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Automatic regulation of clearance in a tilting pad journal bearing

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1. Introduction

Hydrodynamic tilting pad journal bearings are widely used in turbines, compressors, pumps, machine tools and other machines where rotors are revolving for a long time without frequent start and stoppage. Their advantage is in smooth and accurate revolving motion, good vibration damping properties, high stiffness, service durability. But the main problem of such bearings is the value of clearance between the journal and pad. First of all, at assembly of such bearings in many cases there are not adequate means to measure the clearance, the second, at operation of different bearings, the temperature in different cases after starting from 18-20°C at longer work increases (in researched bearings [1] up to 41-43°C, at other experiments [2] – up to 55-65°C; at approach to failure the temperature may increase up to 85°C). Thermal deformation of different elements of bearings is different, it may hardly influence on clearance value, and at absence of means for clearance control it is necessary at assembly to set the maximal clearance for the worse expected work conditions of the bearing. Increased clearance increases shaft eccentricity at revolving and decreases bearing stiffness, decreases dynamic stability of the system. Influence of clearance on properties of bearings also is analyzed in works [3, 4]. One can come to conclusion that without knowledge of clearance between the journal and pad, though accuracy of journal rotation is measured, it is difficulty to solve about work quality possibilities of bearings [5, 6].

2. System of automatic regulation

For solving the problem of clearance control the patent application of multi pad self-aligning automatically regulated bearing was proposed by us [7]. Fig. 1 shows the scheme of automatically controlled three pad bearing. The self-aligning pads *1* are placed on pins around the revolving journal *2* in a body *3*. Pin *4* is conventional, fixed in body *3* by the screw and counter nut. Pin *5* is assembled on balls *6* with possibility of axial motion in body *7*. For that purpose from lower side pin *5* supports to dish type spring *8*, from the upper side it by rolls *9* supports to wedge *10*, from the other side wedge *10* by rolls *11* is rested on self-aligning support *12*. The wedge *10* is driven by servo drive *13* (e.g. by pjezoelectric or other actuators, which are widely used as driving elements [8, 9]). At forward motion of wedge pin *5* moves down decreasing the clearance between pads *1* and journal *2*, and tightening the spring *8*, at backward motion of wedge spring *8* picks up the pin *5* increasing clearance between pins and journal. The hydrodynamic force acting in the bearing is measured by pin *14* which by thread is connected with bushing *15* by thread assembled in body *3*. For measuring purpose the pin *14* is hollow inside, and the measuring stick *16* is assembled in it. The stick *16* is riveted in the head of pin *14*, the head is connected with the pin body by the neck eccentric to axis of pin by value *e* (Fig. 1). Because of eccentricity the neck at action of axial force bends, the bending displacement is enlarged by stick *16* and transmitted to transducer *17*. Transducer *17* is electronically connected with the clearance control system, which by control the axial position of pin *5* keeps the set value of hydrodynamic force acting between the pads and journal. Assembly of pin *14* in additional bushing *15* is necessary for the reason that at assembly of journal *2* in two bearings (one in the front part of body *3*, the other in the rear part of body) it is necessary to keep together an accurate position of journal axis in the body *3* and to keep accurate position direction of eccentricity *e*: it must be in the imaginable plane of action of resulting hydrodynamic radial and tangent forces loading the pad, it is perpendicular to the journal axis. If eccentricity *e* will be turned in accordance with that plane, the radial force will be measured with an error. For this case the pin *14* at its upper side has two cut plane sides *18*, direction of eccentricity *e* and axis of transducer *17* are parallel to these sides (Fig. 1).

Meaning of other parts of bearing design (Fig. 1) there are not explained because it is understandable from the drawing: there are counter nuts for fixing of set position of pins, connecting elements of servo drive *13* with the wedge *10*, etc. Only one support with the pin *5* is used for regulation because at three pads *1* the self centering of pads is achieved.

