Int. Journal of Applied Mechanics and Engineering, vol.11, No.3, pp.501-506

This paper was prepared on the Fourth International Tribology Conference ITC 2006

INCREASE OF RELIABILITY AND TRIBOLOGICAL CHARACTERISTICS OF GYROSCOPES

E. KOVALEV
A.A.Baikov's Institute of Metallurgy and Material Science
Moscow, RUSSIA

M. IGNATIEV
Moscow Institute of Electronic Technique
Moscow, RUSSIA

H. WIŚNIEWSKA-WEINERT* and V. LESHCHYNSKY
Metal Forming Institute
Jana Pawla II Str. No 14, 61-139 Poznan, POLAND
e-mail: t.rozwalka@inop.poznan.pl

Now gyroscopic devices are applied in many industrial areas. Alongside with traditional mechanical gyroscopes, the increasing weight is obtained by systems of mechanical group realizing more perfect principles of their construction, such as gyroscopes with non-contact levitating rotor. However, the complexity of gyroscopic devices entails a number of problems which are not solved till now. One of the most critical problems is the reliability of a rotor and stator at an emergency situation. This situation is characterized by "landing" of a rotor on the stator. It leads to destruction of elements of the stator because of high tangential loading (friction coefficient of ferrite is more than *0.25*). Special coatings for ferrite materials have been developed to decrease the friction coefficient (to *0.07*) and to provide the stator and rotor reliability even under emergency situation: an amorphous (Al-Si)N magnetron coating and coating based on MoS2 nanoparticles with polymer binder. Tests of the developed coating were carried out in vacuum $\left(10^{-4} - 10^{-6}\ \text{Pa}\right)$ at loading *1-5* MPa and sliding velocity *of 0.2-1.0 m/s.* The test results show that the problem of durability of gyroscopes can be successfully solved with the help of new coatings.

Key words: gyroscope; nanoparticles; solid lubricant coating; friction coefficient; wear intensity.

1. Introduction

Now gyroscopic devices are applied in many industrial areas (Magnus, 1974). Alongside with traditional mechanical gyroscopes, the increasing weight is obtained by gyroscopes with non-contact levitating rotor. The cost of these systems is very high: up to some tens thousand EUROS. However, the complexity of gyroscopic devices entails a number of problems which are not solved till now. One of the most critical problems is the reliability of the rotor and stator at an emergency situation. This situation is characterized by "landing" of the rotor on the stator. It leads to destruction of elements of stator because of high tangential loading (friction coefficient of ferrite is more than *0.24*). This problem is very important for systems with electromagnetic suspension of a rotor (Fig.1), where the basic constructional material both for the rotor and stator is ferrite or other ferromagnetic material, having low strength and weak tribological properties (high friction coefficient >*0.25*).

At an emergency situation, the pair "rotor-stator" is under conditions of dynamic instability. Angular rotation speed of the rotor is $\approx 1.2 \cdot 10^{-5}\ \text{min}^{-1}$. At a casual deviation of an axis of the rotor from an axis of

* To whom correspondence should be addressed

the stator on very small distance the surface of the rotor will contact a part of console elements of the stator. The configuration of the stator is similar to two bowls with a wide-angle diffuser. In emergency situation a force of friction will roll the rotor on an internal surface of the stator bowl. In this case, the rotor precesses in a direction that is opposite to the direction of the rotor rotation. As a result of such movement, a periodically revolting force operates on each console element of the stator. This force is summarized from centrifugal force and force of friction. A tribo-dynamic analysis of the gyroscopic system is required to define the hazard of this situation and to estimate the risk of gyroscope destruction under emergency situation. The requirements for anti-friction coatings will be defined based on the results of the tribo-dynamic analysis.

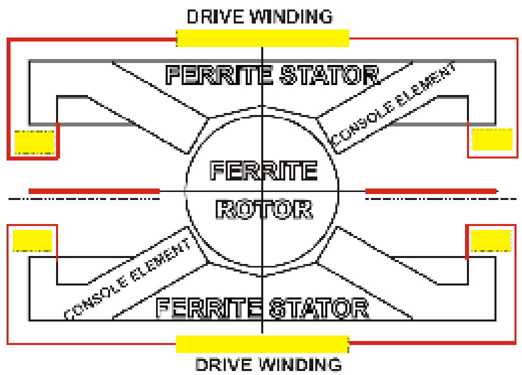

Fig.1. Gyroscopic devices with the electromagnetic suspension of a rotor.

2. Tribo-dynamic analysis

This analysis is realised through a comparison of natural vibration frequency and forcing frequency (Hartog, 1985). If these frequencies considerably differ from each other then the resonance does not come and operating conditions for console elements are favorable. Besides, when the forcing frequency is 2-3 times higher or lower than the natural vibration frequency (without taking into account damping factor), the dynamic factor is easily calculated. This factor allows to estimate an amplitude of forced vibrations in comparison with the static movement caused by the maximal value of revolting force.

