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exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machinery

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Gdańsk

The Influence of Thermoelastic Deformations of Bearing Bush and its External Fixings on Static and Dynamic Properties of Journal Bearings

Part II. Experimental Verification and Theoretical Investigations*

Results of experimental verification of thermoelastohydrodynamic theoretical model of bearing presented in Part I are given. Results of theoretical investigations performed basing on the model and computer program presented are given as well.

Nomenclature

b_0 – coefficient of the temperature influence on the dynamic viscosity of oil,

C_{wt} – specific heat of oil, $[m^2/(s^2 K)]$,

D – journal diameter, [m],

E – modulus of elasticity in tension, $[N/m^2]$,

g – acceleration of gravity, $g = 9.81\ m/s^2$,

h_{min}, H_{min} – dimensional and nondimensional minimum thickness of the lubricating gap, h_{min} [m], $H_{min} = h_{min}/\Delta r$,

h_δ, H_δ – dimensional and nondimensional deformation of the lubricating gap in the radial direction, h_δ [m], $H_\delta = h_\delta/\Delta r$,

K_{ol} – coefficient of heat transfer for oil, $[kg\ m/(s^3 K)]$,

k_p – coefficient of heat transfer for the material of the bush, $[kg\ m/(s^3 K)]$,

L – width of the bearing, [m],

n – rotational speed of the rotor (bearing journal), [rev/min],

p – hydrodynamic pressure in oil film, $[N/m^2]$,

p_0 – oil feeding pressure in axial oil grooves, $[N/m^2]$,

p_{av} – average bearing load, $p_{av} = P_{st}/(LD)$ $[N/m^2]$,

P_{st} – static bearing load, [N],

q_c, Q_c – dimensional and non-dimensional total requirement of the oil $q_c\ [m^3/s]$, $Q_c = q_c/(LD\,\omega\,\Delta r)$, $Q_c = Q_{s1} + Q_{s2} + \cdots +$,

Q_k – hot oil carry over flow (non-dimensional),

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Q_p – non-dimensional flow of oil at the leading edge,

Q_z – fresh flow of oil (non-dimensional) fed to axial oil groove,

q_w, Q_w – dimensional and nondimensional side leakage from axial oil grooves, $Q_w = Q_{w1} + Q_{w2} + \cdots +$,

r_w – radius of the inner surface of the bush,

Δr – radial clearance, [m],

So_0 – related Sommerfeld number, $So_0 = S_0\, \omega/\omega_0$,

$t_c, t_M, t_o, t_p, t_z, T_c, T_M, T_o \cdots$ – dimensional and non-dimensional temperatures respectively of journal, of mixing in axial oil grooves, of the fed oil of the bush, and of the surrounding,

$$t\,[\mathrm{K}],\; T = t\rho\, C_{wt}(\Delta r/r_w)^2/(\mu_0\, \omega),$$

$(t_p)_{max}$, $(t_p)_{min}$ – maximum and minimum temperature on the internal surface of the bush, [K],

$u_{1,2}$ – values of oil film damping in the rotor-bearing system which correspond to the first and the second self-vibration frequency, [rad/s],

$v_{1,2}$ – frequency of self-vibrations in the rotor-bearing system, [rad/s],

α_{0t} – coefficient of heat transfer by free convection, [kg/(s³ K)],

α_t – coefficient of thermal expansion of the material of the bush, [1/K],

γ – attitude angle, [°],

ε – relative eccentricity, $\varepsilon = e/\Delta r$,

ψ, $\psi_{1,2,3,4}$ – circumferential angle coordinate, coordinate of position of the edges of axial oil grooves, $\psi = l/r_w$, [rad], [°],

ρ – density of oil, [kg/m³],

Π_z – non-dimensional pressure due to external fixings

ω – angular velocity of the rotor (of the journal), [rad/s],

ω_{gr} – limit of stability, [rad/s],

ω_0 – reference parameter, $\omega_0 = \sqrt{g/\Delta r}$ [rad/s].

1. Experimental verification of theoretical model

The experimental verification of the theoretical model presented in the first part of this work [1] has been performed basing on the data obtained at the Institute of Fluid Flow Machinery, Polish Academy of Science [2] and the data presented by the French authors [3]. In both cases the verification consisted of checking of the accuracy of the calculated and measured temperature distributions of the internal surface of the bush and of comparing the pressure distributions in the oil film.

