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POWER GYROSCOPES OF STABILIZING SYSTEM

Jiří Šimek*, Jan Šklíba**, Michal Sivčák**, Jan Škoda**

The paper deals with problems concerning power gyroscopes for stabilization of vibro-izolation system. Two variants of gyro support with air drive were designed, namely with gas bearings and precision rolling bearings. Precession frame of the gyro is supported in aerostatic journal bearings to achieve minimum passive resistance. Some special phenomena, such as pneumatic instability, were found in some test regimes both at aerostatic thrust bearing of gyro support and aerostatic journal bearings of the frame support. However, with air inlet pressure level limited to value required for proper function of bearings, no instability was encountered.

Keywords: *stabilized platform, vibration-isolation systems, power gyroscope, aerodynamic tilting pad bearing, aerostatic thrust bearing, aerostatic journal bearing, pneumatic instability, precession frame*

1. Introduction

Stabilizing power gyro is one of the fundamental parts of stabilized platform. Gyro wheel with mass of 3.5 kg, moment of inertia $3.8\times10^{-3}\,\mathrm{kg\,m^2}$ and diameter of 140 mm was designed for speeds up to 35 000 rpm. The paper deals with design and function test of bearing support for both gyro wheel and precession frame. Gyro with pneumatic drive and two types of bearing support, namely in precision ball bearings and aerodynamic tilting pad journal bearings, was tested, as well as aerostatic bearing support of precession frame. The paper includes some test results and describes certain interesting phenomena concerning aerostatic bearings, which were encountered during tests.

2. Power gyroscope

2.1. Pneumatic drive

Pneumatic drive is known to have very small efficiency. With original nozzles of constant cross section it was not possible to achieve speeds higher than 12 000 rpm.

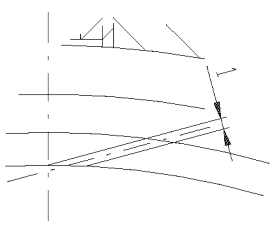

Fig.1: Nozzle of constant cross section

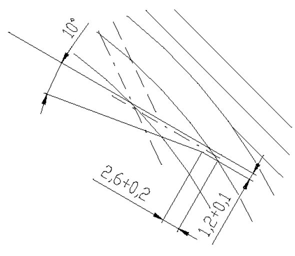

Fig.2: Dilating nozzle

* Ing. J. Šimek, CSc., TECHLAB s.r.o., Sokolovská 207, 190 00 Praha 9
** prof. RNDr. J. Šklíba, CSc., Ing. M. Sivčák, PhD., Ing. J. Škoda, TU Liberec, Studentská 2, 461 17 Liberec

It was therefore necessary to increase nozzle cross-section and in order to achieve supercritical outlet velocity to use dilating nozzles Fig. 2. This modification was possible due to the fact, that nozzles were arranged in the casing plane of separation, so that required shape of nozzles could be manufactured by milling. This modification, increasing the pneumatic drive output more than three times, enabled to raise gyro speed to at least 25 000 rpm. As can be seen form energetic balance in Fig. 3, theoretically it is possible to achieve rotation speed in excess of 35 000 rpm, but due to high friction losses of rotating disk the actual maximum speed was much lower.

