UDC 621.891 (043.3) DOI: 10.18372/0370-2197.2(107).20142

O.V. BRESHEV[1], O.V. BASHTA[1], P.L. NOSKO[1], A.O. BASHTA[2], O.M. SPIVAK[3]
I.V. SEMAK[1]

[1]State University "Kyiv Aviation Institute", Ukraine
[2]National University of Food Technologies, Ukraine
[3]National Transport University, Ukraine

DEVELOPMENT OF A DESIGN SCHEME OF A SINGLE-SUPPORT DRIVE SYSTEM WITH AEROSTATIC BEARINGS

This study introduces an enhanced computational model for a single-support spindle system equipped with aerostatic bearings, intended for use in semi-automatic monocrystal cutting machines. The model addresses critical geometric and operational features specifically, a low relative bearing length ($\lambda < 0.5$) and conical support surfaces which are not adequately considered in conventional calculation methodologies. To improve modeling accuracy, the original bearing geometry is transformed into an equivalent radial configuration, allowing adaptation of an existing method. The approach includes the assessment of radial displacements and evaluates the influence of gas-lubricant parameters, supply pressure, and stiffness coefficients. The proposed model enables more precise estimation of the bearing's load-carrying capacity and stiffness, thereby enhancing the operational stability and performance of the spindle unit. The findings emphasize the importance of accounting for specific geometric deviations in bearing analysis.

Keywords: *aerostatic bearing, spindle unit, single-support system, gas lubrication, radial displacement, lifting force, conical support, contactless drive.*

Introduction. The calculation scheme involves identifying and taking into account those features of the drive that have a crucial effect on its properties, their non-transformation, qualitative and quantitative assessment, and, at the same time, abstracting from insignificant features [1].

The aerostatic bearing of the considered single-support system of the spindle assembly of a machine tool and semi-automatic cutting of single crystals (Fig. 1) [2] has the following features

- a relatively large bearing diameter relative to the length, which leads to a small value of the parameter $\lambda = L/2R$ (relative bearing length), this parameter affects the result of calculating the integrated characteristics of the aerostatic support;

- bearing surfaces have the shape of a cut-off cone, i.e., are inclined to the axis at an angle α.

Existing methods for calculating gas supports [2, 3, 4, 5] do not consider aerostatic supports with small λ (less than 0.5) and conical bearing surfaces with an angle α greater than 20^0. The methodology outlined in [6] does not provide for radial eccentricity, while in our case, exactly radial displacements are calculated. In [2], bearings with λ not less than 0.5 are considered, which is actually twice the calculated case, and if no additional technical measures are taken, the pressure drop in the middle part (and along the entire length) of a short bearing increases, and, accordingly, the bearing capacity decreases. A calculation without taking this feature into account will show an increased bearing capacity.

Calculation methods based on [3, 5] are intended for hybrid bearings, where the bearing capacity directly depends on the rotational speed, and not only on the pressure of the injected air, while the non-contact drive in this study can operate in the suspension mode. Moreover, its entry into the operating mode of rotation involves, at the initial stage, the creation of an erostatic suspension of the rotor, and then the communication of rotation to it at a given angular velocity. The methodology [4] does not include the calculation of supports with an angle α greater than 22^0.

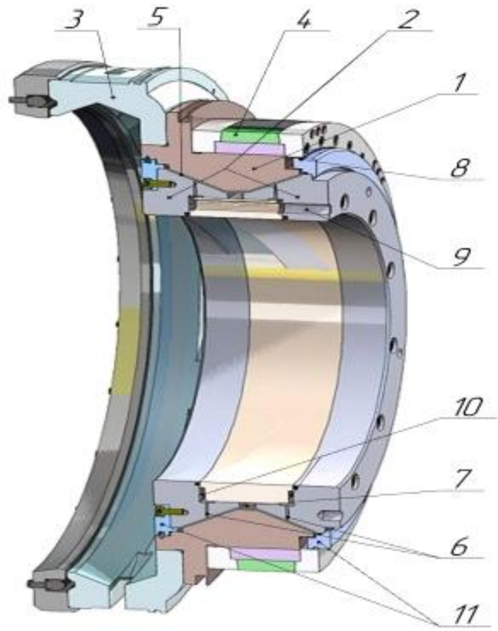

Fig. 1. Model of a single-supported non-contact spindle assembly of a semi-automatic single-crystal cutting machine:

