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*Research Article*

# Automatic Control of Loading Forces in a Tilting Pad Journal Bearing

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Hydrodynamic three-tilting-pad journal bearing is analyzed. It is shown that a carrying force, stiffness of the bearing, and heat generated at journal rotation highly depend on a clearance between the journal and pads. The clearance can be controlled by measurement and control of hydrodynamic force at journal rotation. The bearing with such possibilities is described in the paper. Control of the hydrodynamic force prevents overheating of the bearing at large frequency of journal rotation, keeps necessary stiffness at small frequency of rotation, enables controlling forces loading the journal. Such a bearing is more versatile than being not controlled. Analysis and calculations are presented in the paper.

## 1. Introduction

Hydrodynamic tilting pad journal bearings for their many advantages are widely used in contemporary machines whose rotors are working for a long time without stoppage. Such machines are turbines, compressors, pumps, air blowers, grinding and boring machine tool spindles, and so forth. The advantage of these bearings is that at suitable design and selection the wear of the bearing tilting pads is only at start and stoppage of the rotor, the bearings may be distinguished by efficient damping of vibrations, high load capacity, and very high journal revolving speeds, which are hardly achievable by rolling bearings. At use for machine tool spindles the bearings show high stiffness and accuracy of rotation and work. Although achieving of all these properties in one example of the bearing is limited by a clearance between the journal and pads and by heat generated by journal rotation, in all cases of standard bearings the minimal clearance between the journal and pads is selected in accordance with the maximal speed of journal rotation, because in other cases too small clearance at the maximal speed would lead to increased temperature of lubricating oil film which can destroy the lubricating film and together with it the bearing. For that reason the standard tilting pad journal bearings are made with the warranted averaged clearance between the journal and pads enabling smooth rotation of the journal and maintaining some reasonable stiffness of a junction between the journal and pads. Increased clearance at low frequency of rotation will decrease the stiffness of the bearing. For example, the Kingsbury leading edge groove hydrodynamic bearings [1] are made with standard values of 0.25 preload and 0.0015 units of clearance per unit diameter, and the same clearance is provided from low and up to 105 m/s revolving speeds of the journal. One can see that the stiffness and rotation accuracy of the journal are not secured in all areas of possible journal speeds. For some designs of bearings it is foreseen [1] that at slow rotation of the journal and at start of rotation the bearing can be lubricated under high oil pressure and work as a hydrostatic bearing. Somewhat similar to the Kingsbury bearings are Orion standard tilting pad bearings [2] which, differently from the Kingsbury bearings of one direction of rotation, are reversible. Their maximal clearances are bigger than these of Kingsbury bearings.

The clearances of all these and alike bearings are bigger than desirable for achieving the highest stiffness and rotation accuracy which by [3] for the journal diameter near to 70 mm is about 6 $\mu$m. The author does not analyze dependency of allowable clearance on revolving speed of the journal or temperature conditions in the clearance, although these are the main factors influencing the possibilities of a bearing work.

If to speak about the maximal allowable clearance, research of [3] has showed that at trespassing of some large limits of clearance the amplitudes of rotor vibrations can increase almost to ten times. Friction surfaces of the pads are damaged. The reason of this from the point of view of hydrodynamic lubricating oil film layer is not analyzed in the paper.

In the papers [4, 5] it was shown that at work of bearings without preload and with a bigger clearance vibrations are excited. On the other hand, the authors [6–10] in their works have showed that temperature deflections at starting of the journal and at settled work may significantly change the clearance between the journal and pads and as a result it two different outcomes are possible: the rapid seizure of the journal and pads or the safe running. The increased clearance is necessary for preventing of the seizure, but it will lead to stiffness and work accuracy decrease of the bearing. One can come to conclusion that the automatically regulated clearance could solve the task. Automatic or hand regulating of the clearance for bearing stiffness change is proposed by [11, 12]. The clearance there is regulated by change of pads position in a bearing body or in a journal of special design by pads elastic displacement. The hydrodynamic forces acting in a bearing there are not controlled, but the stiffness may be changed by acknowledging in front dependence between the clearance and stiffness. The process is performed after heat stabilization in a system. Regulating of bearing stiffness at a start of journal rotation and at its operating r.p.m. is also performed by the system [13], but there also the operating conditions cannot be controlled straightly, but by the oil pressure control in one point of a pad. Pad position control can be used not only for regulating the stiffness, but also for dampering of journal vibrations. Patent [14] is proposed for it. There, the oil with changing pressure in the clearance between the journal and pad at journal vibrations is fed to a chamber between the pad and adapter, keeping the pad and changed pressure in the chamber stabilize journal rotation. Similar to it is the work [15] where vibrations are reduced by controlling the pressure of oil injection into the gap between the pad and journal. The acting hydrodynamic forces and real value of clearance in the bearing there are not measured and controlled.

Character for machine tool spindles is that their rotation speeds can vary in large limits and in all these cases the task is to achieve the highest stiffness and rotation accuracy, because on it depends the quality and accuracy of work pieces machined on these machines. Such are grinding wheel spindles of grinding machines, work spindles of cylindrical grinders, the spindles of boring machines, and so forth.