The experimental stand used for experimental investigations in [2] consisted of a journal supported on two precise ball bearings, a cast iron bush lined with bearing alloy (white metal) and a servo-motor pressing on the bush through a ball-shaped element. Such a mode of load transfer ensured its vertical direction and a coaxial positioning of the journal and the bush. The bearing was fed from two axial oil grooves. In order to measure temperature 93 iron-constantan thermocouples were placed in a white metal layer ~ 0.5 mm beneath the slider surface. The pressure of the oil film was measured in the central plane of the bearing with the use of a piezoelectric sensor mounted in the journal. The bearing under investigation had an elliptical clearance and the following geometrical data: journal diameter – 0.14 m, external diameter of the bush – 0.21 m; width of the bearing – 0.113 m, ($l/D \cong 0.8$), horizontal radial clearance 2.1×10^{-4} m, vertical radial clearance – 1.1×10^{-4} m, positioning of the axial oil grooves edge $\psi_1 = 7°$, $\psi_2 = 173°$, $\psi_3 = 187°$, $\psi_4 = 353°$; thickness of the white metal lining ~ 3×10^{-3} m.

The experimental investigation was performed with the following operational parameters: temperature of the supplied oil – 303 K, ambient temperature – 295 K, feeding pressure – 0.2×10^5 N/m^2, dynamic viscosity of oil ($Z-26$) at temperature 303 K – 0.0798 NS/m^2, at temp. 323 K – 0.030 NS/m^2, $P_{st}=34\,000$ N and rotation speed $n=1900$ rev/min.

The following material coefficients were used in the calculations $k_{0t}=0.135$ kgm/s^{3}K, $C_{wt}=2000$ m^2/s^{2}K, $\rho=900$ kg/m^3, $\alpha_t=9\times10^{-6}$ 1/K, $E=1\times10^{11}$ N/m^2.

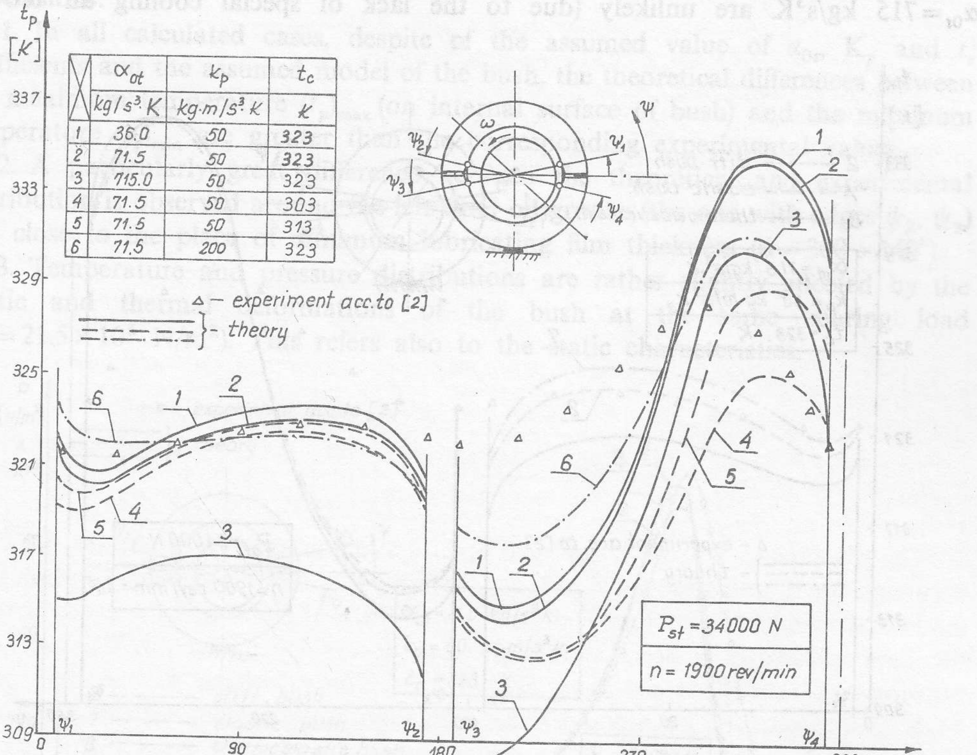


Fig. 1. The temperature distribution on the internal surface of the bush calculated for different values of α_{0t}, k_p, t_c and for the model of the stiff bush compared with the experimental data of [2] (the numbering of the curves consistent with the numbering of the cases in Table 1)

The coefficients α_{0t}, k_p and t_c were considered as parameters which may take different values. The obtained theoretical data can give a good illustration of influence of these coefficients on the basic static characteristics of the bearing.

