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Influence of the profile of longitudinal grooves of various depths on increasing static characteristics of radial gas bearings

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Abstract. Radial gas bearings with one line for supplying compressed gas to the working gap with blind microgrooves of constant width and variable depth of the different profiles are considered. Such as with constant depth microgrooves; triangular or stepwise with decreasing depth of grooves along the course of compressed gas. For them, the radial lifting force, the recovery moment for angular displacements of shaft in the range of constancy of radial and angular stiffness and consumption of compressed gas for the bearing's work are found. Calculations and studies have shown that a radial bearing with longitudinal microgrooves whose depth decreases linearly in the direction of gas flow has a large radial lifting force and a recovery moment for angular displacements of shaft compared to a bearing with microgrooves of constant depth. Bearings with stepped longitudinal microgrooves with greater depth at the entrance to working gap are operable in the range of practical use of bearings and have greater dimensionless radial and angular stiffness and are more economical compared to bearings with microgrooves of constant depth. In practical activities (when designing spindle units), it is not recommended to use gas bearings with a variable depth of microgrooves having a minimum depth at the entrance to working gap. With optimal design parameters, the flow rate of compressed gas for bearings with longitudinal microgrooves of different transverse profiles with alignment of shaft and sleeve is almost the same (the difference is about 7, 8%).

1. Introduction

One of important problems of modern engineering is to achieve high accuracy, rigidity and vibration resistance of equipment. One of the most critical assembly units of machine is its spindle, which is always involved in shaping movements and is subjected to operational loads [1-3]. Already at the design stage, the problem arises of creating such assembly units and elements of metal-cutting machines, which during the entire operational period would provide the specified processing accuracy. Analysis of the design of high-speed spindles on gas-lubricated bearings shows that they include radial and thrust gas-static bearings. The most important operational characteristics of such bearings are the stiffness of the lubricant layer, the recovery moment from the skew axis of the spindle and the load capacity. Therefore, the problem of creating gas bearings to ensure high performance spindles is of great importance in industry. Gas bearings can operate reliably at low and high temperatures and humidity, their use eliminates environmental pollution, reduces the level of vibration and noise. Such bearings are practically free of wear; therefore, high characteristics of spindle rotation accuracy are maintained for almost the entire life of the machines and equipment.

Spindle units with gas bearings are widely used in modern domestic and foreign production in machines and equipment for various purposes [4, 5]. They the ability to work reliably and remain durable at the required high shaft rotation speed (150 thousand min-1 and more). they have a number of advantages among other types of supports (bearings). Firstly, they can operate in a wide range of temperatures and pressures (the viscosity of gases is practically independent of temperature and pressure), as well as in the zone with increased radiation (gases are not subject to phase changes). Secondly, virtually no wear their work surfaces. Third, they have minimal friction losses, and therefore, negligible heat dissipation and can work with minimum sliding speed. Among the various types of radial gas bearings, the most technologically advanced and simplest in design are bearings with longitudinal microgrooves.

2. Literature Review

Studies have shown [1, 6] that for symmetrical gas bearings in the central position, the linear lifting force depends on the relative eccentricity in the range $-0.5 \leq \varepsilon \leq 0.5$. Therefore, the lifting force of bearing will also linearly depend on the relative radial movement of the movable sleeve. The problem of optimizing the structural parameters of the gas bearing [6] was the search for a criterion in which dimensionless rigidity itself was maximum at the minimum flow of gas. Calculations showed that for a fixed value of relative length of microgroove there always exists a value of the microgrooves parameter at which dimensionless stiffness reaches a maximum, and the closer α1 ($0 \leq \alpha_1 \leq 1$) to a maximum, the greater bearing stiffness. But as the relative length approaches unity, the gas consumption rises even faster. Therefore, bearings with a large distance between the gas supply lines have a low ratio of stiffness to gas flow.

Researches and calculations [1] showed that the dimensionless lifting force F^* and the dimensionless recovery moment M^* linearly depend on the relative radial eccentricity, $\varepsilon = \frac{e}{c}$, and angular $\theta = \frac{e_\theta}{c}$ ($e_\theta = l_0 \cdot \upsilon$ – eccentricity on the shaft end face at $e = 0$, $\upsilon \neq 0$) displacements of shaft within: $-0.5 \leq \varepsilon \leq 0.5$; $-0.5 \leq \theta \leq 0.5$.