Dependence of the bearing stiffness, lubricating oil film temperature between the pad and journal, and carrying capacity of the journal are analyzed in this paper. Analytical calculation of hydrodynamic lubrication indices is performed on the grounds of theoretical data, presented in [16]. The solid theoretical and experimental research work of many investigators was used for receiving these dependency. Calculation results, shown in the paper, are for the three-pad journal bearing; the journal is of steel with the carbon content $C \leq 0.5\%$ and its diameter is $D = 70$ mm; the pads are of steel with the bronze antifriction layer; the pad width $B$ is 36 mm, length $L$ is 55 mm, and thickness $l_1$ is 16 mm. Preload of the bearing is accepted 0.2.

We have proposed the automatically controlled tilting pad journal bearing with an automatic control of the clearance between the journal and pads [17, 18]. It enables controlling the stiffness and protecting hydrodynamic film from destruction as a result of thermal deflection of bearing elements, to control the clearance at start of journal rotation, to smooth the start, and to control the radial component of the cutting force at the use of the bearing for machine tool spindles.

## 2. Design of the Bearing

Scheme of the bearing with an automatic control of the clearance between the spindle and pads is shown in Figure 1. Three self-aligning pads (1) of the bearing support the revolving journal (2) and are assembled in the body (3) of the spindle head. Pin (4) supporting the pad in radial direction is made hollow inside and the measurement stick (5) by its front end is fixed by riveting in a pin head; the other part of it is placed with the clearance in the pin hole so that it can freely turn to an angle inside the hole of pin (4). The pin neck is eccentric (Figure 1(b)) to its internal hole and at action of axial force the neck bends and turns the stick (5) to an angle. Turn value is measured by the transducer (6) and its signal is used to control the work of the bearing. Pin (4) is assembled in the bush (7) and they both work as a differential screw what enables keeping the axial plane $y$-$y$ of the neck of pin (4) (Figure 1(b)) perpendicular to the axis of the journal (2). Pin (4) and the bush (7) after their adjustment are fixed by the counter nuts (8). Because the transducer (6) measures deflection in direction, perpendicular to the axis of pin (4), the longitudinal deformation of the pin (4) and stick (5) as a result of their thermal deflection does not influence measurement results.

The movable pin (9) is used for an automatic regulation of the clearance in the bearing. For that purpose it is assembled in the body (10) on balls (11) and from one side it keeps the pad (1) and from the other side through rolls (12) it is supported to the movable wedge (13) which from the other side through rolls (14) supports the self-aligning prop (15); the later by its spherical top supports the flange (16). Wedge (13) is driven by the puller (17) connected with an automatic drive. Plate spring (18) is used for stretching in the system from pin (9) to prop (15) and for the motion of the pin (9) upward at wedge (13) movement to the left side (in the drawing). The third threaded pin (19), supporting the third pad, after adjusting, is hardly fixed in body (3) by the counter nut (20) with a sealer.

At use of the bearing for a machine tool spindle, for example, for cylindrical grinders, the radial cutting force is measured as a difference of previously set hydrodynamic force at an idle run of the spindle and its change at cutting.

The common moment of inertia $I_c$ of a neck cross-section (Figure 1(b)) according to the neutral axis $x$-$x$ (chord $x$-$x$) can be expressed by

$$I_c = \frac{\pi\left(r_1^4 + e_1 r_1^3 + 2e_1^2 r_1^2 - r_2^4 - e_2 r_2^3 - 2e_2^2 r_2^2\right)}{4}, \quad (1)$$