The results of theoretical calculations compared with experimental data obtained in the Institute of Fluid Flow Machinery are shown in Fig. 1 - 3 and in Table 1. Fig. 1 shows the temperature distribution on the internal surface of the bush calculated for the following coefficients $\alpha_{0t}=36$, 71.5, 715 kg/s^{3}K, $k_p=50$, 200 kgm/s^{3}K, $t_c=303$, 313, 323 K and for the stiff non-deformable bush model. It can be seen that the higher is the intensity of the heat transfer to the surroundings (i.e., for higher values of the coefficient α_{0t}), the lower is the temperature of the outer surface of the bush, An increase of the temperature of the journal t_c causes a significant increase of the maximal temperatures. On the other hand, the higher is the heat conductance of the material of the bush, the lower is the difference between the maximal and minimal temperatures. These results confirm expectations. For the experimental investigations a bush made of cast-iron has been used. Its coefficient is close to 50 kgm/s^{3}K. Furtheron, one, should mention that for a journal the values of $t_c=303$K and $\alpha_{0t}=715$ kg/s^{3}K are unlikely (due to the lack of special cooling air blower

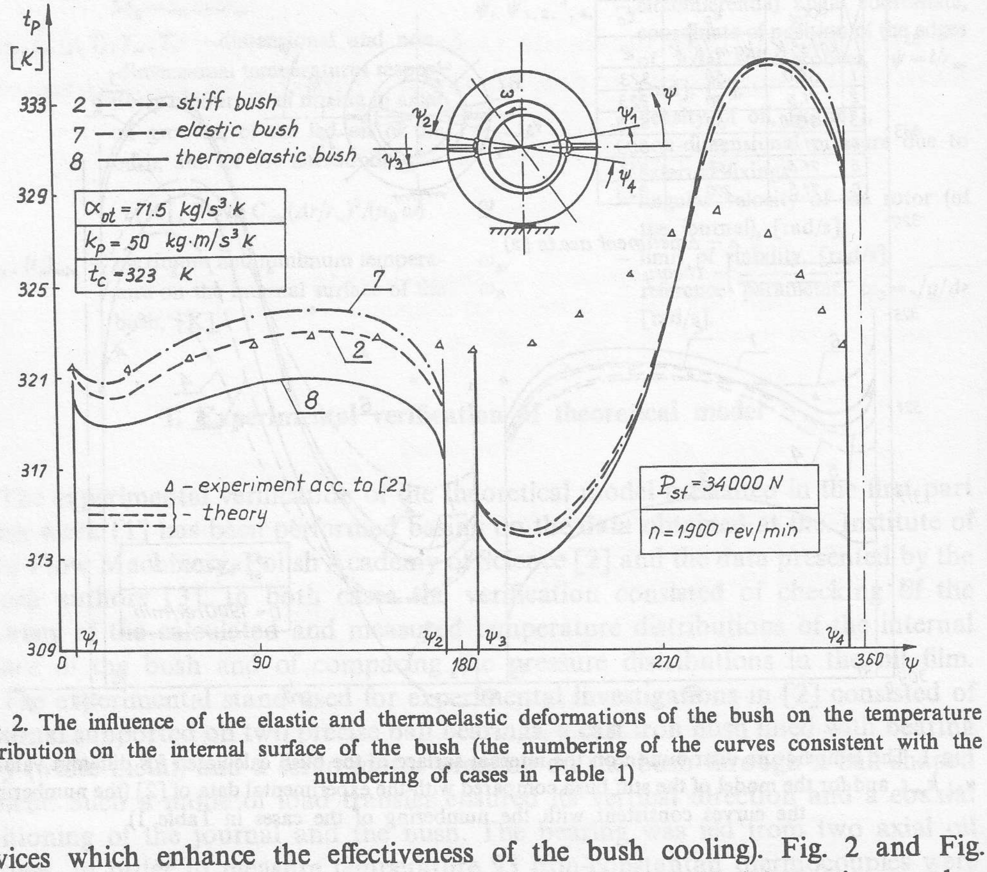


Fig. 2. The influence of the elastic and thermoelastic deformations of the bush on the temperature distribution on the internal surface of the bush (the numbering of the curves consistent with the numbering of cases in Table 1)