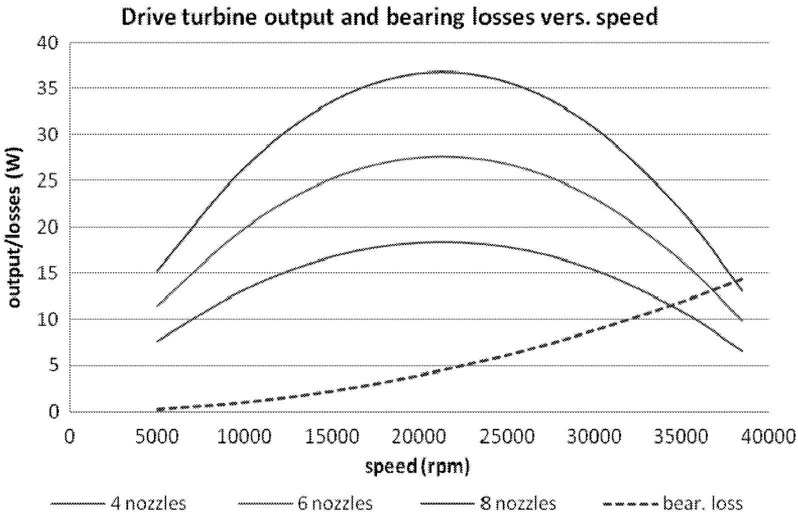

Fig.3: Energetic balance of gyro air drive

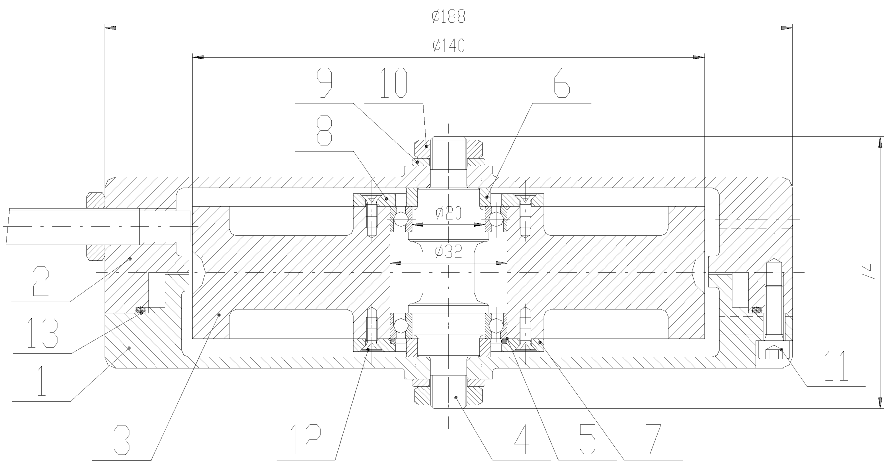

Fig.4: Gyro with precision spindle ball bearings

2.2. Gyro bearing support

As was mentioned above, two variants of gyro support were designed and manufactured. Precision spindle ball bearings with permanent filling of lubricant (see Fig. 4) enable gyro operation up to 43 000 rpm.

Gyroscope 3 is supported in two precision spindle ball bearings 5, arranged on a rigid shaft 4, fastened by nuts 10 in split casing 1, 2. Bearing preload, necessary for elimination of axial clearance, is provided by 'O' ring. Photographs of disassemled gyro parts are shown in Fig. 5. No problems were expected with ball bearing support. However, friction losses were much higher than those of air bearing support, which was manifested by much longer run-down of gyro with air bearings.

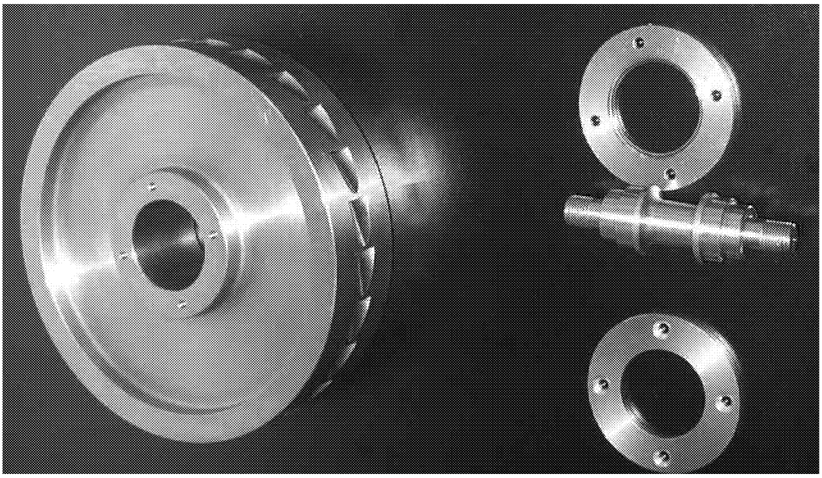

Fig.5: Parts of ball bearing gyro support

Air bearing support (Fig. 6) is a combination of aerodynamic tilting pad journal bearings and aerostatic thrust bearing. Aerostatic thrust bearing had to be used because of vertical rotation axis and high gyro mass, which cannot be supported by aerodynamic bearing. Aerodynamic tilting pad journal bearings guarantee stability of gyro in high speed operation.