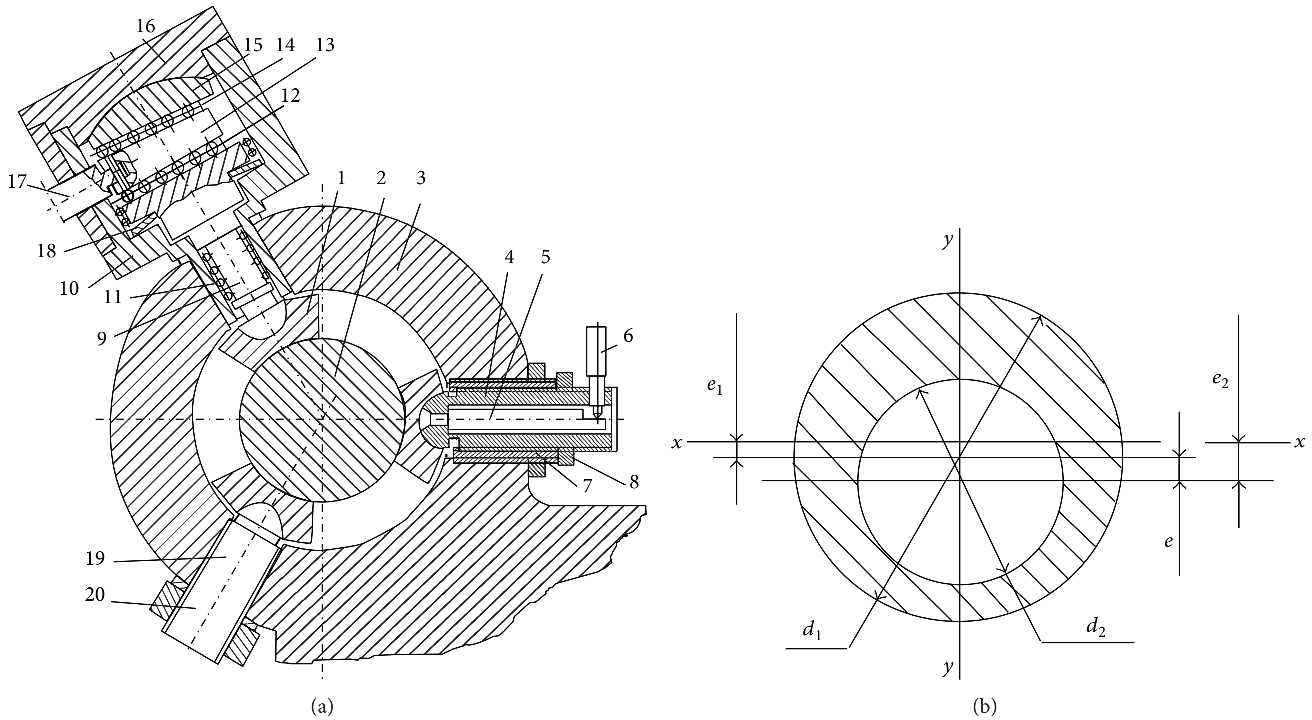



FIGURE 1: Automatically controlled bearing (a) and the cross-section of the eccentric neck of the measuring pin (b).

where $r_1$ and $r_2$ are radiuses of circle diameters $d_1$ and $d_2$; $e_1$ and $e_2$ are eccentricities of the circle diameters $d_1$ and $d_2$ to the chord $x$-$x$, respectively. There in Figure 1(b) the value $e$ is an eccentricity of diameters $d_1$ and $d_2$ one to the other. This value, the same as values $d_1$ and $d_2$, is set at production of the pin (4).

Apart from that because the inertia moment ($I_{c1}$) of a neck circle is segmented over the chord $x$-$x$ and under this chord ($I_{c2}$) is equal to

$$I_{c1} = I_{c2}, \quad I_{c1} = I_1 - I_2; \quad I_{c2} = I_3 + I_5 - I_4 - I_6, \tag{2}$$

where $I_1$ and $I_2$ are inertia moments of segments of diameter $d_1$ and $d_2$ over the chord $x$-$x$, $I_3$ and $I_4$ are inertia moments of circle parts of diameter $d_1$ at distance $e_1$ and of diameter $d_2$ at distance $e_2$ from the chord $x$-$x$; $I_5$ and $I_6$ are inertia moments of two half circles $d_1$ and $d_2$ with regard to the chord $x$-$x$, respectively.

Moments of inertia from $I_1$ to $I_6$ are equal:

$$I_{1(2)} = r_{1(2)}^4 \left[ \frac{\arccos e_{1(2)}}{r_{1(2)}} - \left(\frac{e_{1(2)}}{r_{1(2)}}\right)\left(\frac{2e_{1(2)}^2}{r_{1(2)}^2} - 1\right)\sqrt{1 - \frac{e_{1(2)}^2}{r_{1(2)}^2}}\right] \times \frac{1}{4};$$

$$I_{3(4)} = r_{1(2)}^4 \left[ \frac{\pi}{2} - \frac{\arccos e_{1(2)}}{r_{1(2)}} + \left(\frac{e_{1(2)}}{r_{1(2)}}\right)\left(\frac{2e_{1(2)}^2}{r_{1(2)}^2}\right)\sqrt{1 - \frac{e_{1(2)}^2}{r_{1(2)}^2}}\right] \times \frac{1}{4};$$

$$I_{5(6)} = \frac{\pi r_{1(2)}^4}{8} + \frac{\pi e_{1(2)} r_{1(2)}^3}{4} + \frac{\pi e_{1(2)}^2 r_{1(2)}^2}{2}. \tag{3}$$

Because the eccentricity $e$ is known from (1) and (2) it is possible to find the values $e_1$ and $e_2 = e_1 + e$ and the inertia moment $I_c$. For example, if to get in our case $d_1 = 16$, $d_2 = 11$, and $e = 1.5$ mm, it will be got that $e_2 = 3.58$ mm, $e_1 = 2.08$ mm, and $I_c = 2693\ \text{mm}^4$. Because $e_2 = 3.58$ mm, the load of 1 N will create the bending moment of 3.58 Nmm. At getting a neck length of the pin (4) equal to 5 mm and the length of the measuring stick (5) (see Figure 1(a)) equal to 90 mm, the axial load of 1 N on the pin (4) creates the angular deflection of the pin head to $1.133 \cdot 10^{-8}$ rad and the transducer (6) would measure the deflection of $1.02 \cdot 10^{-3}\ \mu$m. Contemporary transducers can measure deflections in limits of $0.01\ \mu$m, so the measurement method enables measuring the load on the pad (1) in Figure 1(a) in limits of 10 N.

The measurement method does not influence largely on change of bearing stiffness. At neck diameters $d_1 = 16$, $d_2 = 11$ mm the neck cross-section area is equal to 106 mm$^2$. At the pin neck length 5 mm the longitudinal elastic deflection of the pin will be $2.25 \cdot 10^{-7}$ mm/N or its stiffness will be $c_p = 4.45 \cdot 10^6$ N/mm.

