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STUDY OF LUBRICANT FLOW RATE THROUGH A BEARING DEPENDING UPON THE LOCATION OF THE LUBRICANT INLET

by

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FOREIGN TECHNOLOGY DIVISION

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A study is made of the effect of the sliding rate, the specific pressure load, and viscosity of the lubricant and also the location of the lubricant inlet on the flow rate of a lubricant through a bearing. The flow process of the lubricant in the bearing clearance is analyzed. On the basis of theoretical and experimental studies an expression is obtained for the lubricant flow rate through a complexly loaded bearing considering the location of the lubricant inlet. The boundaries of the loaded zone are estimated. The effect of misalignment of the shaft and bearing axes on the thermal operating conditions of the bearing is investigated. Recommendations are made with respect to calculation, planning and design of complexly loaded sliding bearings. [AR1202850]

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А а	*А а*	A, a	Р р	*Р р*	R, r
Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
Ж ж	*Ж ж*	Zh, zh	Ц ц	*Ц ц*	Ts, ts
З з	*З з*	Z, z	Ч ч	*Ч ч*	Ch, ch
И и	*И и*	I, i	Ш ш	*Ш ш*	Sh, sh
Й й	*Й й*	Y, y	Щ щ	*Щ щ*	Shch, shch
К к	*К к*	K, k	Ъ ъ	*Ъ ъ*	"
Л л	*Л л*	L, l	Ы ы	*Ы ы*	Y, y
М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

* ye initially, after vowels, and after ъ, ь; e elsewhere. When written as ё in Russian, transliterate as yё or ё. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

FOLLOWING ARE THE CORRESPONDING RUSSIAN AND ENGLISH DESIGNATIONS OF THE TRIGONOMETRIC FUNCTIONS

Russian	English
sin	sin
cos	cos
tg	tan
ctg	cot
sec	sec
cosec	csc
sh	sinh
ch	cosh
th	tanh
cth	coth
sch	sech
csch	csch
arc sin	$\sin^{-1}$
arc cos	$\cos^{-1}$
arc tg	$\tan^{-1}$
arc ctg	$\cot^{-1}$
arc sec	$\sec^{-1}$
arc cosec	$\csc^{-1}$
arc sh	$\sinh^{-1}$
arc ch	$\cosh^{-1}$
arc th	$\tanh^{-1}$
arc cth	$\coth^{-1}$
arc sch	sech^{-1}
arc csch	csch^{-1}
rot	curl
lg	log

STUDY OF LUBRICANT FLOW RATE THROUGH A BEARING DEPENDING UPON THE LOCATION OF THE LUBRICANT INLET

V. A. Karamzin

The high heat intensity of bearings of modern machines and, in particular, the crankpin bearings of engines require the correct control of heat flows.

Basically the heat removal depends on the quantity and quality of lubricant flowing through the bearing, since it is one of the main factors affecting the temperature conditions of the bearing, and consequently its reliability and life [1-3].

The purpose of this work was the study of the effect of the main parameters, characterizing the operating conditions of the bearing, on its working capacity. We investigated the effect of the slip rate, specific load, pressure and viscosity of lubricant, and also the location of the lubricant inlet on the flow rate of lubricant through the bearing during complex loading. The tests were conducted on a special stand for testing bearings [4, 5].

Tested was a bearing consisting of two bushings 1.75 mm thick made of high-tin aluminum alloy ($d = 66$ mm, $l = 27$ mm). The diametral clearance was 70 microns, ratio $l/d = 0.41$.

In the process of testing the [illegible] conditions varied from 1400 to 3200 rpm; the average [illegible] load from 29 to 140 kgf/cm^2; the oil pressure from 1 to 3 kgf/cm^2; the temperature of oil, fed to the bearing, from 50 to 110° C; the location of lubricant inlet relative to the direction of loading component, referred to the shaft, varied from -22 to 108°.

The solution of Reynolds equation for pressure in a plain bearing of infinite length was obtained by Zhukovskiy and Sommerfeld. In this case the equation for pressure in partial derivatives is reduced to ordinary differential equation, and the solution is obtained by simple integration.

