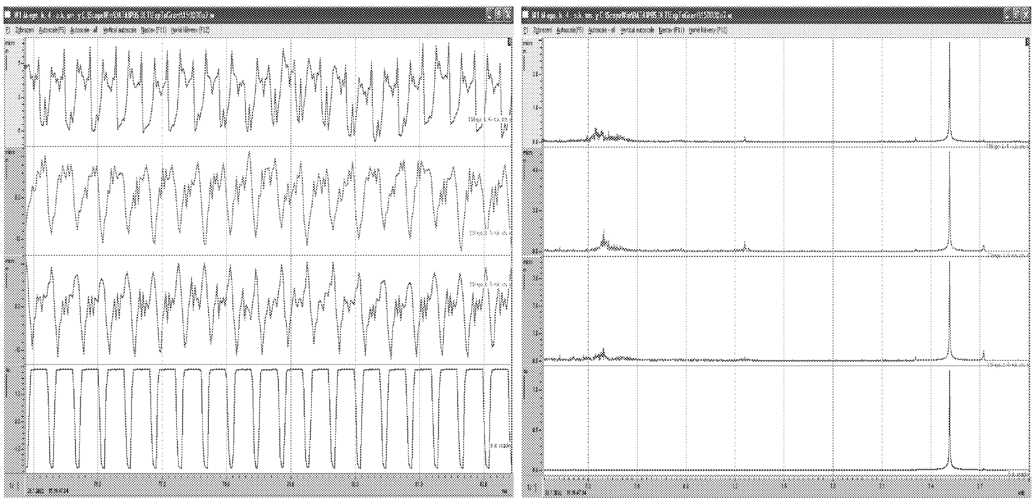


Fig.9: Vibration signals and frequency spectra at 150.000 rpm; top down are signals: 1) rotor – impeller side, direction y, 2) rotor – free end, direction x, 3) rotor – free end, direction y, 4) speed markers

to deformation of pad elastic support), the rotor returned to normal operation regime with vibration amplitudes up to 5 μm.

Recently the new TPJB was applied to power gyroscope for stabilization of vibroizolating platform – see Fig. 11 [7, 8].

Gyro flywheel 2 is driven by compressed air and carried by two TPJB 30 mm in diameter with pads 3 supported on elastic elements 4. The flywheel with vertical axis of rotation is carried by aerostatic thrust bearing 7, because its mass of 3.5 kg is too high for aerodynamic thrust bearing. Two relative sensors were installed for rotor vibration measurement. Sensor B&K In-081 13 observes outer flywheel surface, miniature sensor Micro-epsilon S05-10 12 is fastened directly to one of bearing pads and traces journal surface.

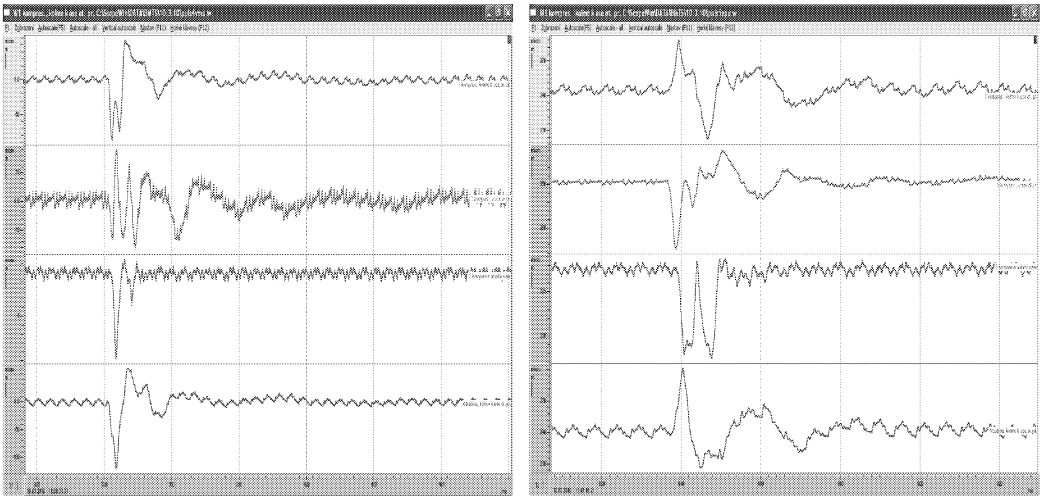


Fig.10: Reaction of rotor in TPJB with elastically supported pads to external shock; top down are signals : 1) rotor – compressor side, direction y, 2) rotor – compressor side, direction x, 3) rotor – axial direction, 4) rotor – turbine side, direction y