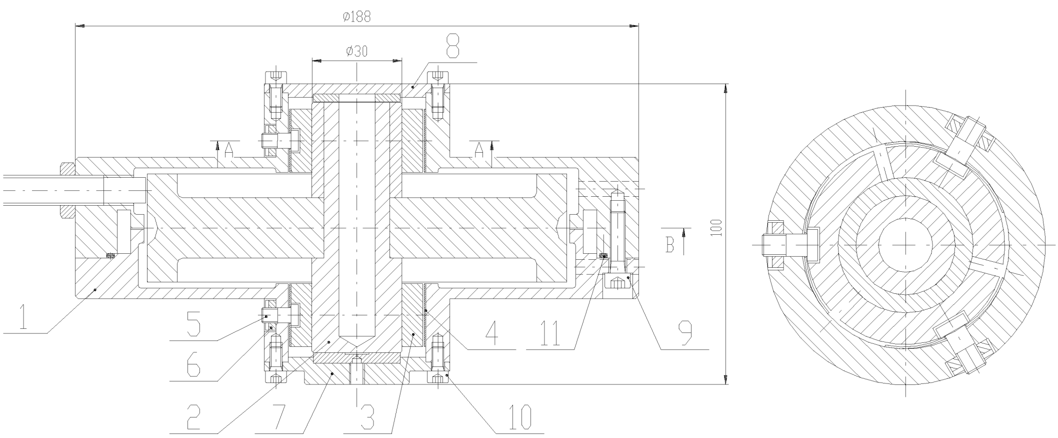

Fig.6: Air bearing gyro support

Gyro rotation axis is held by two tilting pad journal bearings with the pads 3 supported on elastic elements 4, which are bent to a shape, enabling pad tilting by rolling on its outer radius. Elastic elements make also possible shift of the pads in radial direction in case, that bearing clearance was reduced to unacceptably low value. By means of pins 5 with nuts 6 it is possible to adjust basic bearing clearance, so that requirements on manufacture accuracy are significantly reduced in comparison with common tilting pad bearing design. Elastic elements also provide additional damping due to their friction on inner casing surface and squeezing out of the air from the space between elastic element and bearing casing. Aerostatic thrust bearing 7 and aerodynamic thrust bearing 8 secure axial position of gyro wheel within the casing 1. Fig. 7 shows photographs of aerostatic thrust bearing and aerodynamic journal bearing pads (left) and of gyro wheel with the shaft (right).

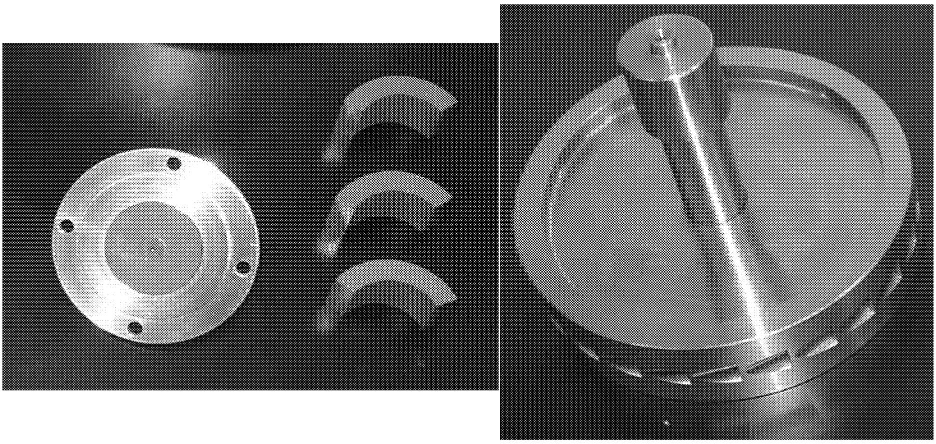

Fig.7: Parts of gas bearing support and gyro wheel

Aerostatic thrust bearing is fed by compressed air through the small orifice – nozzle, which leads to shallow chamber, distributing the pressure to bigger area. Greater volume of gas in chamber can cause pneumatic instability called 'air hammer'; depth of the chamber is therefore very small of the order 0.1 mm. However, even with very small volume of the chamber pneumatic instability can occur, if certain conditions are fulfilled. The instability is manifested by vibration with constant frequency independent of rotation speed (see Fig. 8).