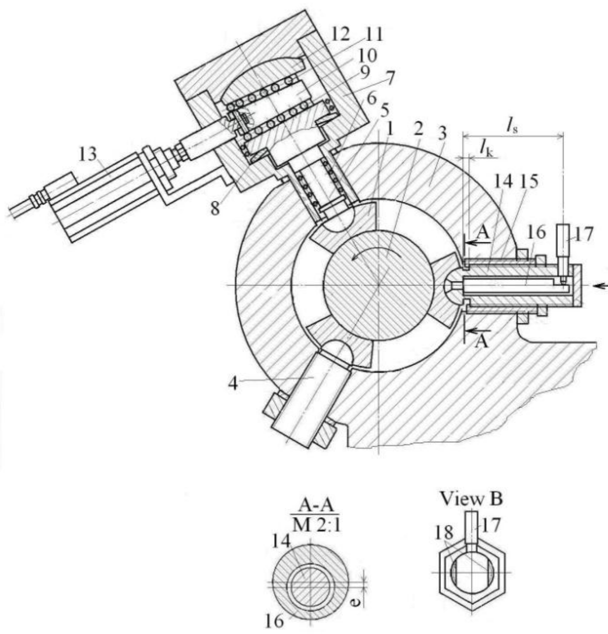

Fig. 1 Scheme of hydrodynamic bearing with automatic control

3. Analysis of bearing properties

At design of the bearing it is desirable to find the more sensitive one which would give the biggest elastic displacement of stick *16* measured by transducer *17* and together with it would guaranty enough strength of the pin which could keep the load of hydrodynamic force produced at revolving of the journal *2*. For that reason it is necessary to know the maximal load which can be created by the hydrodynamic force and strength of pin neck. The hydrodynamic radial force of oil film wedge F_0 is expressed by equation [10, 11]

$$F_0 = 5.1\times10^{-11}\mu n D B^2 L C_L / \Delta^2 \text{ , N} \tag{1}$$

where μ is oil viscosity coefficient in centipoises (cP); n is revolving frequency of the journal rev/min; D, B, L are diameter of the journal, width and length of the pad accordingly; C_L is coefficient, $C_L = \dfrac{1.25}{1+(B/L)^2}$; Δ is value of clearance between the journal and the pad, in mm. Diameter of the pin head sphere of such a bearing is d_s = 24 mm.

Lubrication oil for three pad journal bearings at large revolving frequency of a journal in common is accepted with very small viscosity, approximately 4 cP at temperature 55°C. It coincides with the lubricant viscosity ISO 6, a little below the Shell Tellus oils Do 10 lubricant. Such is the oil И5А produced in Ukraine. Further calculations we will do on the ground of that oil. Let for calculations we will accept the bearings of cylindrical grinder which has three pad bearing with journal diameter D = 70 mm and pad dimensions B = 36, L = 55 mm, number of journal revolutions 3000 rev/min. Fig. 2 shows dependence of force F_0 on clearance Δ for such a bearing. Calculated value of force F_0 at clearance Δ = 10 μm is 26722 N.

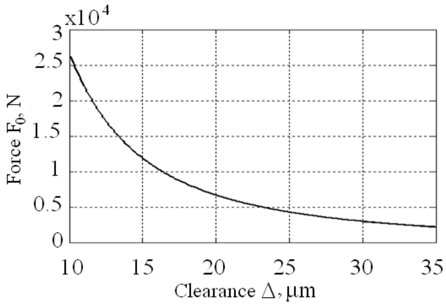

Fig. 2 Dependence of force F_0 on the clearance between the pad and journal

Fig. 3 shows the scheme of cross-section of the neck eccentric to internal hole of the pin (stick *16* is not shown there).

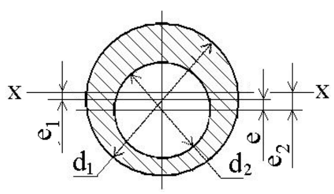

Fig. 3 Eccentric neck of the measuring pin

At eccentric position of an external diameter d_1 of the neck according to an internal hole d_2 the neutral axis x-x of inertia moment of the neck cross-section will be displaced upward to value e_1 from axis of symmetry of external diameter d_1 and to value e_2 from the axis of symmetry of internal diameter d_2. Eccentricity of external diameter according to internal diameter is equal e. At marking the area moment of inertia of external diameter d_1 to upper side from axis x-x by letter I_1, moment of inertia of internal diameter d_2 by I_2, moments of inertia of cross-section d_1 and d_2 from axis x-x to axis of symmetry of these cross-sections by I_3 and I_4 accordingly, and area moments of inertia of lower halves of circles d_1 and d_2 by I_5 and I_6, one can see that common area moment of inertia of the first part of cross-section placed over the axis x-x is equal $I_{c1} = I_1 - I_2$, while of the cross-section placed under the axis x-x is equal $I_{c2} = I_3 + I_5 - I_4 - I_6$. Diameters d_1 and d_2 and eccentricity e are known from the pin design. Eccentricity e_2 is equal $e_2 = e_1 + e$, so there is left not known eccentricity e_1. Because

$$I_{c1} = I_{c2} \tag{2}$$

by insertion of values from I_1 to I_6 into Eq. (1) it is possible to find the value e_1, necessary for the calculation of area moment of inertia of the neck cross-section.