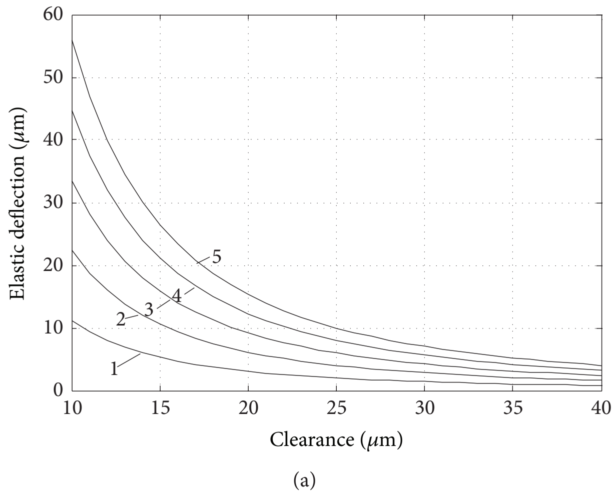



FIGURE 2: Dependence of calculated elastic deflection on the clearance and speed of journal rotation: (a) clearance from 10 to 40 $\mu$m and (b) clearance from 40 to 80 $\mu$m, speed in m/s for curves from 1 to 5: 1—4, 2—8, 3—12, 4—16, and 5—20.

Stiffness $c_c$ of a spherical pin head contact with the pad sphere can be expressed by

$$c_c = \frac{9.81 \cdot 10^4 d_s^2}{(16k_s)}, \tag{4}$$

where $d_s$ is the diameter of the sphere; $k_s$ is the coefficient, $k_s = 5\,\text{mm}^2\,\text{mm/N}$. Getting $d_s = 24$ mm, we will receive $c_c = 7.06 \cdot 10^5$ N/$\mu$m. Common stiffness of a pin head contact and head neck on the ground of equation $1/c_{\text{com}} = 1/c_c + 1/c_p$ can be found equal $c_{\text{com}} = 6.096 \cdot 10^5$ N/$\mu$m.

Common stiffness of the unmovable pin (19), keeping in mind that its sphere is connected with the body by a neck of 4 mm length and 20 mm diameter, is $6.773 \cdot 10^5$ N/mm.

The movable pin (9) apart from contact of the spherical pin head with the pad adds the deflection of a long neck of its pin, deflection of the rollers (12) and (14) with the planes of pin (9), wedge (13) and prop (15) and spherical contact of this prop with the sphere of the flange (16); its stiffness is equal to $3.7 \cdot 10^5$ N/mm.

## 3. Analysis of Hydrodynamic Lubrication Dependency on Bearing Work

At journal rotation the created hydrodynamic radial force $F_0$ of oil film wedge on the ground of data [16] is expressed by

$$F_0 = \frac{9.56 \cdot 10^{-7} \mu v B^2 L C_L}{\Delta^2} \ (\text{N}), \tag{5}$$

where $\mu$ is the oil viscosity coefficient in centipoises (cP), $v$ is the revolving speed of the journal in m/s, $B$, $L$ are the width and length of the pad accordingly, $C_L$ is the coefficient, $C_L = 1.25/(1 + (B/L)^2)$, and $\Delta$ is the value of diametrical clearance between the journal and pads, in mm.

Figures 2(a) and 2(b) show elastic deflections, which would be created at rotation of the journal in the bearing with

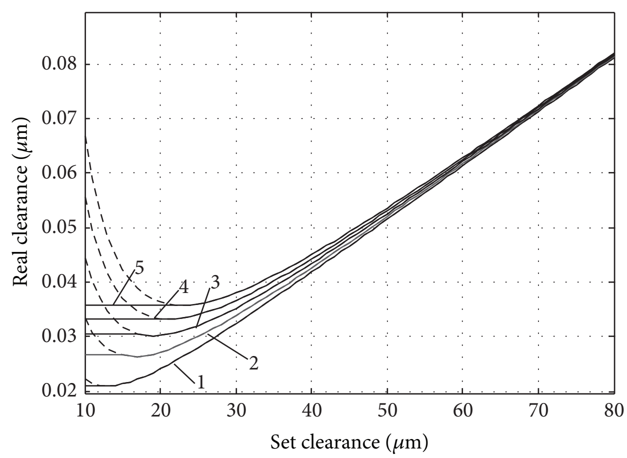



FIGURE 3: Dependence of the real clearance on the set clearance and on speed of the journal, speed in m/s for curves from 1 to 5: 1—4, 2—8, 3—12, 4—16, and 5—20.

all unmovable pins of the type (19) (Figure 1(a)) with their stiffness and at speeds from 4 to 20 m/s. Because the curves in the graphics sharply fall down and, as it is seen from Figures 2(a) and 2(b), for better view, the graphics are shown in two different scales and in Figure 2(a) deflections are shown for clearances from 10 to 40 $\mu$m, in Figure 2(b), for clearances from 40 to 80 $\mu$m.

One can see that such large elastic deflections at small clearance are impossible because at increase of elastic deflections the clearance would grow up, the increased clearance automatically would lead to the smaller hydrodynamic force, and, as a result, the clearance would decrease till some equilibrium would be got. For that reason in Figure 3 the sum of the set clearance and elastic deflections is shown from which it is seen that at the beginning all sums would fall down (engraved lines), and after that it stabilizes and later goes up. The clearances, showed by the engraved lines, are impossible: the real clearance at any small initially set clearance would