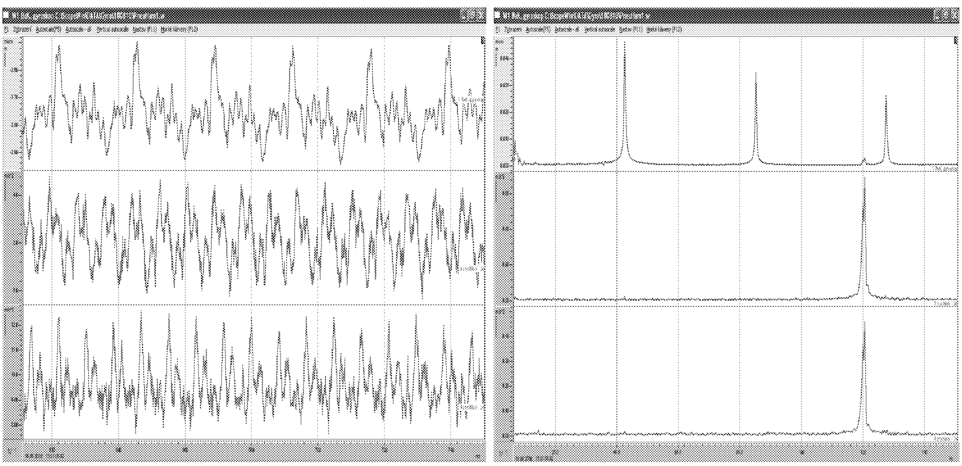

Fig.8: Vibration signals in time domain and respective frequency spectra indicating pneumatic instability of aerostatic thrust bearing (rotational speed 2 540 rpm – 42.4 Hz, frequency of pneumatic instability 120 Hz)

Top down in Fig. 8 are signals from relative sensor, tracing gyro surface, and from two accelerometers fastened to gyro frame in vertical and horizontal directions. Frequency of pneumatic instability of 120 Hz can be seen on both frame vibration signals. With gyro mass of 3.5 kg this frequency corresponds to thrust bearing stiffness of $2\times10^6\,\mathrm{N\,m^{-1}}$, which is quite close to the calculated value of $2.48\times10^6\,\mathrm{N\,m^{-1}}$. The important fact for gyro operation is, that for air supply pressure lower than 0.3 MPa no pneumatic instability appeared and that this pressure level was quite sufficient for separating sliding surface of gyro shaft from thrust bearing.

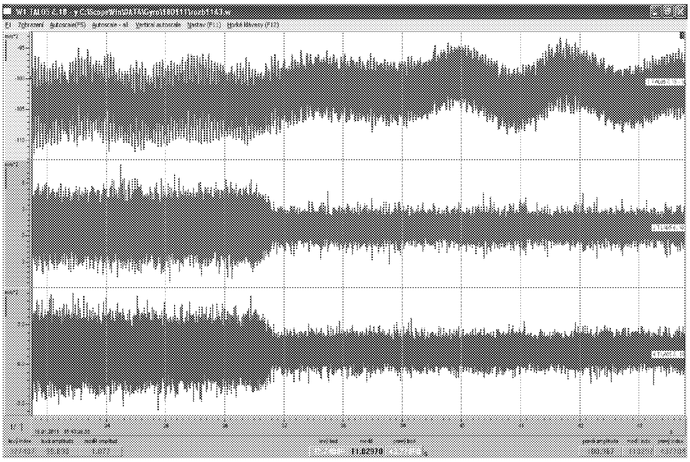

Fig.9: Vibration of gyro in air bearings around 15 000 rpm

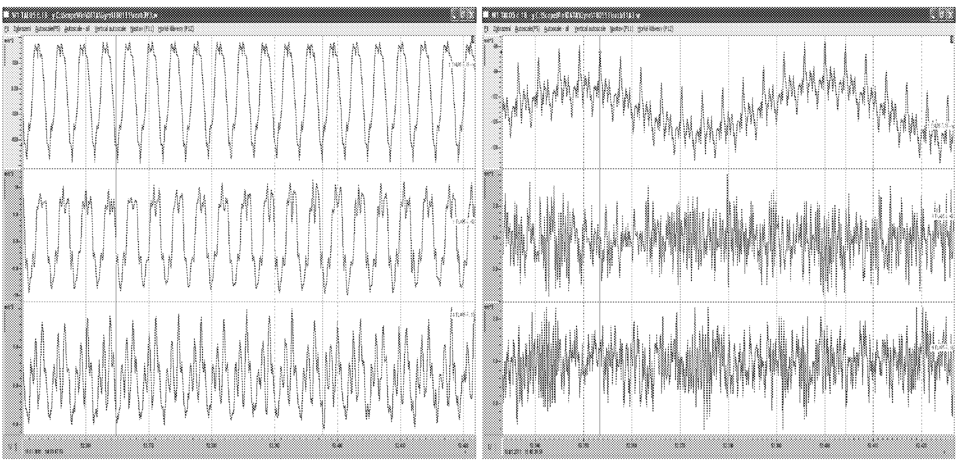

Fig.10: Comparison of vibration level at precession frame with ball bearings (left) and aerodynamic bearings (right)