For gas bearings, the effect of radial displacements on the recovery moment M and angular displacements on the lifting force F is practically zero [1].

Calculations and studies have shown that a bearing with longitudinal microgrooves whose depth decreases in the direction of gas flow (figure 1 b, c) [7, 8] has a significantly larger radial lifting force and recovery moment compared bearings with microgrooves constant depth. Since the gas pressure in bearings with microgrooves with a variable depth of microgrooves are functions of the longitudinal coordinate, that for calculation of characteristics of bearings was used spline methods (approximating the gas pressure in the grooves by cubic polynomials) [7,8] and cyclic sweep [9-11].

Radial bearings with stepped longitudinal microgrooves with greater depth at the entrance to the working clearance (figure 2, d) [9] are operable in the range of practical use of bearings and have greater dimensionless radial stiffness and are more economical compared to bearings with microgrooves constant depth. In perspective of further research, it is planned to optimize the design parameters of the microgrooves whose depth decreases in the direction of gas flow and stepped microgrooves with a greater depth in the zone of compressed gas supply to the working clearance of radial gas bearings according to the criteria [1].

An important characteristic of the operation of gas bearings with stepwise longitudinal microgrooves is their response to the angular displacements of the shaft (figure 1). But in scientific journals there are no calculations and studies of such bearings.

The shaft of a radial bearing with stepped longitudinal microgrooves is rotated through an angle $\upsilon = e_0/l_0$ under the action of an external load (figure 2). Working clearances in the radial bearing at $\upsilon \neq 0$:

$$h_1 = (c+\sigma)(1-\nu\cdot\vartheta\cdot\xi\cdot\cos\varphi);\quad h_2 = (c+\sigma_1)(1-\nu_1\cdot\vartheta\cdot\xi\cdot\cos\varphi);$$
$$h_3 = c\cdot(1-\nu_1\cdot\xi\cdot\cos\varphi),$$

where $\nu = \frac{c}{c+\sigma}$, $\nu_1 = \frac{c}{c+\sigma_1}$ are parameters for changing gap as a result of applying microgrooves and $\xi = \frac{z}{l_0}$ is the dimensionless axial coordinate.