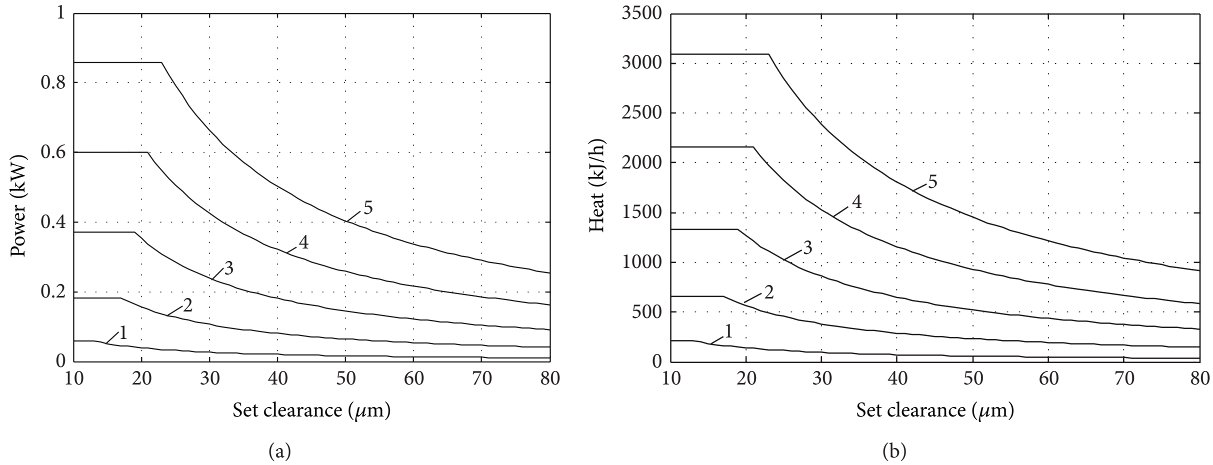



FIGURE 4: Dependence of the power (a) and heat generation (b) on revolving speed of the journal and on the set clearance, speed in m/s for curves from 1 to 5: 1—4, 2—8, 3—12, 4—16, and 5—20.

increase as a result of elastic deflections, and this clearance would stabilize at the level of horizontal lines, as it is shown in Figure 3, and only at further increase of the set clearance the real clearance would go up as it is seen from Figure 3. The smaller clearance than at stabilized level may be got only in case if the initial interference in the system between the pads and journal would be formed, which is unallowable, because the journal should start to revolve being clamped between the pads.

Such conclusions can be got from Figure 3. (1) The maximal value of a hydrodynamic force in all cases of speeds is limited dependent on elastic deflection increase by the loading force action. (2) Because the loading force very steeply falls at increase of the clearance and decrease of revolving speed, the better results and bigger versatility of the bearing work can be achieved at the possibility of clearance control.

Because the existence of phenomenon of an automatic clearance increase as a result of elastic deflections is evident, all other researches of dependency in our work were made with evaluation of such an increased clearance.

The power $P$ loss because of friction in a carrying lubricant layer for one pad is expressed by

$$P_1 = \frac{1.988 \cdot 10^{-3} \mu v^2 BLC_f}{\Delta} \text{ (kW)}; \qquad (6)$$

the heat $Q$ generated by that is expressed by

$$Q_1 = \frac{5.529 \cdot 10^{-10} \mu v^2 BLC_f}{\Delta} \text{ (kJ/h)}, \qquad (7)$$

where in both cases $C_f = 1 + 0.1C_L$; $C_L$ was shown earlier.

For three-pads-power $P$ and heat $Q$ will be tripled. Figures 4(a) and 4(b) show dependency of $P$ and $Q$ on revolving speed $v$ and clearance $\Delta$. Journal revolving speeds for all cases of curves from 1 to 5 are the same as for Figure 2—from 4 to 20 m/s accordingly.

Maximal temperature $T_x$ of a hydrodynamic lubricant layer between the journal and pad is calculated by

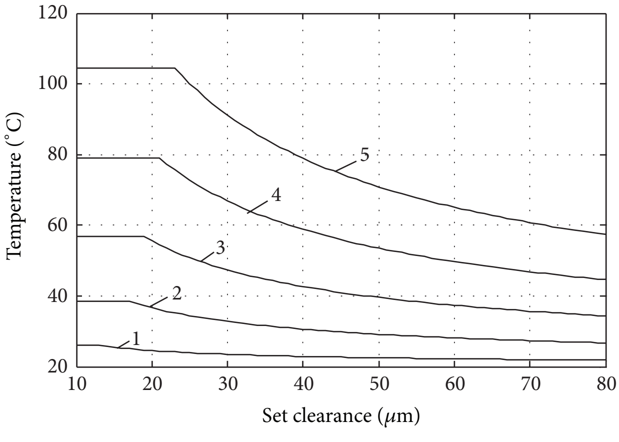
$$T_x = \frac{\lfloor Q + \left(K_p + K_j M_2 + Q_{\text{oil}}\right) T_m \rfloor}{\left(K_p + K_j + Q_{\text{oil}}\right)} \ (^\circ\text{C}), \qquad (8)$$

where $Q$ is the heat generated by the journal revolution, $K_p$ is the coefficient evaluating the part of heat carried out by the pads, $K_j$ is the coefficient, evaluating the part of heat, carried out by the journal, and $Q_{\text{oil}}$ is the part of heat carried out by the oil layer between the bearing and pads. On the grounds of [16] $Q_{\text{oil}} \cong 0.1(K_p + K_j)$,