Amplitude measured on gyro surface did not change much with rotation speed, because this amplitude does not represent vibration in journal bearings, but mainly misalignment of gyro outer surface relative to sliding surface of bearing journals. Due to lack of space it was not possible to install relative sensors in location of bearing journals, by which misalignment would be eliminated. No symptoms of instability were indicated up to maximum speed of 15 000 rpm. As indicates Fig. 9, rapid change in acceleration in directions of precession axis (middle signal), as well as in direction perpendicular to precession axis (lower signal) occurred in neighbourhood of maximum speed. Gyro exhibited subharmonic vibration with frequency of 20 Hz (1/12 of rotation speed). This frequency corresponds to eigenfreguency of gyro wheel on elastic elements with stiffness of $2\times10^4\,\mathrm{N\,m^{-1}}$. It proves, that elastic elements of tilting pad journal bearings isolate gyro from the casing and is documented by comparison of vibration level at precession frame in Fig. 10. Sequence of vibration signals in Figs. 9 and 10 is the same as in Fig. 8. While with ball bearing support RMS value of vibration at precession frame at about 16 700 rpm was $6.3\,\mathrm{m\,s^{-2}}$ (left diagram), with air bearings at 15 200 rpm it was practically one order lower, namely $0.73\,\mathrm{m\,s^{-2}}$ (right diagram).

2.3. Aerostatic bearings of precession frame

As is apparent from Fig. 11, gyro casing is supported in two aerostatic journal bearings 3 of precession frame. In order to minimize parasitic moments, compressed air is supplied through orifice in one of aerostatic bearing journals.

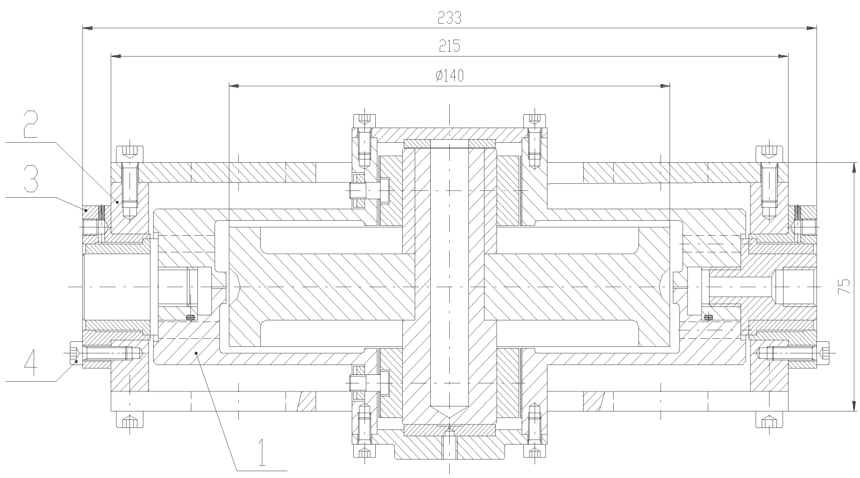

Fig.11: Cross section of gyro with air bearing support mounted in precession frame

Photograph of complete gyro with air bearing support mounted in precession frame is in Fig. 12. To show aerostatic thrust bearing the frame is reversed, so that the bearing is on top.

Aerostatic bearings of precession frame were designed alternatively with porous surface, in order to minimize parasitic moments, and with drilled orifices, to achieve greater load carrying capacity. Both types of bearings worked quite satisfactorily. However, with higher air inlet pressure pneumatic instability – air hammer – appeared once more in porous bearing variant. Onset of instability was dependent on air inlet pressure as well as on bearing load. Bearing loaded with mass of precession frame only, which is much lower than gyro with the casing, exhibited no instability up to the highest inlet pressure of 0.5 MPa. When the bearings were loaded with the mass of the casing with gyro, pneumatic instability appeared with air inlet pressure higher than 0.35 MPa.