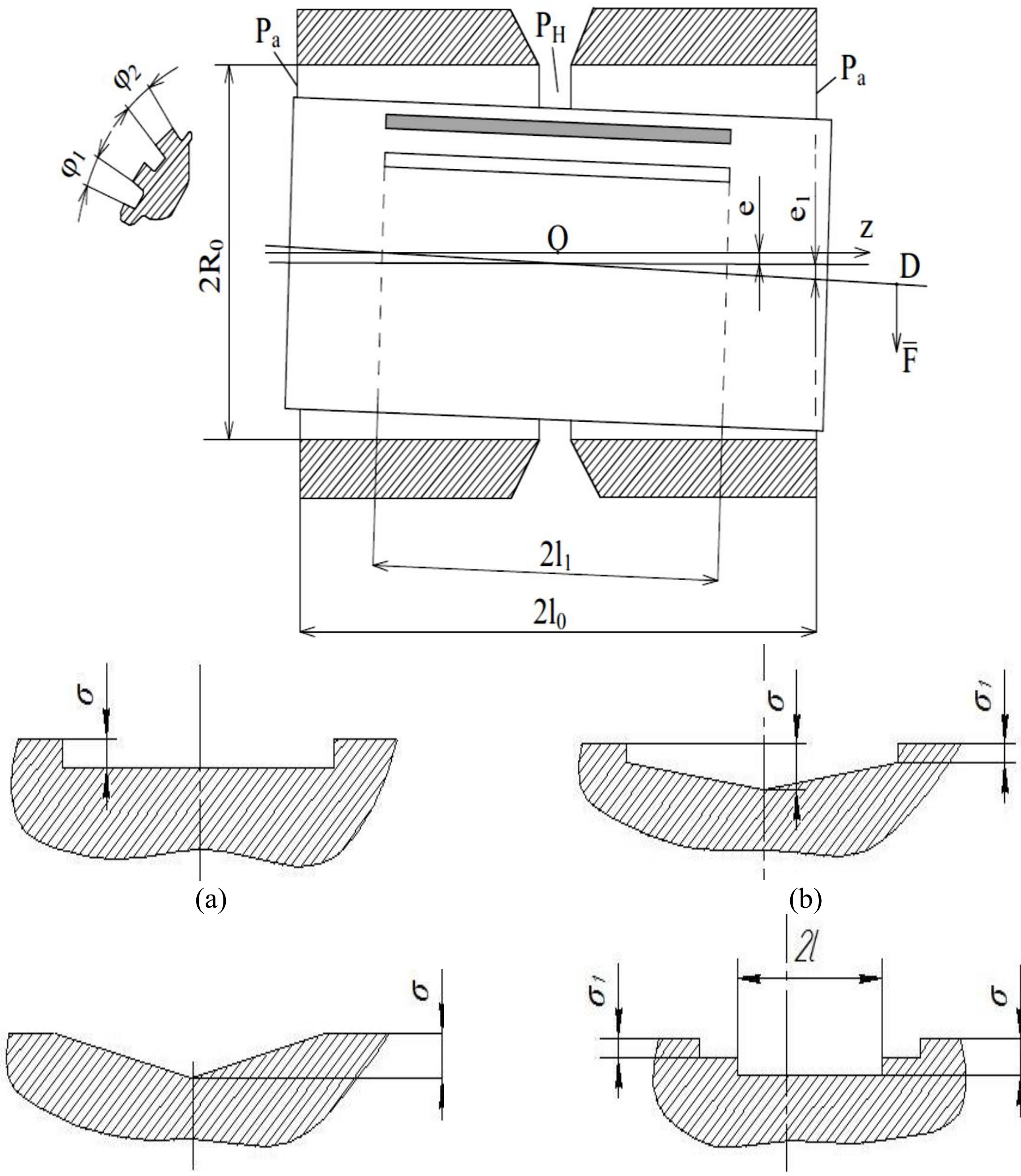

Figure 1. Radial bearing with longitudinal microgrooves for radial and angular movements of shaft under the action of force *F*:
(a) – Microgrooves of constant depth
(b), (c) – Microgrooves depth is decreases according to a linear law along the gas flow
(d) – Stepped microgrooves with a maximum depth at the inlet from an external source of compressed gas.

According to the methodology [1], the dimensionless pressure squares U_1, U_2, U_3 – the square of dimensionless gas pressure in the profiled (U_1, U_2) and smooth zones (U_3) of support with longitudinal stepwise grooves (figure 2, b) of the gas bearing are found from the expressions:

$$\left.\begin{aligned} U_1 &= P_H^2 + \chi_7 b_{02}\xi + \vartheta((\lambda\chi_7 - c_2)e^{\lambda\theta\xi} + c_2 e^{-\lambda\theta\xi} - \frac{3\beta}{\lambda^2\theta^2}\cdot\chi_7))\cdot b_{02}\cdot\cos\varphi\,;\\ U_2 &= 1 + b_{02}(c_1+\xi) + \vartheta\left(ce^{-\lambda\theta_1\xi} - (B_4 + cB_3)e^{\lambda\theta_1\xi} - \frac{3\beta_1}{\lambda^2\cdot\theta_1^2}\right)\cdot b_{02}\cdot\cos\varphi\,;\\ U_3 &= 1 - \chi_5\cdot b_{02}(1-\xi) + \vartheta\left(c_4 e^{\lambda\xi} + c_3 e^{-\lambda\xi} - \frac{3}{\lambda^2}\cdot\chi_5\right) b_{02}\cdot\cos\varphi\,. \end{aligned}\right| \quad (1)$$

The restoring moment of a radial gas bearing with stepped microgrooves with angular displacements of the shaft (figure 2):

$$M = 4p_a R_0^3\cdot M^*.$$

where M^* – the dimensionless recovery moment.

$$M^* = \lambda^2(\int_0^{\alpha}\xi d\xi\int_0^{\pi}\sqrt{u_1}\cdot\cos\varphi\cdot d\varphi + \int_{\alpha}^{\alpha_1}\xi d\xi\int_0^{\pi}\sqrt{u_2}\cos\varphi\, d\varphi + + \int_{\alpha_1}^{1}\xi d\xi\sqrt{u_3}\cos\varphi\, d\varphi.$$