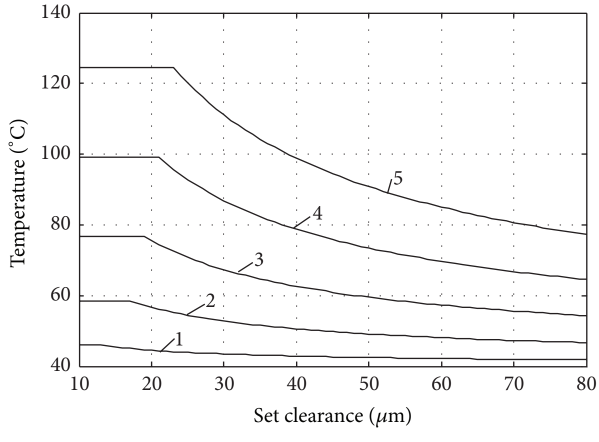
$$\frac{1}{K_p} = \frac{1}{\alpha_{m1} S_1 z} + \frac{1}{\sqrt{2(B+L)\,\alpha_{m2}\lambda_1 S_1}\,\left(B_2 + \text{th}(m_1 l_1)\right) z / \left(1 + B_2 \text{th}(m_1 l_1)\right)}, \qquad (9)$$

where $\alpha_{m1}$ is the coefficient depending on lubricating oil properties, the journal revolving speed $v$, and the diametrical clearance $\Delta$ between the pads and journal; for the journal of steel with the carbon content $C \le 0.5\%$ the speed $v > 2$ m/s, and the coefficient $\alpha_{m1} = 24.65 v^{0.2}/\Delta^{0.5}$ kJ/(m$^2$hK) ($\Delta$—in m), $S_1$ is the carrying pad surface area in m$^2$ (in our case $S_1 = 0.002$ m$^2$), $z$ is the number of pads (in our case $z = 3$), $\alpha_{m2}$ is the coefficient of heat transfer from not loaded surface of the pad to oil (in our case $\alpha_{m2} = 1880$ kJ/(m$^2$hK)), $\lambda_1$ is the coefficient of the pad material heat conduction in kJ/(mhK) (in our case $\lambda_1 = 209$ kJ/(mhK)), and $l_1$ is the thickness of the pad in m,

$$B_2 = \sqrt{\frac{\alpha_{m2} S_1}{\left[2(B+L)\lambda_1\right]}}; \qquad m_1 = \sqrt{\frac{\alpha_{m2} 2(B+L)}{(\lambda_1 S_1)}};$$
$$\frac{1}{K_j} = \frac{1}{(\alpha_{m1} S_3 + M_1)} + \frac{1}{(\alpha_{m1} S_1 z)}. \qquad (10)$$




(a)




(b)

FIGURE 5: Dependence of the maximal temperature of lubricating oil layer on the speed and clearance: (a) the inlet oil temperature is 20°C and (b) the inlet oil temperature after heat stabilization is 40°C; speed in m/s for curves from 1 to 5: 1—4, 2—8, 3—12, 4—16, and 5—20.

$S_3$ is the area of nonloaded surface of the bearing in $m^2$; we are getting the cylindrical surface only on the pad length $L$ = 0.055 m; all this area is equal to $S_3 = 0.0105\ m^2$; $M_1$ is the coefficient, $M_1 = 2\sqrt{\alpha_2\lambda_2 U_2 S_2}$, and $\alpha_2$ is the coefficient of heat transfer to surrounding media; in our case at speed $v > 2$ m/s $\alpha_2 = 55.3 + 5.85v$ (kJ/(m$^2$ hK)), $\lambda_2$ is the coefficient of journal metal heat conduction (in our case $\lambda_2 = 165$ kJ/(mhK)), $U_2$ is the perimeter of a heat conducting surface of the journal in m ($U_2 = 0.22$ m), $S_2$ is the area of a journal diameter cross-section in m$^2$ ($S_2 = 0.0154\ m^2$), $M_2$ is the coefficient, and $M_2 = \alpha_{m1}S_3/(\alpha_{m1}S_3 + M_1)$.

Our experiments [19] have showed that after starting of a journal rotation and artificial cooling of the lubricating oil by its flow through the radiator the oil temperature in time of approximately 1 h increases from 20°C near to 40°C and after that stabilizes at this level. For that reason Figures 5(a) and 5(b) show dependency of maximal temperature of the oil film layer between the pads and journal on the revolving speed $v$ and clearance $\Delta$ at an inlet temperature of the lubricating oil 20°C (at initial period of bearing work) and the inlet temperature 40°C (after stabilization of oil temperature). By the Kingsbury (2012), the maximal allowable oil outlet temperature can be near to 90–100°C. Let us get the maximal temperature 90°C. At such conditions at the inlet oil temperature 20°C the journal could work with maximal speed 20 m/s at an approximate clearance near to 38 $\mu$m (set clearance 30 $\mu$m, see Figure 5(a) and real clearance 38 $\mu$m, see Figure 3). At speed 16 m/s the journal could work at all set clearances because the maximal temperature there reaches only 80°C, Figure 5(a). For the other speeds (from $v$ = 4 to 12 m/s in Figure 5(a)) the temperature would not reach 60°C. At inlet temperature 40°C the journal at speed 20 m/s could work only at set clearance approximately 51 $\mu$m, see Figure 5(b) (real clearance near to 53.5 $\mu$m, see Figure 5(b)). Work with the speed of 16 m/s at these conditions can begin at the set clearance 27 $\mu$m, see Figure 5(b), and the real clearance approximately 35 $\mu$m, see Figure 2. For the curves from 1 to 3 allowable speeds can be used in all limits of the set clearance. Because the temperature and hydraulic pressure depend on the clearance and revolving speed, by controlling the hydrodynamic pressure it is possible to control the clearance, and temperature, not straightly, but indirectly by controlling the hydrodynamic force, which is made by our bearing. Influence of journal revolving speed on temperature is clearly shown in the work of [20] and many other works.