The solution for a bearing of zero length is obtained by Okvirk. His solution is based on the assumption that in the bearing the axial gradient of pressure is an order of magnitude higher than radial pressure gradient. This assumption pertains to a bearing of finite length. Although the equation of pressures in the oil layer is nonlinear and nonhomogeneous, its approximate solution can be obtained by various mathematical methods.

Musket and Morgan obtained the solution of this problem with expansion of pressure into power series with respect to χ [6].

Cameron and Wood investigated this problem, using the Southwell method [7].

The variational method was developed by Weber and Hays.

Some investigators, for example, Stodol, Yanovskiy, Korovchinskiy [8], represented the solution of the problem, i.e., the sought function of pressure distribution, in the form of the product of two functions

$$p(\varphi; \omega) = D(\varphi) \cdot f(\omega),$$

where $D(\varphi)$ depends only on φ, and $f(\omega)$ depends only on ω.

Shibel' and Khanovich used the same method, but with considerable limitations.

The exact and complete solution of this problem was obtained by Tao, who considered the viscosity a constant or a function of only pressure [9]. In this investigation the Reynolds equation

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right)=6\mu U\frac{dh}{dx} \qquad (1)$$

after reduction to dimensionless form and substitution

$$p(\varphi; z)=\xi(\varphi)+\zeta(\varphi; z) \qquad (2)$$

is expressed by two equations

$$\frac{d}{d\varphi}\left[(1+\chi\cos\varphi)^3\frac{d\xi}{d\varphi}\right]=\frac{6\nu r}{Uc^2}\cdot\frac{dH}{d\varphi}, \qquad (3)$$

$$\frac{\partial}{\partial\varphi}\left[(1+\chi\cos\varphi)^3\frac{d\zeta}{d\varphi}\right]+\frac{\partial}{\partial z}\left[(1+\chi\cos\varphi)^3\frac{\partial\zeta}{\partial z}\right]=0. \qquad (4)$$

To these equations will correspond the following boundary conditions

$$\xi(-\pi)=\xi(\pi);\quad \frac{d\xi}{d\varphi}(-\pi)=\frac{d\xi}{d\varphi}(\pi), \qquad (5)$$

$$\zeta(-\pi; z)=\zeta(\pi; z);\ \frac{\partial\zeta}{\partial\varphi}(-\pi; z)=\frac{\partial\zeta}{\partial\varphi}(\pi; z), \qquad (6)$$

$$\zeta(\varphi; -l/2)=\zeta(\varphi; l/2)=-\xi(\varphi). \qquad (7)$$

The solution of equation (3) represents the function of pressure distribution for a bearing of infinite length

$$\xi(\varphi)=\frac{6\nu r\chi}{Uc^2(2+\chi^2)}\cdot\frac{(2+\chi\cos\varphi)\sin\varphi}{(1+\chi\cos\varphi)^2}. \qquad (8)$$

During the solution of equation (4) the method of separation of variables was used, which led to the finding of eigenvalues and eigenfunctions.

For finding the eigenvalues there was applied the method of expansion into series with respect to orthogonal functions, proposed by Ramachandra [10].

Thus, the final expression of pressure distribution in a bearing of finite length has the form

$$p(\varphi; z) = \frac{6vr\chi}{Uc^2(2+\chi^2)} \cdot \frac{(2+\chi\cos\varphi)\sin\varphi}{(1+\chi\cos\varphi)^2} - \\ - \frac{1}{(1+\chi\cos\varphi)^{\frac{3}{2}}} \sum_{m=1}^{\infty} \operatorname{ch} \lambda_i^{\frac{1}{2}} z B_n^i \sin n\varphi. \quad (9)$$

In connection with the fact that the obtained final expression causes difficulties during calculations, it was somewhat simplified by the use of the approximate solution proposed by Vorner [9].