devices which enhance the effectiveness of the bush cooling). Fig. 2 and Fig. 3 illustrate the elastic deformations (calculations with thermal deformations neglected) and the thermoelastic deformations (calculations with both thermal and elastic deformations considered) of the bush. Fig. 3 shows the calculated clearance and the verification of the pressure distribution in the oil film. The comparison of the basic static characteristics for all the above mentioned cases is presented in Table 1. A model of the bush in the form of a thick-walled ring fixed on the external surface on bottom sector of 10° has been assumed for calculation. It was also assumed that the relation between oil dynamic viscosity (Z-26) and temperature is given by an empirical dependance

$$\mu = \frac{4.3066}{t\text{-}273} - 0.06366 \ \mathrm{NS/m^2} \quad \text{for } t \leqslant 313 \ \mathrm{K},$$

$$\mu = \frac{27859}{t-273} - 0.02565 \ \mathrm{NS/m^2} \quad \text{for } t > 313 \ \mathrm{K}.$$

The analysis of the above mentioned figures and Table 1 yields the following conclusions:

1. In all calculated cases, despite of the assumed value of α_{0t}, K_p and t_c coefficients and the assumed model of the bush, the theoretical differences between the maximum temperature $(t_p)_{max}$ (on internal surface of bush) and the minimum temperature $(t_p)_{min}$ are greater than the corresponding experimental values.

2. A particularly great difference between the theoretical and experimental distribution is observed around the left axial oil groove (the one with edges ψ_2, ψ_3) and close to the place of minimum lubricating film thickness ($\psi = 300 \div 340°$).

3. Temperature and pressure distributions are rather slightly affected by the elastic and thermal deformations of the bush at the same bearing load ($p_{av} = 21.5 \times 10^5$ N/m²). This refers also to the static characteristics.

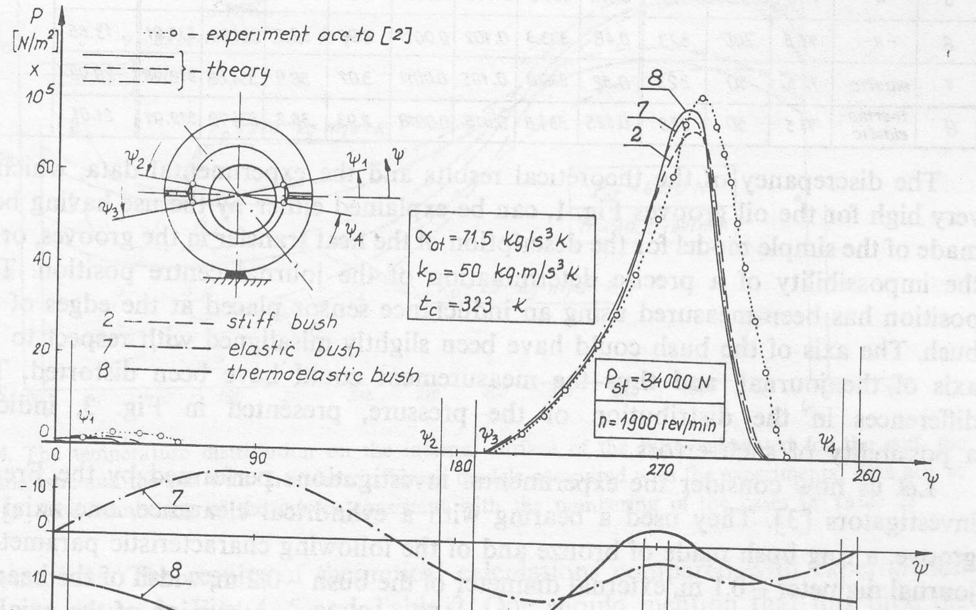


Fig. 3. The pressure distribution in the oil film and the distribution of lubricating gap deformations calculated for the stiff, the elastic and the thermoelastic bush models compared with the measured pressure distribution of ref [2] (the numbering of the curves consistent with the numbering of the cases in Table 1)

4. The influence of the coefficients α_{0t}, k_p on the basic static characteristics of the bearing such as friction power of oil expense, the minimum thickness of the lubricating clearance is small despite the fact that the values of these coefficients highly influence the temperature distributions on the internal surface of the bush. However, the influence of the journal temperature t_c proved to be significant.