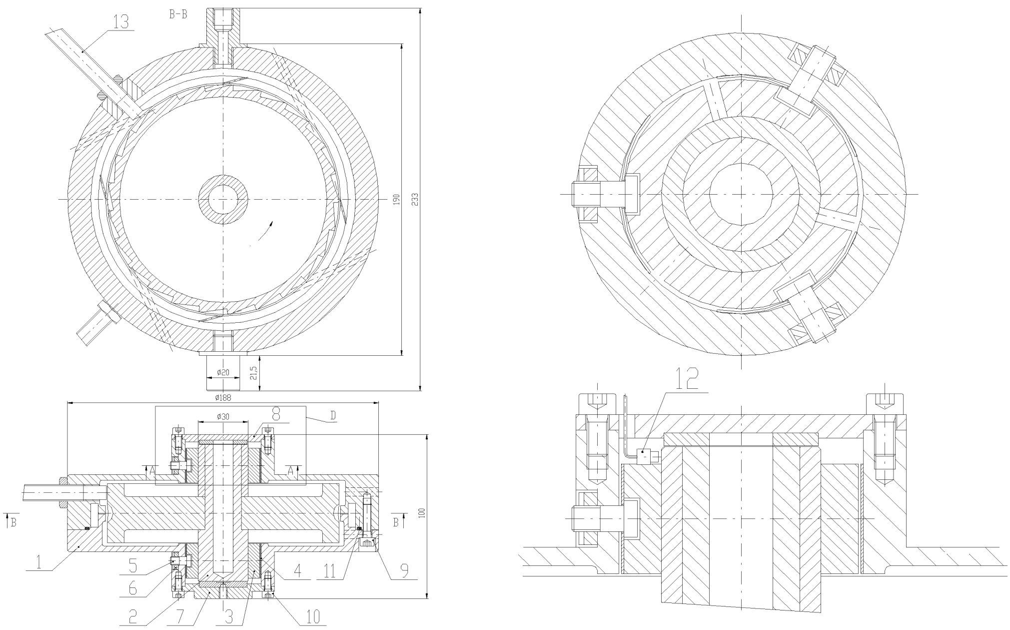


Fig.11: Power gyro supported in gas lubricated bearings with relative sensors

As can be seen from Fig. 12, maximum rotor excursions at 18.000 rpm did not exceed 10 μm (double-amplitude of vibration), while outer surface of gyro wheel exhibits due to run-out excursions more than four times higher. Rotational speed (300 Hz) is clearly visible at signals from relative sensors, but not apparent on acceleration at gyro casing, where prevails blade frequency of 7.57 kHz (25 blades/pockets).

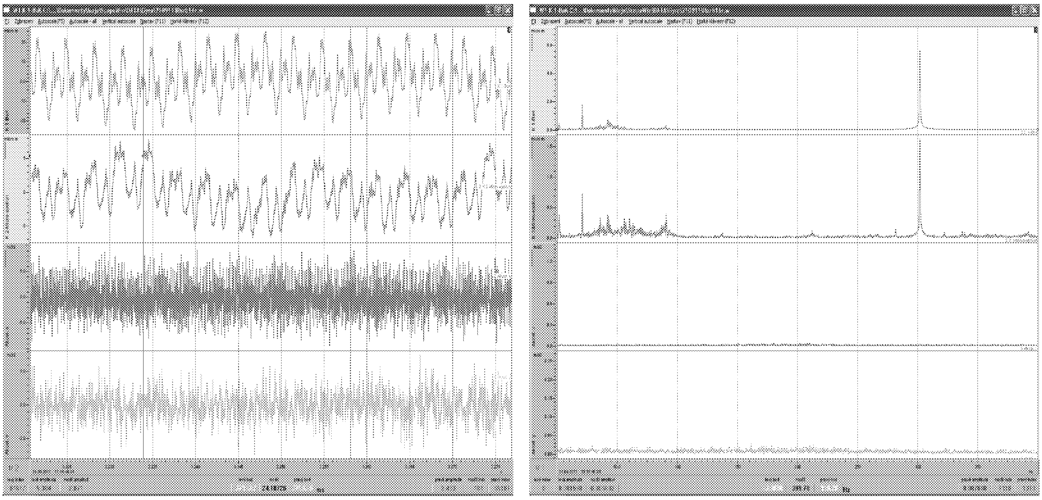


Fig.12: Vibration signals and frequency spectra at 18.000 rpm; top down are signals: 1) gyro wheel outer surface, 2) bearing journal surface, 3) acceleration on gyro casing – radial direction, 4) acceleration on gyro casing – axial direction

Due to low stiffness of bearings and elastic support of the pads, gyro frame is well isolated from gyro wheel vibration, which is desirable. As is documented by Fig. 13, with alternative ball bearing support of gyro wheel RMS value of vibration measured at precession frame at 16.700 rpm was $6.3\,\mathrm{m.s^{-2}}$, while with aerodynamic bearings at 15.200 rpm RMS it was practically one order lower, namely $0.73\,\mathrm{m.s^{-2}}$.