For this on the basis of expression (7) let us write

$$-\xi(\varphi) = \sum_{N=1}^{\infty} c_N \theta_N(\varphi) \psi_N\left(\frac{l}{2}\right). \quad (10)$$

By expanding $\xi(\varphi)$ into series with respect to functions θ, we obtain

$$P(\varphi; z) = \sum_{N=1}^{\infty} a_N \theta_N(\varphi) \left[1 - \frac{\psi_N(z)}{\psi_N\left(\frac{l}{2}\right)}\right]. \quad (11)$$

Having made the assumption about the fact that

$$\left[1 - \frac{\psi_N(z)}{\psi_N(l/2)}\right] \approx \left[1 - \frac{\psi_1(z)}{\psi_1(l/2)}\right],$$

we finally have

$$P(\varphi; z) \approx \left[1 - \frac{\psi_1(z)}{\psi_1(l/2)}\right] \sum_{N=1}^{\infty} a_N 0_N(\varphi) = \left[1 - \frac{\psi_1(z)}{\psi_1\left(\frac{l}{2}\right)}\right] \xi(\varphi), \tag{12}$$

where $\psi_1(z) = \operatorname{ch} \lambda_1 z$.

The quantity of lubricant flowing through the plain bearing in the process of operation is made up of two magnitudes: the quantity of oil flowing through the loaded or working zone M_1, and the quantity of oil flowing through the unloaded zone M_2

$$M = M_1 + M_2. \tag{13}$$

The loaded zone of the bearing is characterized by relatively small thickness of the oil layer and high pressures. In the unloaded zone the oil pressure does not exceed the pressure in the line, but local thicknesses of the oil layer are relatively great.

The ratio between M_1 and M_2 can be different depending upon the construction of the bearing, the eccentricity, the oil feed pressure and especially on the place of supply of lubricant.

For determination of the lubricant flow rate through the loaded zone there is considered the outflow of fluid in axial direction (z).

The total flow rate of lubricant through the loaded zone of the bearing is obtained by integration of the elementary flow within the boundaries of the loaded zone (φ_1 и φ_2)

$$M_1 = \int_{\varphi_1}^{\varphi_2} \frac{h^3}{12\mu} \cdot \frac{\partial p}{\partial z} r d\varphi, \tag{14}$$

where h — present thickness of the oil layer, determined from expression $h = \frac{\eta}{2}(1+\chi\cos\varphi)$; μ - viscosity of oil; r - radius of the bearing.

Considering the earlier obtained expression of pressure distribution $p = f(\varphi; z)$ in the flow equation through the loaded zone and performing a series of transformations, we obtain the expression in the form

$$M_1 = d^2\eta\omega\frac{r}{16}\cdot\frac{\chi}{(2+\chi^2)}\times \times\left[\lambda_1 \operatorname{th}\lambda_1\left(\frac{l}{2}\right)\right]\int_{\varphi_1}^{\varphi_2}(1+\chi\cos\varphi)(2+\chi\cos\varphi)\sin\varphi d\varphi, \qquad (15)$$

where η - diametral clearance; ω - angular velocity; χ - relative eccentricity.

The values of characteristic parameter λ_1 as a function of χ are presented below:

χ . . .	0,2	0,4	0,6	0,8
λ_1 . . .	1,04	1,16	1,28	1,35

Finally the expression of flow rate through the loaded zone of the bearing can be written so:

$$M_1 = d^2\eta\omega\zeta_c. \qquad (16)$$

The value of the integral entering the expression M_1 was computed for various boundaries of the loaded zone when $\varphi_2 = 180°$ and $\varphi_1 = 50\text{-}110°$ and at values of $\chi = 0.6\text{-}0.9$.

The values of flow rate through the loaded zone and the results of analytical calculations coincided for various values relative to eccentricity at certain values of the boundaries of the loaded zone.