Table 1

Calculated basic static characteristics of the experimental bearing (bearing geometry acc. to [2])

$P_{st}=34000$ N ($p_{av}=21.5\times10^5$ N/m²), $n=1900$ rev/min, $D=0.14$ m, $L/D=0.8$, $\Delta r=210\times10^{-6}$ m														
Nº of case	Bush model in calculations	α_{ot} $\frac{kg}{s^3 K}$	k_p $\frac{kg\,m}{s^3 K}$	t_c K	ε —	γ o	q_c m³/s $\times10^{-3}$	q_w m³/s $\times10^{-3}$	g_M kW	h_{min} m $\times10^{-6}$	$(t_p)_{max}$ K	$(t_p)_{min}$ K	$(t_p)_{max}-(t_p)_{min}$ theory K	$(t_p)_{max}-(t_p)_{min}$ experiment K
1	stiff	36.0	50	323	0.49	334	0.102	0.0017	3.0	37.9	335.09	315.11	19.98	
2	–II–	71.5	50	323	0.49	334	0.101	0.0017	3.0	37.9	334,71	314.35	20.36	
3	–II–	715.0	50	323	0.49	334.6	0.095	0.0017	2.99	38.5	330.65	307.97	22.68	
4	–II–	71.5	50	303	0.44	346.7	0.115	0.0012	6.6	59.1	325.14	312.61	12.53	6.9
5	–II–	71.5	50	313	0.475	339.9	0.108	0.0014	4.02	46.5	329.05	306.87	16.18	
6	–II–	71.5	200	323	0.48	333.3	0.102	0.0017	2.96	39.0	331.36	317.81	13.55	
7	elastic	71.5	50	323	0.52	330.8	0.105	0.0017	3.07	36.9	334,88	313.96	20.92	
8	thermo-elastic	71.5	50	323	0.525	334.6	0.115	0.0018	2.93	36.2	334.78	312.97	21.81	

The discrepancy of the theoretical results and the experimental data, which is very high for the oil grooves Fig. 1, can be explained either by the use having been made of the simple model for the description of the heat transfer in the grooves, or by the impossibility of a precise determination of the journal centre position. This position has been measured using an inductance sensor placed at the edges of the bush. The axis of the bush could have been slightly misaligned with respect to the axis of the journal, and thus the measurement could have been distorted. The differences in the distribution of the pressure, presented in Fig. 3, indicate a possibility of such errors.

Let us now consider the experimental investigations performed by the French investigators [3]. They used a bearing with a cylindrical clearance, one axial oil groove, a ring bush made of bronze and of the following characteristic parameters: journal diameter – 0.1 m, external diameter of the bush – 0.2 m, width of the bearing – 0.08 m ($L/D \cong 0.8$); radial clearance $\Delta r = 152\times10^{-6}$ m, positioning of the axial oil groove edge $\psi_1 = 99°$, $\psi_2 = 441°$, the temperature of the oil supplied – 313 K, ambient temperature – 313 K, feeding pressure 0.7×10^5 N/m², dynamic viscosity of oil at the temperature of 313 K – 0.0277 Ns/m².

In reference [3] there are presented results of experimental verification of one of the cases examined, i.e. $P_{st} = 6000$ N and $n = 4000$ rev/min. For the calculations the following values of material coefficients were taken: $\rho = 860$ kg/m³, $C_{w\tau} = 2000$

m^2/s^2 K, $k_{ol}=0.13$ kg m/s^3 K, $E=1.1\times10^{11}$ N/m^2, $\alpha_t=20\times10^{-6}$ 1/K, $\alpha_{0t}=80$ kg/s^3 K, $k_p=25$ kg m/s^3 K and $k_p=250$ kg m/s^3 K (according to [3]), whereas the temperature of the journal was taken to be $t_c=316.8$ K (i.e., the temperature on the journal edges taken by the by the authors of [3] $t_c=316$ K plus some estimated, average increase $\sim$ 0.8 K). The viscosity-temperature characteristics of oil used in the investigations is as follows [3]

$$\frac{\mu}{\mu_0}=3.287-3.064\,\frac{t-273}{t_0-273}+0.777\,\frac{(t-273)^2}{(t_0-273)^2}\,.$$

The pressure in lubricating film was measured with the aid of a precise manometer and holes in the lubricating clearance, whereas the temperatures were measured with Iron-Constantan termocouples built-in $\sim$ 0.5 mm beneath the slider surface. The calculations were performed for two values of coefficient k_p, for the stiff bush model and for the deformable bush model including elastic and thermal