The tribo-dynamic model of a gyroscope under emergency situation represents the system consisting of rods (consoles) of variable section of a complex configuration with one fixed end on which the periodically revolting force operates. The equation of compelled fluctuations of elastic (isotropic) bodies is presented below (Thomson and Dahleh, 1997)

$$\left[(\lambda+\mu)\nabla div + \mu\nabla^2 + \rho\omega_1^2\right]\xi = -F \exp j\omega_1 t \tag{2.1}$$

where λ and μ – the Lame factor, ρ – the material density, ξ – the displacement of all points of a body at fluctuation, ω_1 – the frequency of revolting force, F – the revolting force, t - time

Knowing the strength characteristics of ferrite (Mn-Zn type ferrite 1500 NM3 was applied) it is possible to define requirements for the coating friction coefficient. If this requirement is satisfied, it will be possible to avoid gyroscope destruction under emergency situation.

The differential equation of bending free oscillations of the rod with a constant cross-section is presented below

$$\frac{EI}{sp}\frac{\partial^4 v}{\partial x^4} + \frac{\partial^2 v}{\partial t^2} = 0 \tag{2.2}$$

where $v = v(x,t)$ - a deflection of the current point of an axis of a rod, EI - bending rigidity of section of a rode, $s\rho$ - intensity of weight of a rod (weight of unit of length), s - the area of cross section.

The differential equation compelled bending fluctuations of a rod with constant cross-section is presented below

$$\frac{\partial^4 v}{\partial x^4} + \frac{\rho s}{EI}\frac{\partial^2 v}{\partial t^2} = \frac{q(x)\sin\omega t}{EI} \tag{2.3}$$

where $q(x)\sin\omega t$ - the intensity of the distributed revolting loading, the frequency of revolting force. After replacement $v(x,t) = y(x)\sin\omega t$ Eq.(2.3) can be transformed as

$$y^{IV} - \frac{\rho s\omega^2}{EI}y = \frac{q(x)}{EI}. \tag{2.4}$$

At the concentrated perturbations, the differential Eq.(2.4) on each of the regions becomes homogeneous, and revolting forces are included into boundary conditions (or conditions of interface of regions). In this case (Hartog, 1985)

$$y = C_1 S(\alpha x) + C_2 T(\alpha x) + C_3 U(\alpha x) + C_4 V(\alpha x) \tag{2.5}$$

where C_1, C_2 - the constants determined by boundary conditions, S, T, U, V – Krylov's functions. Using methods of Hartog (1985) and Weaver (1995) the maximum bending stress in console elements of the stator have been determined

$$\sigma_{bs} = \sigma_{bss}\beta \tag{2.6}$$

where $\sigma_{bss} = \dfrac{Fl}{W}$ - stresses which would arise in the system at the static action of the maximum value of the revolting force, β – the factor of dynamism, l – the rod length, W - the moment of resistance of section of the console.

On conditions of strength

$$\sigma_{bs} = \frac{Fl}{W}\beta \le [\sigma]_{bs} \tag{2.7}$$

where $[\sigma]_{bs}$ bending strength limit. The frequency of the basic tone of own oscillations of the console is defined in accordance with the Hartog (1987) Eq.(2.8)

$$p = \frac{3.52}{l^2}\sqrt{\frac{EI}{\rho s}}. \tag{2.8}$$

For the definition of the basic tone of own oscillations of console elements the Eq.(2.8) was applied. Because of the significant spread in values of the elasticity module of the ferrite, the step method for estimation of own frequencies has been proposed. The module of elasticity is varied in the range $0.45 - 2.15 \cdot 10^{11}$ Pa with step $0.34 \cdot 10^{11}$ Pa and related frequency was calculated for each value. Numerical values of key parameters in the Eq.(2.8):

$$l = 5 \cdot 10^{-3} M, \qquad I = \pi d^4/64 = 3.97 \cdot 10^{-12} m^4,$$

$$S = \pi d^2/4 = 7.065 \cdot 10^{-6}, \qquad \rho(1500\ NM3) = 3900\, kg/m^3 .$$

The frequency of revolting force in an initial stage of an emergency situation is equal to $\approx 0.125 \cdot 10^5 s^{-1}$. A comparison of frequencies of own oscillations ($p_{\min} \approx 0.327 \cdot 10^6 s^{-1}$, $p_{\max} \approx 0.784 \cdot 10^6 s^{-1}$) and the revolting force shows that they differ in the order of magnitude. Therefore the resonance will not arise.