Depending on it the maximal stiffness of bearings and maximal carrying capacity can be defined. Carrying property $F_3$ for a three-pad journal bearing is calculated by

$$F_3 = F_0 \left[ \frac{1}{(1-0.5\xi)^2} - \frac{1}{(1+\xi)^2} \right] (\mathrm{N}), \tag{11}$$

where $\xi = 2e/\Delta$.

Stiffness of a carrying oil film wedge is defined by the equation $c_o = F_3/e$ (N/$\mu$m).

Figures 6(a) and 6(b) show dependency of the stiffness of the hydrodynamic lubricating wedge layer at eccentricity $e$ = 1 $\mu$m. There, the same as for Figure 2, because at bigger clearance the curves that are quickly falling down are going near to one another, the curves of dependency at clearance up to 40 $\mu$m (Figure 6(a)) are drawn in one scale, at the clearance from 40 to 80 $\mu$m (Figure 6(b))—in the other scale. Because Figures 5(a) and 5(b) show that at speeds from $v$ = 4 to 12 m/s at all values of clearances temperature do not overcome 90°C, at these speeds the bearing can work with the maximal stiffness from 1830 to 2020 N/$\mu$m. For the cases 4 and 5 (revolving speeds 16 and 20 m/s accordingly) if not to overcome the temperature 90°C, the stiffness is limited from approximately 1110 N/$\mu$m at the first case to 210 N/$\mu$m at the second case.

## 4. Results and Discussion

The main properties of the hydrodynamic tilting pad journal bearing, especially at its use for machine tool spindles, are high stiffness, prevention from overheating, and versatility

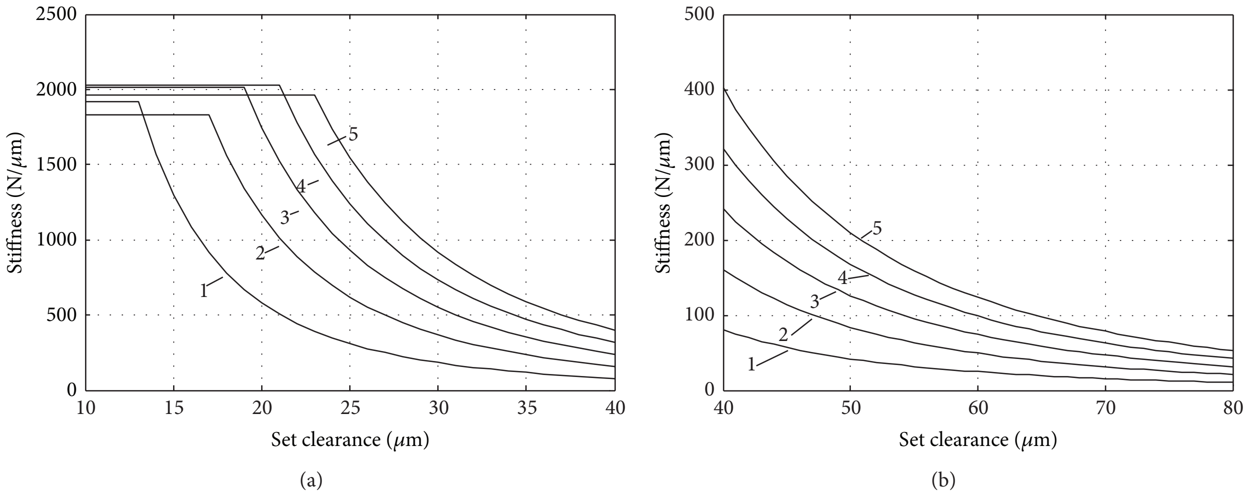



FIGURE 6: Dependence of the stiffness of a lubricant layer on the clearance and revolving speed of the journal: (a) one scale for the clearance up to 40 $\mu$m and (b) the other scale for the clearances from 40 to 80 $\mu$m, speed in m/s for curves from 1 to 5: 1—4, 2—8, 3—12, 4—16, and 5—20.

which means to secure these properties at different speed of journal rotation. Apart from that the necessary feature of the bearing, used for machine tool spindles, is a possibility to control the cutting force at machining. Such a three pad journal bearing is presented in the paper. Character for the bearing design is that the pin load is measured by an angular deflection of a pin neck; this deflection is transmitted with high multiplication to the transducer by the stick, turning inside the hole of the pin; such design prevents the influence of longitudinal thermal deflection of the pin or stick on measurement accuracy. Apart from that, the design warrants that the angular deflection would be in the plane, perpendicular to the axis of journal which also is necessary for the accurate measurement of the force.

For showing the possibilities of this bearing the analysis of properties of hydrodynamic tilting pad bearings was made. Analytical calculations showed dependency of the stiffness, heat generation, and temperature of the lubricating film at journal rotation. On the ground of this analysis it is possible to define what loading hydrodynamic forces must be kept by an automatic control of the bearing in purpose to achieve the highest stiffness and allowable heat generation.