It is possible to write that moments of inertia from I_1 to I_6 are equal

$$\left.\begin{aligned}
I_{1(2)} &= \frac{r_{1(2)}^4}{4}\left(arccos\frac{e_{1(2)}}{r_{1(2)}} - \left(2\frac{e_{1(2)}^2}{r_{1(2)}^2}-1\right)\frac{e_{1(2)}}{r_{1(2)}}\sqrt{1-\frac{e_{1(2)}^2}{r_{1(2)}^2}}\right)\\
I_{3(4)} &= \frac{r_{1(2)}^4}{4}\left(\begin{aligned}&\frac{\pi}{2} - arccos\frac{e_{1(2)}}{r_{1(2)}} + \\ &+\left(2\frac{e_{1(2)}^2}{r_{1(2)}^2}-1\right)\frac{e_{1(2)}}{r_{1(2)}}\sqrt{1-\frac{e_{1(2)}^2}{r_{1(2)}^2}}\end{aligned}\right)\\
I_{5(6)} &= \frac{\pi r_{1(2)}^4}{8} + \frac{\pi e_{1(2)} r_{1(2)}^3}{4} + \frac{\pi e_{1(2)}^2 r_{1(2)}^2}{2}
\end{aligned}\right\} \tag{3}$$

there $r_{1(2)}$ are radiuses of diameters d_1 or d_2 respectively; $e_{1(2)}$ are their eccentricities according to axis x-x.

Common area moment of inertia I_c of the neck cross-section according to axis x-x is equal

$$I_c = \frac{\pi}{4}\left(r_1^4 + e_1 r_1^3 + 2e_1^2 r_1^2 - r_2^4 - e_2 r_2^3 - 2e_2^2 r_2^2\right) \tag{4}$$

Because the pin is loaded by the load going through the pin axis and this axis coincides with the hole axis it is necessary to find not coincidence of axial load going through e_2 with the neutral axis x-x. Inertia moments from I_1 to I_4 are expressed by complicated equations, for that reason value e_1 was calculated by the method of approximation.

The maximal strain σ received in the neck of a pin and angular deflection of the pin head are evaluated accordingly by the equations

$$\left.\begin{aligned}\sigma &= \frac{F_0}{\pi\left(r_1^2 - r_2^2\right)} + \frac{F_0\left(r_1 - e_1\right)}{I_c} \\ \theta &= \frac{F_0 e_1 l_k}{I_c}\end{aligned}\right\} \tag{5}$$

there l_k is the neck length.

Displacement δ at the point measured by transducer *17* will be proportional to length ratio of measuring stick l_s to neck length l_k , it is l_s/l_k. Because θ is proportional to l_k and deflection measured by the transducer depends on l_k one can go to conclusion that length of the neck l_k does not influent on deflection measured by the transducer, but length l_s.

Table shows dependence between neck diameters d_1, d_2, e_1, strain σ at eccentricity e = 2 mm and load of the pin with force F_0 = 26722 N, cross-section area A in mm^2, and neck stiffness c, N/μm; displacement δ is for load 1 N.

Table
Dependence of e_1, strain σ, and measurement displacement on neck diameters

d_1, mm	d_2, mm	e_1, mm	σ, MPa	δ, μm	A, mm^2	c, N/μm
16	11	3.19	261	4.8×10^{-4}	106	2.25×10^{7}
15	10	2.84	283	5.4×10^{-4}	98	2.43×10^{7}
14	9	2.48	310	6.1×10^{-4}	90	2.65×10^{7}

As it is seen, such diameters of the neck assure necessary strength of the neck. Apart of that, modern sensors can measure displacements in nanometer limits. In such a case the measurement method enables to measure load on a pin *14* (Fig. 1) in limits of 1-10 N. At increase of diameter d_1 measurement sensitivity decreases, for that reason for proposed dimensions of the bearing Fig. 1 the neck diameters and eccentricity e should be got of that kind.