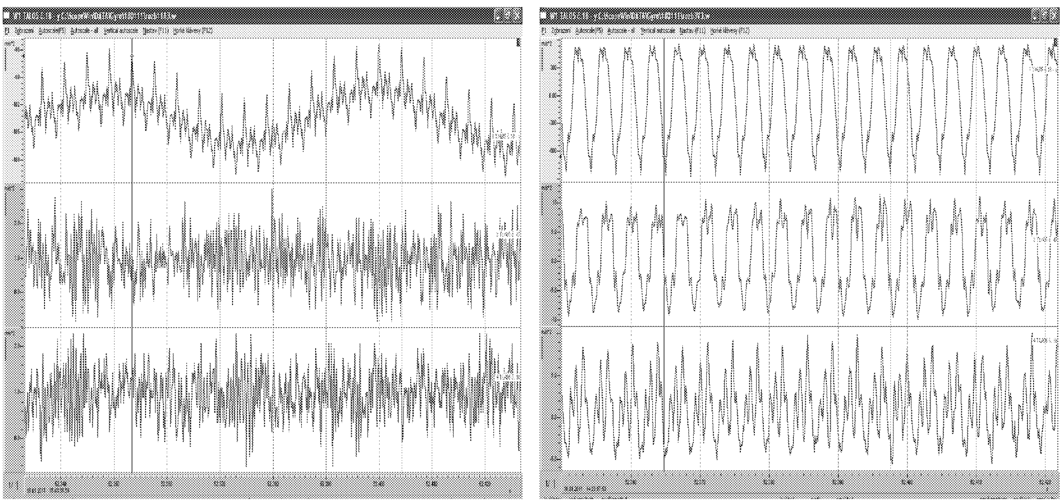


Fig.13: Vibration signals of gyro supported in air bearings (left) and ball bearings (right); top down are signals: 1) gyro wheel outer surface, 2) acceleration at precession frame – horizontal direction, 3) acceleration at precession frame – vertical direction

The difference in friction losses of air bearings and ball bearings is documented in Fig. 14, comparing run-down curves of both bearing variants. As can be seen, the initial slope of both curves is practically the same, because friction losses of gyro wheel prevail. However, with decreasing speed began to show the difference between bearing friction; after 600 seconds gyro with air bearings was running still with about 3.500 rpm, gyro with ball bearings had only 1.500 rpm. The difference in speed further increased and while gyro with ball bearings stopped after about 10 minutes, with air bearings it was running more than 20 minutes.

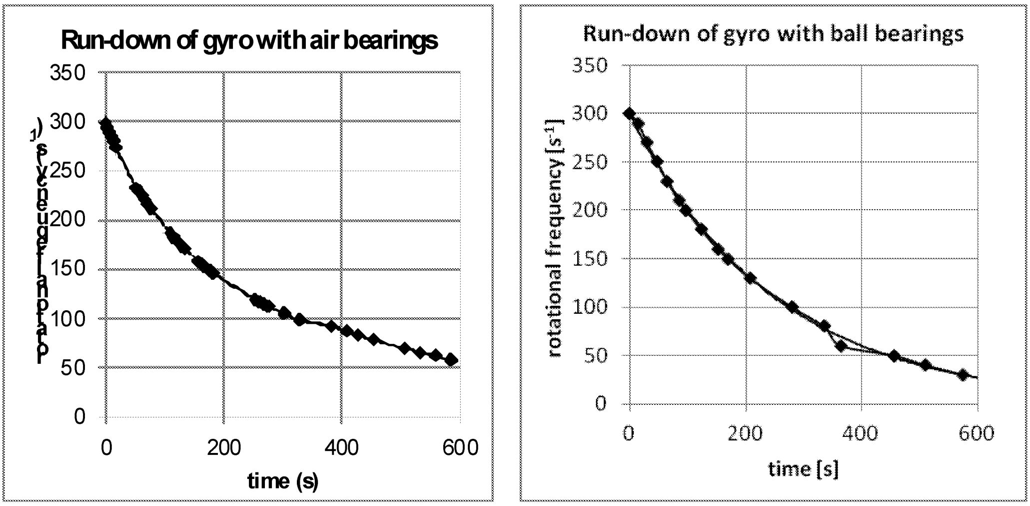


Fig.14: Run-down of gyro with air bearing (left) and ball bearings (right) [9]

6. Conclusion

New type of aerodynamic tilting pad journal bearing was designed and successfully tested in several applications with different operating conditions. Bearing operated safely with high-speed rotors up to 180.000 rpm and even with external impulse excitation, as well as with gyro wheel with mass of 3.5 kg up 18.000 rpm. The bearing according to Czech patent [4] No. 300719 has very simple design and low requirements on accuracy of manufacture. The bearing has therefore wide potential of other applications in different fields of rotating machinery.

Acknowledgement

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