1 - movable support part of the aerostatic bearing;

2 - fixed support part;

3 - tool mounting device;

4 - magnetic system (secondary element of a synchronous electric machine);

5 - gap with gas (air) lubricant;

6 - lubricant (process air) flow restrictors and feeders;

7 - annular chamber of air lubrication supply;

8 - labyrinth outlet for air lubrication;

9 - inlet channel;

10 - intermediate annular chamber;

11 - labyrinth seals rings.

Presentation of the main material and discussion on the research results. Among the listed methods for calculating aerostatic bearings, the closest to the studied single-support system of a contactless drive is the method described in [7]. However, it considers aerostatic bearings that do not have a taper, so to use it, we will accept the following assumptions and a calculation model in which all parameters of a radial thrust bearing are correlated with the parameters of a radial bearing (Fig. 2) to obtain the reaction of the lubricating layer on the surface of the same area, with similar conditions of gas lubrication supply.

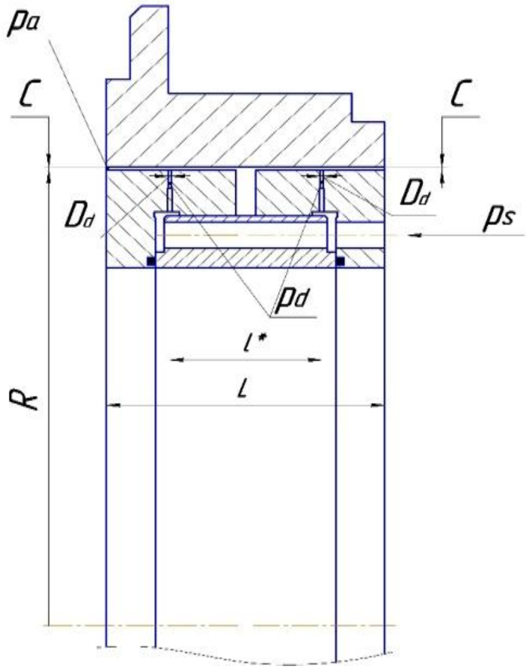

Fig. 2. Design scheme of an aerostatic bearing

The calculation scheme includes the following changes:

The bearing contact surfaces, with an angle of inclination relative to the axis of rotation of α, are rotated to a horizontal position at which $\alpha = 0$. The rotation was performed around a point at the middle radius of $R = (R_{\min} - R_{\max})/2$, which made it possible to achieve identity with the method of [7], while leaving the basic design without significant changes.

The total length of the aerostatic bearing *L* consists of two lengths of tapered parts and a space between them, which is a pressurized chamber (not drained) that allows them to be connected into a single bearing surface.

To be more compliant with the calculation methodology [8], the labyrinth seal rings were removed.

The initial value of the boost line spacing $b = l^*/L \approx 0.5$ is varied during the calculation to obtain the maximum lift force *Wr*.

Let's set the main parameters and the necessary ranges for their variation:

Assume that the lubricant for the bearing will be the air, which at a steady-state operating mode has a temperature of $t = 40^0$C, which corresponds to the air viscosity

$\mu = 1{,}9 \cdot 10^{-5}\ \dfrac{kg}{m \cdot \sec}$ and the speed of sound $a = 355.6$ m/s.

According to the design model, the bearing has a radius of R = 175 mm, a length of L = 97 mm, and a variable average clearance of C = 10...40 μm.