Given the squares of dimensionless pressure U_1, U_2, U_3 (1) in the working sections of bearing, we obtain:

$$\begin{aligned} M^* &= \\ &= \lambda^2\int_0^{\alpha}\xi d\xi\int_0^{\pi}\left(\sqrt{\begin{matrix}P_H^2 + \chi_7 b_{02}\xi + \vartheta((\lambda\chi_7 - c_2)e^{\lambda\theta\xi} + c_2 e^{-\lambda\theta\xi} - \frac{3\beta}{\lambda^2\theta^2}\cdot\chi_7))\\ \cdot b_{02}\cdot\cos\varphi\times\cos\varphi\cdot d\varphi\end{matrix}}\right)\\ &+ \int_{\alpha}^{\alpha_1}\xi d\xi\int_0^{\pi}(\sqrt{\begin{matrix}1 + b_{02}(c_1+\xi) + \vartheta\left(ce^{-\lambda\theta_1\xi} - (B_4 + cB_3)e^{\lambda\theta_1\xi} - \frac{3\beta_1}{\lambda^2\cdot\theta_1^2}\right))\\ \cdot b_{02}\cdot\cos\varphi\times\cos\varphi\cdot d\varphi\end{matrix}}\\ &+ \int_{\alpha_1}^{1}\xi d\xi\int_0^{\pi}\sqrt{1 - \chi_5\cdot b_{02}(1-\xi) + \vartheta\left(c_4 e^{\lambda\xi} + c_3 e^{-\lambda\xi} - \frac{3}{\lambda^2}\cdot\chi_5\right) b_{02}\cdot\cos\varphi\cdot\cos\varphi\cdot d\varphi}\,. \end{aligned}$$

In the working range θ, the actual expression $M^* = \vartheta k_\upsilon^*$ [1, 10], where k_ϑ^* is the derivative of M^* with respect to θ at $\theta = 0$, that is, the angular stiffness in the linearity range M^*.

$$\begin{aligned} &\left(\frac{dM^*}{d\vartheta}\right)_{\vartheta=0} = k_\vartheta^* = \\ &\quad = \lambda^2\int_0^{\alpha}\xi d\xi\int_0^{\pi}\frac{(\left(\lambda\chi_7 - c_2 e^{\lambda\theta\xi} - \frac{3\beta}{\lambda^2\theta^2}\cdot\chi_7\right))\cdot b_{02}\cdot\cos^2\varphi\cdot d\varphi}{2\sqrt{P_H^2 + \chi_7 b_{02}\xi}} + \\ &\quad + \int_{\alpha}^{\alpha_1}\xi d\xi\int_0^{\pi}\frac{(\left(-(cB_3 - B_4)e^{\lambda\theta_1\xi} + ce^{-\lambda\theta_1\xi} - \frac{3\beta_1}{\lambda^2\theta_1^2}\right))b_{02}\cdot\cos^2\varphi\cdot d\varphi}{2\sqrt{1 + (c_1+\xi)\cdot b_{02}}} + \\ &\quad + \int_{\alpha}^{1}\xi d\xi\int_0^{\pi}\frac{(c_4 e^{\lambda\xi} + c_3 e^{-\lambda\xi} - \frac{3}{\lambda^2}\cdot\chi_5)\cdot b_{02}\cdot\cos^2\varphi\cdot d\varphi}{2\sqrt{1 - \chi_5(b_{02}(1-\xi))}}. \end{aligned} \quad (2)$$

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The dimensionless flow rate of compressed gas Q^* for operation of the radial gas bearing are found at $\theta = 0$ [1]:

$$Q^* = \frac{2(P_H^2 - 1)}{\lambda(1 - \alpha + \frac{\alpha \nu^3}{a_e + \nu^3(1 - a_e)})}.$$