Force F_0 shows the hydrodynamic force which acts on the pins at journal rotation. If the journal is at center of the bearing, it is from all pad directions is acted by the same force and is in equilibrium state, the load carrying force (F_3) comprises only at dislocation of the journal from its equilibrium (centered) position to eccentricity e_j. This force for a three-pad journal bearing is expressed by the equation

$$F_3 = F_0\left(\frac{1}{\left(1-0.5\chi\right)^2} - \frac{1}{\left(1+\chi\right)^2}\right) \tag{6}$$

or stiffness c_j of lubricant hydrodynamic wedge film is $c_j = F_3 / e_j$, N/μm. There $\chi = 2e_j / \Delta$, where Δ at this case is in μm.

Fig. 4 shows the dependence of force F_3 on value Δ at the eccentricity e_j = 1 μm. It is seen that at smaller values of Δ carrying force of the bearing smartly increases. The stiffness c_c of the contact between the spherical pin head and the pad sphere can be expressed by the equation

$$c_c = \frac{9.81 d_s^2}{16 k_s} \tag{7}$$

where d_s is diameter of the sphere; k_s is a coefficient, k_s = 0.5 mm^{2}μm/N. Assuming d_s = 24 mm, we obtain c_c = 706 N/μm.

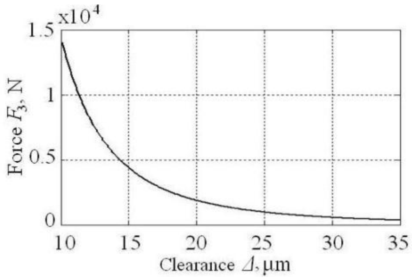

Fig. 4 Dependence of load carrying force on clearance in a bearing at displacement of the journal from equilibrium center to 1 μm

The common stiffness c_{com} of the pin head contact and the head neck can be found from the equation $\frac{1}{c_{com}} = \frac{1}{c_c} + \frac{1}{c_p}$ and for the neck with d_1 = 16, d_2 = 11 because of its high stiffness the common stiffness leaves practically the same, in the limits c_{com} = 705 N/μm. At not automatically controlled bearing the third pin would also be of the same stiffness. At load with the force F_0=26722 N elastic deflection in mechanical contacts of the bearing would be 37.9 μm. If to assemble the bearing with such interference (that at revolving it could work with the clearance 0.01 mm), after stoppage of rotation the pins will remain prestressed with interference of 27.9 μm, or the interference force will be F_{in} = 19670 N, the journal could not start revolving for such interference. For that reason the bearing could not work with the clearance of 0.01 mm at rotation.

Fig. 5 shows the minimal value of set clearance Δ with which the bearing at revolving of the journal could work keeping initial interference force after stoppage (the same – at starting) equal zero (F_{in} = 0). This force is defined by the equation

$$F_{in} = F_0 - c_{com}\Delta = 0 \tag{8}$$

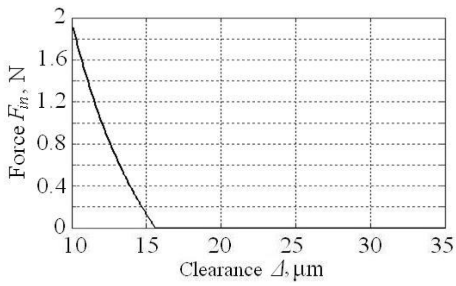

Fig. 5 Dependence of force F_{in} on set clearance Δ

The minimal clearance which is possible to achieve without initial interference at not revolving journal is Δ = 0.0156 mm. One must keep in mind that this clearance must be kept at cold bearing. At its heat, e.g. if temperature between the bearing body and the journal would change to 1°C, at the radius of external diameter of the body equal to 90 mm, the clearance would change to 1.1 μm. After stoppage of heated system a new start of it would be with an interference force. It influences hardly on the bearing wear.

The use of automatically controlled bearing enables to set necessary clearance between the pads and the journal at its start and at work and in such a way to achieve necessary working conditions of the bearing. Research of the bearing with controlled work conditions would give better illustration of its work accuracy dependencies (revolving accuracy, damping and vibrations properties, etc.) than at "blind" research, when the hydrodynamic force conditions in the bearing are not known. Apart of that, it is possible to show, that control of the clearance in a bearing at start, at idle run of the journal and at work enables to control power losses and heat of the bearing.

3. Conclusion

Analysis of the possibilities of a new design of hydrodynamic tilting pad journal bearing with self-aligning pads shows that by control of the clearance between the journal and pads it is possible to keep in front defined conditions of the bearing operation, increase its stiffness and at its research to receive the picture, better showing dependencies of functioning quality of the bearing on its real state.