The process air is supplied through the inlet from an external source with a pressure of p_s= 0.3...0.63 MPa. It enters the gap between the movable and fixed bearing surfaces through two rows of flow control devices with a diameter of Dd = 0.5 mm, and a number of N = 22 in one supercharging row, the distance between the rows (lines) l^* = 40 mm. In this case, the air lubricant passing through the gap of the aerostatic bearing is throttled twice in the flow control device (feeder) and in the working gap.

Let's assume that a radial aerostatic bearing is lightly loaded, i.e., the radial and angular displacements of the moving support part from the coaxial position are small in comparison with the maximum possible displacements, which depend on the size of the radial gap. In this case, the calculation of aerodynamic forces caused by displacements of the moving support part is reduced to determining the degraded stiffness matrix, in the absence of rotation, it consists of only two elements: $K_{\mathrm{r}}^{\varepsilon}$ and K_{γ} the specific coefficients of radial and angular stiffness, respectively.

These coefficients fully characterize the bearing capacity and stiffness of the radial suspension. The initial data for calculating the specific stiffness coefficients $K_{\mathrm{r}}^{\varepsilon}$ and K_{γ} are the physical parameters of the considered aerostatic bearing, which are described in [9]:

- gas lubricant supply pressure p_s;

- external pressure p_a;

- constant i, depending on the type of flow control device;

- dimensionless geometric parameters - elongation λ and relative separation of the boost lines b;

- dimensionless mode parameter $\overline{m}$ (a complex value characterizing the design and operating conditions of the support).

Conclusions. The study develops a design scheme for a single-support drive system with aerostatic supports, which allows for a more accurate assessment of its characteristics. The limitations of existing calculation methods are revealed and an improved approach is proposed that takes into account the peculiarities of the bearing geometry and operating mode. The calculation results indicate the importance of taking into account small values of the parameter λ and the taper of bearing surfaces to improve the accuracy of modeling the bearing capacity of the system. The proposed scheme makes it possible to improve the efficiency of the machine tool spindle assembly and ensure the stability of its operation.

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Стаття надійшла до редакції 19.05.2025

Breshev Oleksii Volodymyrovych - PhD in Engineering, Department of Applied Mechanics and Materials Engineering, State University "Kyiv Aviation Institute", Ukraine, 1, Lubomyr Huzar Ave., Kyiv, Ukraine, 03058, E-mail: abreshev@gmail.com, https://orcid.org/0009-0007-4176-775X

Bashta Oleksandr Vasylovych - PhD in Engineering, Associate Professor, Department of Applied Mechanics and Materials Engineering, State University "Kyiv Aviation Institute", 1, Lubomyr Huzar Ave., Kyiv, Ukraine, 03058, E-mail: oleksandr.bashta@npp.kai.edu.ua, https://orcid.org/0000-0001-7914-897X

Nosko Pavlo Leonidovych - Doctor of Technical Sciences, Professor, Professor of the Department of Applied Mechanics and Materials Engineering, State University "Kyiv Aviation Institute", 1, Lubomyr Huzar Ave., Kyiv, Ukraine, 03058, tel. 406-78-42, E-mail: nau12@ukr.net, https://orcid.org/0000-0003-4792-6460

Bashta Alla Oleksiivna - PhD in Engineering, Associate Professor, Department of Health Products Technology, National University of Food Technologies, 68 Volodymyrska str. Kyiv, Ukraine, 01601, tel. 289-54-72, https://orcid.org/0000-0003-0310-3788

Spivak Oleksandr Mykolaiovych - PhD in Engineering, Associate Professor, Department of Electromechanics and Railway Rolling Stock, National Transport University, Kyiv Educational and Scientific Institute of Railway Transport, 19 I. Ohienko St., Kyiv, Ukraine, 03049 tel.: +38 (050) 915-23-93, E-mail: alexspi @ukr.net, https://orcid.org/0000-0002-2876-4067

Semak Inna Viktorivna - Senior Lecturer, Department of Applied Mechanics and Materials Engineering, State University "Kyiv Aviation Institute", 1 Lubomyr Huzar Ave., Kyiv, Ukraine, 03058, E-mail: inna.semak@npp.kai.edu.ua, https://orcid.org/0000-0001-9742-3226 tel.:+38 067 357 39 93.