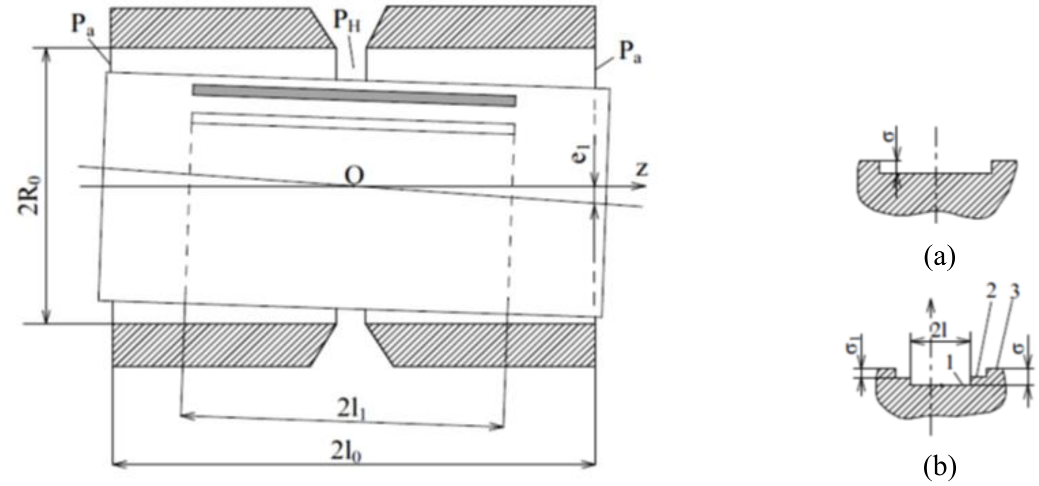

Figure 2. Radial bearing with longitudinal microgrooves at angular shaft movements

3. Results

In radial bearings with longitudinal microgrooves of constant depth (figure 2, a), for a fixed value of the width of microgrooves a_e, there are optimal values of the depth and length of microgrooves (figure 3, 4) at which function K_ε^*/Q^* reaches its maximum. But the design parameter of microgrooves a_e is not included in the values by which the optimization is performed. Originality of choice a_e is complicated by fact that as the microgrooves width decreases their optimum depth increases, and the dimensionless radial rigidity of the bearing increases closer to maximum value (figure 3).

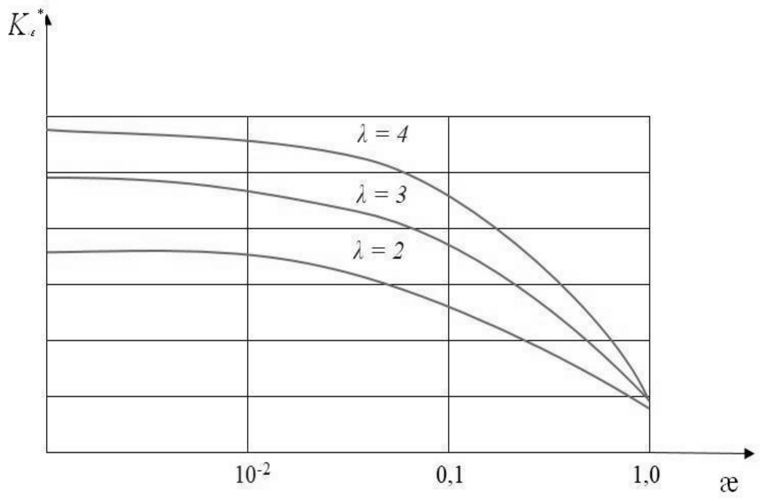

Figure 3. Dependence of dimensionless radial rigidity, K_ε^*, of a radial bearing with longitudinal microgrooves at $P_H = 3$ on the relative width of microgrooves a_e.

Depending on the optimum values of parameters ν and α on a_e (fig. 4) at the dimensionless $P_H = 3$ gas pressure supplied to the input of working clearance of the radial bearing, it is possible to draw an incorrect conclusion that longitudinal microgrooves should be tried to be made narrow and deep. But the Reynolds equations [1] are valid only if the gas flow is laminar in the working clearance of the bearings and depth of microgrooves must at least two orders of magnitude (Critical R_e = 500 to 1000) should be less than their width. Therefore, the actual width of microgrooves during calculations [1, 10] and use in spindle units and in production [1, 12] is always limited by reasonable limits $a_e = 0.25\ to\ 0.5$.

For a fixed value of the relative microgroove length α for a radial bearing with longitudinal microgrooves, the depth of which decreases in the direction of gas flow (figure 1, b), there are values ν and β (microgroove depth parameter) for which the dimensionless stiffness K_ε^* of the radial bearing reaches a maximum ($K_\varepsilon^* = 1.884$), which is significantly larger (by 54%) than at bearing with microgrooves of constant depth.