Dependence of bending stresses on friction coefficient is presented in Fig.2. The strength of the applied ferrite material is equal to *41* MPa. From Fig.2 it is clear that the probability of the process of destruction of console elements of the stator is reduced to a minimum if the safety factor is equal to *1* (at friction coefficient <*0.144*). For real designs the safety factor has been increased twice. Therefore in a real product it is necessary to use a coating with the friction coefficient *0.07-0.075*.

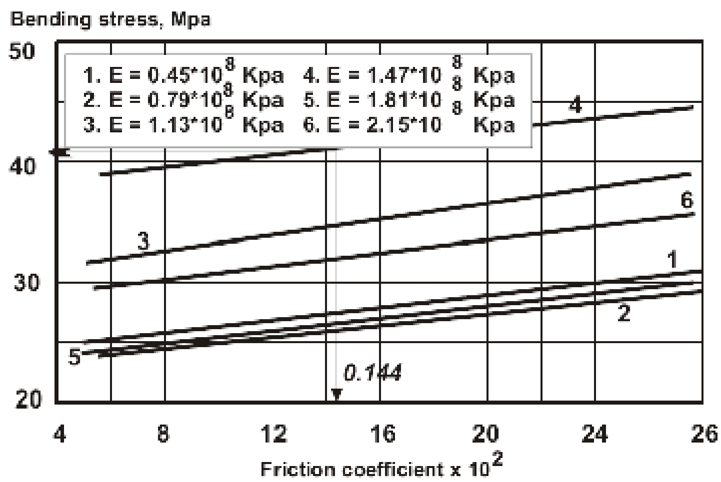

Fig.2. Bending stress *vs.* friction coefficient for Mn-Zn type ferrite *1500 NM3*.

The following methods for coating deposition on Mn-Zn ferrite were developed: (1) Coating containing MoS_2 nanoparticles with *80-150 nm* size. The coating was deposited with the help of the following binding components: epoxy resin; Praymer type plastic; (2) Magnetron deposition of amorphous (Al-Si)N coating. (Al-Si)N coatings were deposited with a commercially available sputter device (Leybold-Heraeus Z400) by reactive RF magnetron sputtering from a (Al-Si) target in a mixed Ar-N_2 atmosphere. The (Al-Si) target was of *96%* purity with main impurities of potassium, chromium and iron. The gas flow of *38* standard cm^3min^{-1} argon and *4* standard cm^3min^{-1} for N_2 was adjusted by mass flow controllers. The partial pressure of the working and rest gas was measured with a quadrupole mass analyzer, which was connected with the sputter chamber by a pressure converter. The total pressure was kept at $8 \cdot 10^{-3} mbar$. Evacuation was accomplished with a $4001 s^{-1}$ turbo-molecular pumping system. Prior to coating deposition the target was sputter etched for *15 min*.

Tests of coatings were applied to special ferrite samples (disk-disk scheme) under vacuum conditions $\left(10^{-4} - 10^{-6}\,\mathrm{Pa}\right)$ at loading *1-5* MPa and with sliding velocity $0.2 - 1.0\, m/s$ with the help of the method described by Kovalev *et al.* (2005).

The duration of the test has been chosen within the range of: *5-10* minutes - for the definition of the average wear intensity; *60-120* minutes for the estimation of the average friction coefficient. The chosen test duration some times surpasses the time from the landing of the rotor on the stator up to full stop of the gyroscope. Thus, it is possible to assume that if the developed coatings keep their serviceability during tests (absence defoliation from the ferrite), they could be applied for real gyroscopes to prevent their destruction at emergency situation.

3. Results and discussion

Tribological properties of coatings are presented in Figs 4 and 5 for different testing conditions. The friction coefficient for the developed coatings decreases with an increase in loading. An elastic contact is kept in a tested range of loadings at which the adhesive component of friction force decreases. The falling characteristic of the friction coefficient versus sliding velocity is observed for friction pairs with coating based on MoS2 nanoparticles. A reduction of the friction coefficient is defined by reduction of time of frictional bond, and also by structural features of MoS2 texturing of coating.

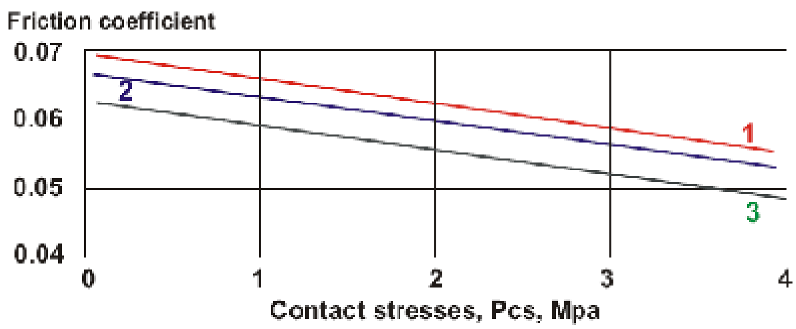

Fig.4. Friction coefficient *vs.* contact stresses for Mn-Zn type ferrite *1500* NM3 (sliding velocity $v_s = 0.5\, m/s$, temperature – *298 K*, vacuum – 10^{-5} Pa). (1) – Ferrite – (Al-Si)N; (2) – Ferrite - (MoS_2 + epoxy resin); (3) – Ferrite - (MoS_2 + Praymer).