Thus, the relationship of the change of boundaries of the loaded zone to the relative eccentricity was established:

χ . . .	0,6	0,75	0,8	0,85	0,9
$\varphi_2 - \varphi_1$	85	92	100	111	125

For the investigated bearing with change of the relative eccentricity from 0.6 to 0.9 the extent of the boundaries of the loaded zone $(\varphi_2 - \varphi_1)$ varied from 85 to 125°, which can explain the complex character of the load, the vector diagram of which is shown in Fig. 1, and also by the insufficient rigidity of the connecting-rod big end, in which the bushings of the tested bearing were installed.

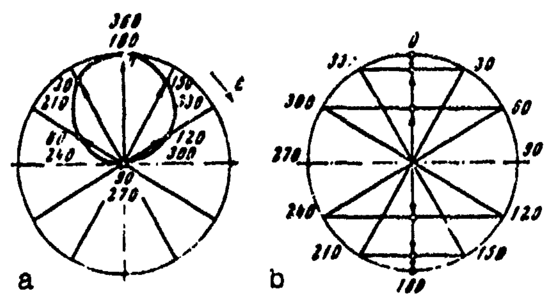

Fig. 1. Vector diagrams of loading, pertaining to the shaft (a) and bearing (b).

The flow rate of lubricant through the unloaded part of the oil layer is usually determined by proceeding from the equation of flow of fluid through a narrow slot, the length of which is taken equal to the expanded length of the unloaded part, and the height - to the averaged thickness of the unloaded part of the oil layer

$$M_2 = Ak_\omega \frac{p_\kappa \eta^2}{\mu} \cdot \frac{d}{l}. \qquad (17)$$

The flow rate of oil through the unloaded zone of the oil layer depends, as was indicated above, on the place of supply or on how far the place of supply of lubricant is from "favorable." As is known, the "favorable" place of supply of lubricant is characterized by the best filling of the wide part of the bearing clearance by the outflowing oil.

The resistance to the flow of oil into the clearance between openings in the shaft neck and the surface of the bushings depends upon in which clearance region the holes fall, serving for supply of lubricant. Coefficient A depends on the place of supply of lubricant, which is characterized by the corresponding thickness of the oil layer. In case of coincidence of the lubricant supply place with "favorable" coefficient $A_{бл} = 1$.

Therefore it is possible to write

$$\frac{M_2}{M_{2max}} = \frac{A}{A_{бл}} \approx \frac{h}{h_{бл}} = 1 - \frac{\chi}{1+\chi}(1-\cos\alpha), \qquad (18)$$

where α - the angle between the oil hole and the "favorable" place of lubricant supply.

With a more complex form of loading, which takes place in this experiment, the expression for $A/A_{бл}$ is somewhat complicated and is considered by the introduction of coefficient a, depending on χ and angle α between the location of the lubricant inlet and the "favorable" lubricant supply place

$$a = 0{,}27\,[1 + (1{,}5\chi - 1{,}125)0{,}0174\alpha]. \qquad (19)$$

As the results of the conducted experiments showed, all other conditions being equal, the "favorable" lubricant supply place with change of pressure of the oil being supplied is somewhat displaced in the direction opposite the rotation of the shaft.

Considering the displacement of the "favorable" lubricant supply place to angle φ' from the direction of loading component, we have

$$\alpha = \varphi - \varphi'.$$

By substituting the value of α in formula (19), finally we have

$$\frac{M_2}{M_{2\max}} = 1 - \frac{1}{a} \cdot \frac{\chi}{1+\chi} [1 - \cos(\varphi - \varphi')]. \qquad (20)$$

In the presented expression the value of angle φ' is taken from 0 to 20°.

Thus, the flow rate of lubricant through the unloaded zone of the bearing at any location of the lubricant inlet is determined from expression (21) and will be equal to

$$M_2 = \left[1 - \frac{\chi}{a(1+\chi)} \cdot (1 - \cos\alpha)\right] k_{\omega} \frac{p_{M}\eta^3}{\mu} \cdot \frac{d}{l}, \qquad (21)$$

where p_{M} - oil pressure.

As numerous experiments showed, the flow rate of lubricant through the unloaded zone depends also on the speed of rotation of the shaft, which is considered by the introduction of coefficient k_{ω}.