References

1. **Marcinkevičius, A.H.** 1995. Stabilization of Grinding Processes and Machine Tool Work, Vilnius, Technika. 312 p. (in Lithuanian).
2. **Hargreaves, D.J.; Fillon, M.** 2007. Analysis of a tilting pad journal bearing to avoid pad fluttering, Tribology International 4 (40): 607-612. http://dx.doi.org/10.1016/j.triboint.2005.11.019.
3. **Reddy, D.S.K.; Swarnamani, S.; Prabhu, B.S.** 2000. Thermoelastohydrodynamic analysis of tilting pad journal bearing – theory and experiments, Tribology Transactions 1(43): 82-90.
4. **Hori, Y.** 2006. Hydrodynamic Lubrication, Tokio, Springer Verlag, 360 p.
5. **Vekteris, V.; Čereška, A.: Jurevičius, M.** 2010. Diagnostic research of adaptive hydrodynamic segmental bearings with elastic ring in a rotor system, Mechanika 2(82): 58-62.
6. **Vekteris, V.; Mokšin, V.** 2010. The modular principle of design of adaptive film lubrication bearings, Mechanika 4(84): 53-57.
7. **Marcinkevičius, A.H.** 2010. Multi-pad self aligning automatically regulated bearing, Patent Application No 2010107, Vilnius, 6 p. (in Lithuanian).
8. **Minchenia, V.T.; Stepanenko, D.A.; Zimmerman, C.K.; Bansevičius, R.; Bubulis, A.; Kundu, P.; Gishkeliuk, I.A.** 2010. Simulation of vibrational piezoelectric actuators of the micro-robot using the finite element method, JVE Journal of Vibroengineering 12(3): 314-319.
9. **Patašienė, L.; Ragulskis, K.; Federavičius**, **A.** 2010. Development and experimental analysis of layered actuators, JVE Journal of Vibroengineering 12(3): 333-339.
10. **Marcinkevičius, A.H.** 2010. Hydrodynamic tilting pad journal bearing with automatic control, Warshaw, The Archive of Mechanical Engineering 4(57): 343-354.
11. **Reshetov, D.N.** 1972. Parts and Mechanisms of Metal Cutting Machine Tools, Vol. 2, Moscow, Maschinostroenye: 520 p. (in Russian).

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AUTOMATINIS TARPELIO REGULIAVIMAS ATRAMINIŲ ĮDĖKLŲ VELENO GUOLYJE

R e z i u m ė

Straipsnyje aprašomas hidrodinaminis atraminių įdėklų veleno guolis su automatiškai valdomu tarpeliu tarp veleno ir įdėklų. Tarpelis gali būti valdomas, matuojant hidrodinaminę jėgą, veikiančią įdėklo atraminį pirštą. Guolio darbo analizė parodo didesnes automatiškai valdomų guolių galimybes. Surenkant tokio tipo nevaldomus guolius neįmanoma užtikrinti mažo tarpelio tarp veleno ir guolio, kai velenas sukasi dideliu dažniu, nes, jei velenas būtų statinės būsenos, t. y. nesisuktų, guolio įdėklai užspaustų veleną didele įvaržos jėga ir tokiomis sąlygomis jo negalima būtų paleisti suktis. Hidrodinaminių jėgų, veikiančių tarp įdėklų ir veleno, valdymas įgalina padidinti tarpelį (tiksliau – sumažinti įvaržą tarp veleno ir įdėklų iki nulio) veleno paleidimo suktis metu ir reguliuoti tarpelį velenui sukantis.

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AUTOMATIC REGULATION OF CLEARANCE IN A TILTING PAD JOURNAL BEARING

S u m m a r y

Hydrodynamic tilting pad journal bearing with an automatic control of the clearance between the journal and pads is described in the paper. The clearance can be controlled by measuring the hydrodynamic force acting on the supporting pin of the pad. Analysis of operation of the bearing shows higher possibilities of bearings with automatic control. At assembly of that type of not controlled bearing it is impossible to achieve small clearance between the pad and the journal at journal rotation with high frequency because in the case of static state, it is when the journal is not rotating, the bearing pads should grasp the journal with high interference force, and the bearing could not start working at such conditions. Control of hydrodynamic forces acting between the pads and the journal enables to regulate the clearance at starting and at operation.

Keywords: hydrodynamic bearing, clearance regulation, force measurement stiffness calculation.

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