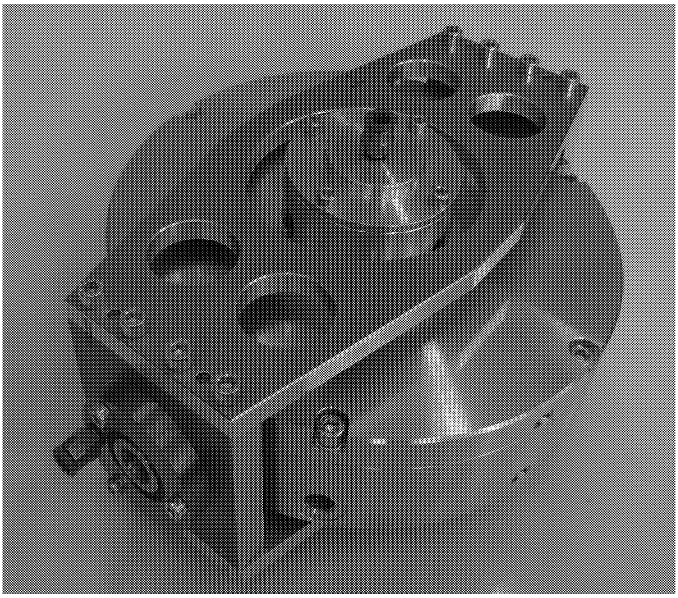

Fig.12: Photograph of gyro with air bearing support mounted in precession frame

inlet pressure (MPa)	vibration frequency (Hz)	v_{xef} ($\mathrm{mm\,s^{-1}}$)	v_{yef} ($\mathrm{mm\,s^{-1}}$)	v_{zef} ($\mathrm{mm\,s^{-1}}$)	a_{xef} ($\mathrm{mm\,s^{-2}}$)	a_{yef} ($\mathrm{mm\,s^{-2}}$)	a_{zef} ($\mathrm{mm\,s^{-2}}$)
0.36	263	0.30	0.53	0.32	0.60	0.75	0.46
0.40	266	0.41	1.30	0.34	0.76	2.02	0.53
0.45	275	0.92	4.25	0.57	0.48	6.69	1.30
0.50	276	1.22	**7.34**	0.95	0.76	**11.26**	1.43

Tab.1: Dependence of vibration frequency, RMS value of acceleration a_{ef} and velocity v_{ef} on pressurized air inlet pressure

As is clear from Table 1, frequency of vibration as well as vibration level were dependent on air inlet pressure, though influence on frequency is only slight. Frame vibrations were measured simultaneously by Triaxial DeltaTron accelerometer Brüel & Kjær 4524-B (sensitivity $10\,\mathrm{mV/(m\,s^{-2})}$, measuring range $\pm 500\,\mathrm{m\,s^{-2}}$, frequency range $0.25\,\mathrm{Hz} \div 5\,\mathrm{kHz}$) and 3 accelerometers Techlab TLA05 (sensitivity $40\,\mathrm{mV/(m\,s^{-2})}$, measuring range $\pm 50\,\mathrm{m\,s^{-2}}$ and frequency range of $0 \div 4.5\,\mathrm{kHz}$). B&K 4524-B sensor was connected to apparatus TECHLAB TVLD-6, enabling to evaluate acceleration as well as velocity of vibration. Table 1 shows measured RMS (root mean square) values of acceleration (measured by TLA05) and RMS values of velocity (measured by B&K 4524-B) in all three directions. RMS values are statistic quantities determined from very big number of measured points and have therefore high testifying ability.

Frequency spectra from vibration signals of acceleration in directions y (vertical), x (direction of precession axis) and z (direction perpendicular to precession axis) and velocities in directions x, y and z, measured on precession frame are shown in Fig. 13 and 14 in multi-projection (top down are acceleration in directions y, x, z and velocities in directions x, y, z). As can be seen both from the Table 1 and Figs. 13 and 14, intensity of pneumatic instability grows very quickly with increasing air inlet pressure.