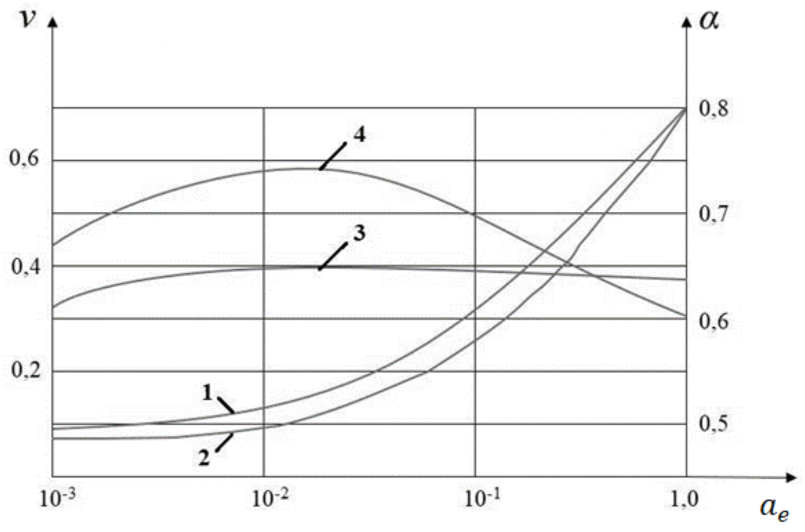

Figure 4. Dependence of P_H = 3 on the parameters ν (curves 1, 2) and the relative length α (curves 3, 4), on the relative width of microgrooves a_e.

A radial bearing with a maximum depth of microgrooves on the gas boost line (figure 1 b, c) has extremes of dimensionless radial stiffness at the coefficient of depth of microgrooves β and the clearance smoothness parameter ν and according to the criteria [1], no static instability zones were found for such bearings at $0 \le \beta \le 1$ and $0.2 \le \nu \le 1$.

4. Discussion

Calculations showed that with increasing only the relative step length α_1, without changing the depth of longitudinal microgrooves, the dimensionless radial and angular stiffness of the bearing increases for the profile of microgrooves, in which the step depth decreases in the direction of gas flow (figure 1, d, figure 2, b).

In radial gas bearings with stepped microgrooves with a maximum depth on the compressed gas supply line to the working clearances (figure 2, b), dimensionless radial stiffness are larger (about 12%), angular stiffness and the ratio K_ε^*/Q^* are large (25%) in comparison with a bearing in which microgrooves has constant depth and design parameters are optimal (maximum K_ε^*/Q^*).

5. Conclusions

A radial bearing with longitudinal microgrooves whose depth decreases linearly (figure 1 b, c) in the direction of gas flow has a large radial lifting force and a recovery moment for angular displacements of the shaft compared to a bearing with microgrooves of constant depth (figure 1, a).

Bearings with stepped longitudinal microgrooves with greater depth at the entrance to the working clearance (figure 2, b) are operable in the range of practical use of bearings and have large dimensionless radial and angular stiffness and more economical ($\Phi = K_{\varepsilon}^{*}/Q^{*}$) in comparison with bearings in which microgrooves of constant depth.

The decrease in radial lifting force (about 4,67 %) of bearing with stepped microgrooves (figure 1, d) in comparison with bearing with longitudinal microgrooves whose depth decreases linearly (figure 2, b, c) is explained by an increase in the accuracy of approximate calculations, since the length of section with microgrooves are $0 \leq \alpha_1 \leq 0{,}669$ (figure 2, b, c), and in the stepped are approximated by cubic polynomials at $0{,}4 \leq \alpha_1 \leq 0{,}669$ and $0 \leq \alpha_1 \leq 0{,}4$.

In practical activities (when designing spindle units), it is not recommended to use gas bearings with microgrooves of variable depth, which have a minimum depth at the entrance to the working gap.

With optimal design parameters, the flow rate of compressed gas for bearings with longitudinal microgrooves of different transverse profiles at $\varepsilon = 0$ (figure 2) is almost the same (the difference is about 7.8%).

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