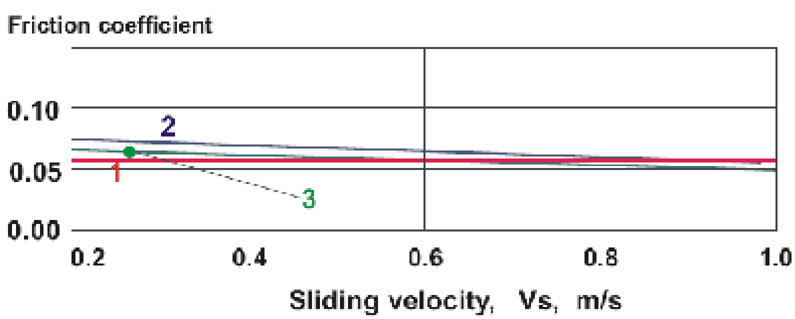

Fig.5. Friction coefficient *vs.* sliding velocity for Mn-Zn type ferrite *1500* NM3 (loading – $p_{cs} = 3$ MPa , temperature – *298 K*, vacuum – 10^{-5} Pa). (1) – Ferrite – (Al-Si)N; (2) – Ferrite – (MoS_2 + epoxy resin); (3) – Ferrite - (MoS_2 + Praymer).

A coating based on MoS2 nanoparticles has a lower friction coefficient at higher sliding velocity, but increased wear intensity (Tab.1). It is caused by defects in coating surface layers and important mass transfer. The application of this coating can result in imbalance of the rotor and saturation by wear debris of internal volume of the gyroscope. This coating could be applied only in expendable gyroscopes.

The friction coefficient of the (Al-Si) N coating remains constant for all applied sliding velocities. The wear intensity is much lower than for MoS2 based coating. Such result is most likely explained by amorphous coating structure. This coating is the most suitable for systems which should have increased service life.

Table 1. Contact stresses – 5 MPa, sliding velocity – $0.5\,m/s$).

No	Material of Samples	Coating	Wear Intensity, (Linear wear/sliding distance)
1	Mn-Zn Ferrite *1500*NM3	MoS_2 + epoxy resin	10^{-6}
2	Mn-Zn Ferrite *1500*NM3	MoS_2 + Praymer	10^{-6}
3	Mn-Zn Ferrite *1500*NM3	(Al-Si)N	10^{-8}

4. Conclusion

One of the most critical problems of modern gyroscopes with non-contact levitating rotor is the reliability of the rotor and stator at an emergency situation: destruction of system elements resulting from "landing" of the rotor on the stator.

A gyroscope dynamic model was developed to define requirements for antifriction coatings allowing avoidance of destruction at emergency situation.

Special coatings for ferrite materials have been developed to decrease the friction coefficient (to 0.07) and to provide the stator and rotor reliability even under emergency situations: the amorphous (Al-Si)N magnetron coating and coating based MoS2 nanoparticles with a polymer binder.

The (Al-Si)N coating is the most suitable for systems with long service life. The coating should be deposited on console elements surfaces with initial roughness better than $1\mu k$.

The coating based MoS2 nanoparticles with polymer binder is lees expensive, but it is characterized by higher wear intensity. It could be applied for relatively cheap systems with short service life.

Acknowledgments

The coating based on MoS_2 nanoparticles presented in this paper was developed in the frame of work supported by the INTAS Project Nr 04-80-7362 NANOBLEBUS.

References

Hartog J.P.D. (1985): *Mechanical vibrations*. – Dover Publications, pp.436.

Hartog J.P.D. (1987): *Advanced Strength of Materials*. – Dover Publications, pp.388.

Kovalev E., Ignatiev M., Leshchynsky V. and Wisniewska-Weinert H. (2005): *Friction and wear of dissusion MoS2 solid lubricant coatings*. – Operation Problems 2/2005 (57) ISSN 1232-9312 pp.171-180.

Magnus K. (1974): *Gyroscope: theory and applications*. – M.: Mir, pp.526.

Thomson W.T. and Dahleh M.D. (1997): *Theory of Vibration with Applications*. – 5th Edition, Prentice Hall, pp.5-23.

Weaver W., Timoshenko S.P. and Young D.H. (1990): *Vibration Problems in Engineering*. – 5Th Edition, Published in Jan 1990 by Wiley US, pp.624.

Received: May 28, 2006

Revised: August 1, 2006