The values of coefficient k_{ω} are presented on Fig. 2.

Finally the total flow rate of lubricant through the bearing for any location of the lubricant inlet is written in the form

$$M = d^2\omega\eta\zeta_{c} + \frac{p_{M}\eta^3}{\mu} \cdot \frac{d}{l}\left[k_{\omega} 1 - \frac{\chi}{a(1+\chi)} (1 - \cos\alpha)\right]. \qquad (22)$$

This formula can be used during engineering calculations of multi-loaded plain bearings, having supply of lubricant through one hole.

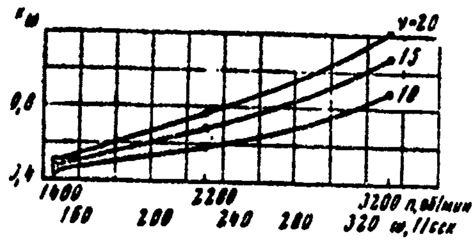

Fig. 2. Relationship of coefficient k_ω to velocity conditions n and oil viscosity ν.

Designations: об/мин = rpm; сек = s.

The value of coefficient ζ_c can also be computed by the formula obtained as a result of processing the experimental data

$$\zeta_c = 1{,}6 + \left(\frac{\chi}{1-\chi}\right)^{1{,}1}. \qquad (23)$$

In the conducted investigation the value of the relative eccentricity for various operating conditions was determined according to the value of the loading factor [1], which in turn was computed according to the value of the mean effective pressure, equal to the ratio of the effective load to the area of projection of the bearing

$$k_{ф} = p_{эф}/dl. \qquad (24)$$

Fig. 3 shows graphs of the relationship of the flow rate of lubricant to the feed pressure $p_н$ and viscosity ν for n = 2600 rpm and φ = 40°. Analogous graphs were obtained for n = 1400-3200 rpm and φ = 108-(-22°). The flow rate through the loaded zone M_1 was determined as M = $f(p_н)$ when $p_н$ = 0.

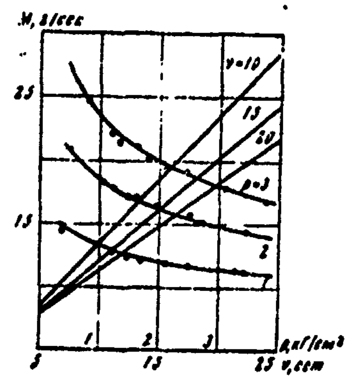

Fig. 3. Relationship of flow rate of lubricant to the feed pressure and viscosity.
Designations: сек = s; кГ = kgf; сст = cSt.

On the basis of experimental data graphs are constructed of the flow rate of lubricant depending on the location of the lubricant inlet for ν = 20 cSt and p = 3 kgf/cm^2 (Fig. 4). Analogous graphs were obtained for ν = 10 and 15 cSt and p = 1 and 2 kgf/cm^2.

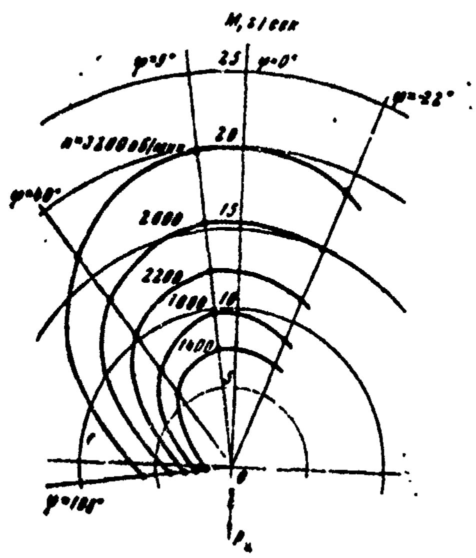

Fig. 4. Relationship of flow rate of lubricant to the location of lubricant inlet for various velocity conditions when ν = 20 cSt and p = 3 kgf/cm^2.
Designations: сек = s; об/мин = = rpm.

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