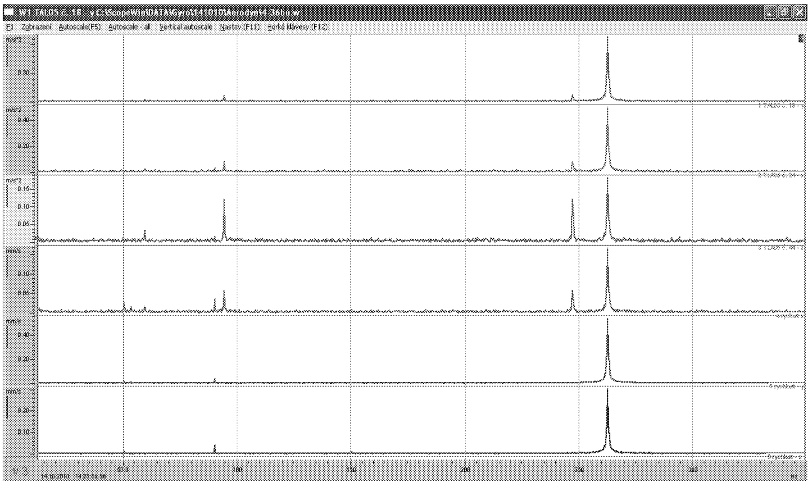

Fig.13: Pneumatic vibration in journal bearings of precession frame with air inlet pressure of 0.36 MPa – frequency of vibration 263 Hz, $a_{\max} = 0.77\,m\,s^{-2}$, $v_{\max} = 3.9\,mm/s$

The highest values of acceleration and velocity were measured in vertical direction, which is coincident with direction of static load. No pneumatic instability was encountered with air inlet pressure lower than 0.35 MPa. This is very important fact, because as well as in aerostatic thrust bearing, inlet pressure of 0.3 MPa is quite sufficient for separation of sliding surfaces and proper function of bearings.

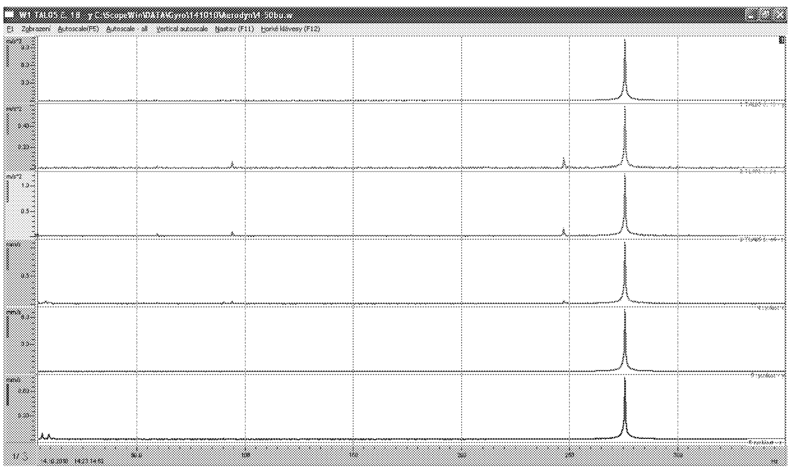

Fig.14: Pneumatic vibration in journal bearings of precession frame with air inlet pressure of 0.5 MPa – frequency of vibration 275 Hz, $a_{\max} = 30.9\,m\,s^{-2}$, $v_{\max} = 18.0\,mm/s$

From measured frequencies it was possible to determine stiffness of aerostatic bearing of precession frame; for mass of about 2 kg, corresponding to load acting on one bearing, and frequency of 260 Hz one gets value of stiffness $5.34\times10^{6}\,\mathrm{N\,m^{-1}}$ for inlet pressure of 0.3 MPa. Corresponding calculated value of bearing stiffness is $5.4\times10^{6}\,\mathrm{N\,m^{-1}}$, which is in quite good accordance with measured value.

3. Conclusions

Power gyro with pneumatic drive and two variants of bearing support was designed and successfully tested. Due to friction losses higher than expected, pneumatic drive had to be modified, using dilating nozzles to achieve above critical velocity of gas. Modification of nozzles enabled to reach minimum rotational speed necessary for stabilization of the system. Aerostatic thrust bearing of air bearing support and aerostatic journal bearings of precession frame exhibited in some conditions pneumatic instability, which would disturb stabilizing function of the gyroscope. However, as was proved by tests, by limiting the pressurized air inlet pressure by 0.30 MPa, no symptoms of pneumatic instability appeared, while both aerostatic bearings operated quite satisfactorily.

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