Брешев Олексій Володимирович - кандидат технічних наук, кафедра прикладної механіки та інженерії матеріалів, Державний університет "Київський авіаційний інститут", Україна, пр-т Любомира Гузара, 1, м. Київ, Україна, 03058, E-mail: abreshev@gmail.com, https://orcid.org/0009-0007-4176-775X

Башта Олександр Васильович - кандидат технічних наук, доцент, кафедра прикладної механіки та інженерії матеріалів, Державний університет "Київський авіаційний інститут", пр-т Любомира Гузара, 1, м. Київ, Україна, 03058, E-mail: oleksandr.bashta@npp.kai.edu.ua, https://orcid.org/0000-0001-7914-897X

Носко Павло Леонідович - доктор технічних наук, професор, професор кафедри прикладної механіки та інженерії матеріалів, Державний університет "Київський авіаційний інститут", пр-т Любомира Гузара, 1, м. Київ, Україна, 03058, тел. 406-78-42, E-mail: nau12@ukr.net, https://orcid.org/0000-0003-4792-6460

Башта Алла Олексіївна - кандидат технічних наук, доцент, кафедра технології оздоровчих продуктів, Національний університет харчових технологій, вул. Володимирська, 68. Київ, Україна, 01601, тел. 289-54-72, https://orcid.org/0000-0003-0310-3788

Співак Олександр Миколайович - кандидат технічних наук, доцент, кафедра "Електромеханіка та рухомий склад залізниць", Національний транспортний університет, навчально-науковий Київський інститут залізничного транспорту, вул. І. Огієнка, 19, м. Київ, Україна, 03049 тел.: +38 (050) 915-23-93, E-mail: alexspi @ukr.net, https://orcid.org/0000-0002-2876-4067

Семак Інна Вікторівна - старший викладач кафедри прикладної механіки та інженерії матеріалів Державного університету «Київський авіаційний інститут», пр. Любомира Гузара, 1, м. Київ, Україна, 03058, E-mail: nau12@ukr.net

О.В. БРЕШЕВ, О.В. БАШТА, П.Л. НОСКО, А.О. БАШТА, О.М. СПІВАК, І.В. СЕМАК

РОЗРОБКА КОНСТРУКТИВНОЇ СХЕМИ ОДНООПОРНОЇ ПРИВОДНОЇ СИСТЕМИ З АЕРОСТАТИЧНИМИ ПІДШИПНИКАМИ

У статті розглянуто удосконалену розрахункову схему для однопідтримної безконтактної шпиндельної системи верстата для напівавтоматичного різання монокристалів. Особливу увагу приділено геометричним та режимним характеристикам аеростатичних опор, зокрема малому значенню відносної довжини підшипника ($\lambda < 0,5$) та конічній формі опорних поверхонь. Виявлено, що наявні методи розрахунку не враховують вплив цих особливостей, що призводить до неточностей у визначенні несучої здатності. Запропоновано адаптовану модель, яка дозволяє трансформувати конічну опору у площинну для використання існуючих методик. Уточнено фізико-механічні параметри повітряної змазки, а також визначено діапазони змін вхідних параметрів для досягнення максимальної підйомної сили. Результати дослідження підкреслюють важливість врахування ексцентриситету та жорсткості при моделюванні поведінки підшипника у нерухомому стані. Запропонована схема покращує точність розрахунків і підвищує ефективність роботи шпиндельного вузла. Це сприяє підвищенню надійності та стабільності функціонування верстатів з безконтактним приводом.

Ключові слова: аеростатичний підшипник, шпиндельний вузол, одноопорна система, газове мастило, радіальне зміщення, підйомна сила, конічна опора, безконтактний привід.

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