From the point of view of hydrodynamic bearings the paper pointed out the phenomenon that at small set clearance in statics between the pads, and journal the elastic deflections in a junction of pads and pins at journal rotation markedly increase the clearance. It decreases the hydrodynamic force. For that reason the equilibrium is achieved when the common sum of the set clearance and elastic deflection is constant. Such a state exists in some limits of clearance value. We did not meet the emphasis of it in other publications.

## 5. Conclusions

(1) Because the work possibilities of a tilting pad journal bearing are markedly limited by the temperature generated at revolving of a journal, the standard bearings are made with such increased values of clearances between the journal and pads which would warrant the allowable maximal temperature at the maximal revolving speed of the journal. At smaller speeds as a result of such increased clearance the lifting hydrodynamic force of the bearing strongly decreases and the bearing can lose right performance.

(2) Minimal achievable clearance and maximal hydrodynamic lifting force in a bearing are limited by the sum of the set clearance and its increase as a result of an elastic deflection created by the hydrodynamic force. Some total values of equilibrium of the set clearance plus elastic deflections are kept in some limits: the small set clearance raises the hydrodynamic force, but as a result elastic deflections increase and it induces decrease of hydrodynamic force. The equilibrium is kept only in some limits of the set clearance. At further increase of that clearance the hydrodynamic force falls down together with decreased elastic deflection, stiffness of the bearing also sharply decreases, and finally the hydrodynamic force and stiffness can reach the limits, at which the bearing would be not able to work properly.

(3) The bearing with an automatic control of the clearance can keep such hydrodynamic forces which would warrant the necessary temperature, keep maximal allowable stiffness at good conditions, and control the external force, loading the journal. At its use for machine tool spindles, measurement of the radial cutting force enables controlling the technological process. Cutting forces are controlled by comparing the previously defined hydrodynamic forces at idle work and at machining.

## Nomenclature

$B$: Width of the pad (mm)
$L$: Length of the pad (mm)

$P$: Power loss because of friction in the carrying lubricant layer (kW)
Q: Heat generated by the journal friction in the lubricant layer (kJ)
$e$: Eccentricity of the external and internal diameters of the measuring pin neck cross-section one to the other (mm)
$v$: Revolving speed of the journal (m/s)
$B_2$: Coefficient
$c_c$: Stiffness of the spherical pin head contact with the pad sphere (N/mm)
$c_{\text{com}}$: Common stiffness of the pin neck together with its contact with the pad sphere (N/mm)
$c_p$: Stiffness of the pin neck of 5 mm length (N/mm)
$c_0$: Stiffness of the lubricant layer between the journal and pads (N/mm)
$C_f$: Coefficient
$C_L$: Coefficient
$d_1, d_2$: External and internal diameter of the pin neck cross-section (mm)
$d_s$: Diameter of the contact sphere between the pin head and pad (mm)
$e_1, e_2$: Eccentricities of the external and internal diameters to the neutral axis of moment of inertia of the neck cross-section (mm)
$k_s$: Coefficient of a contact displacement at the contact of a pin head with pad ($\text{mm}^2$ mm/N)
$r_1, r_2$: Radiuses of external and internal diameters of the pin neck cross-section (mm)
$F_0$: Hydrodynamic force generated by the journal rotation (N)
$F_3$: Carrying property of the three-pad journal bearing (N)
$I_{c1}, I_{c2}$: Moments of inertia of two halves of neck cross-sections according to the neutral axis ($\text{mm}^4$)
$I_3, I_4$: Moments of inertia of the circle part of diameter $d_1$ at the width $e_1$ and of diameter $d_2$ at the width $e_2$ to the neutral axis of the moment of inertia ($\text{mm}^4$)
$I_5, I_6$: Moments of inertia of the two half circles $d_1$ and $d_2$ with regard to the neutral axis of inertia ($\text{mm}^4$)
$K_j$: Coefficient of the heat conduction from the revolving journal to oil (kJ/($\text{m}^2$hK))
$K_p$: Coefficient of the heat conduction from the pad to oil (kJ/($\text{m}^2$hK))
$l_1$: Thickness of the pad (m)
$m_1$: Coefficient (1/m)
$M_1$: Coefficient (kJ/(hK))
$M_2$: Coefficient (kJ/(hK))
$Q_{\text{oil}}$: Heat carried out by the lubricating oil layer between the bearing and pads to oil (kJ/($\text{m}^2$hK))
$S_1$: Area of the carrying surface of the pad ($\text{m}^2$)
$S_2$: Area of the journal diameter cross-section ($\text{m}^2$)
$S_3$: Area of the external not carrying surface of the pad ($\text{m}^2$)
$T_x$: Maximal temperature of the hydrodynamic lubricant layer between the journal and pads (°C)
$U_2$: Perimeter of the heat conducting surface of the journal (m)
$z$: Number of the bearing pads
$\mu$: Coefficient of the oil viscosity (cP)
$\Delta$: Diametrical clearance between the journal and pads (mm)
$\alpha_{m1}$: Coefficient of the heat transfer from the journal to surrounding oil (kJ/($\text{m}^2$ hK))
$\alpha_{m2}$: Coefficient of the heat transfer from the not loaded surface of the pad to oil (kJ/($\text{m}^2$ hK))
$\lambda_1$: Coefficient of the pad material heat conduction (kJ/(mhK))
$\lambda_2$: Coefficient of the journal metal heat conduction (kJ/(mhK))
$\xi$: Coefficient.

## Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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