Fig. 4. The temperature distribution on the internal surface of the bush calculated for the stiff, the thermoelastic and the stiff with "heat barrier" bush models compared with the experimental data acc. to [3] (the numbering of the curves consistent with the numbering of the cases in Table 2)

deformations. The results of theoretical calculations compared with experimental data are shown in Figs 4, 5 and Table 2. One should mention that this time the calculated temperature difference is smaller than the corresponding difference measured experimentally; this is just the opposite situation to that described before. In Fig. 4, 5 and in Table 2 one can also find calculated results for a bearing with a so called "heat barrier". The term "heat barrier" means that the calculations were performed for a model of bush in form of a ring with a piece of width cut-out which simulates the existence of an axial oil groove. Thus one can eliminate the heat

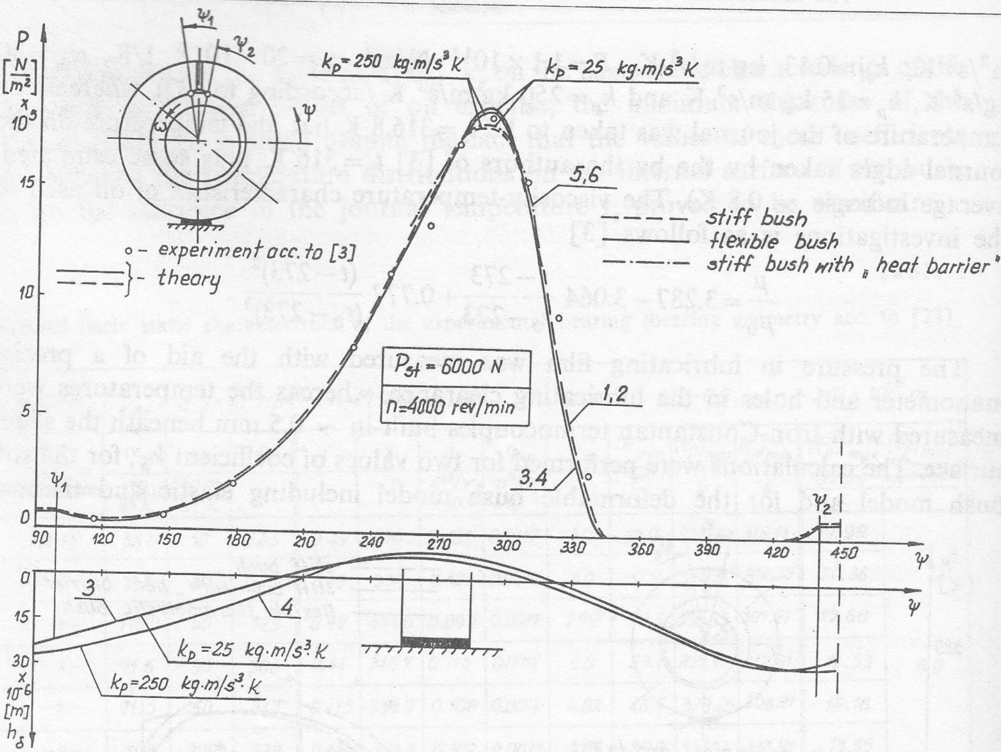


Fig. 5. The pressure distribution in the oil film and the distribution of the lubrication clearance deformation calculated for the stiff, thermoelastic and the stiff with "heat barrier" bush model and for the two values of k_p compared with the measured pressure distribution given in [3] (the numbering of the curves consistent with the numbering of the cases in Table 2)

transfer through this part and simulate the real barrier for heat transfer due to the cooling influence of oil feeding channels, grooves and dividing planes. The condition for a periodical solution in the conduction equation cannot apply here. Such a model of a stiff bush does not include the changes in other operating conditions of the bearing. The figures and the tables clearly inducate that the best compatibility with the experiment was achieved in case of a bearing with a"heat barrier", The verification mentioned above has proved that the thermoelastic deformations of the bush ($p_{av} = 7.5 \times 10^5$ N/m²) and of the coefficient k_p have little influence on the basic static characteristics. A much better fit has been obtained for the pressure distribution in the oil film. This can be explained by the fact that in case of a bearing with a cylindrical clearance it is much easier to determine precisely the parameters of the lubricating clearance.

With the case of a "heat barrier" excluded, one can notice quite significant difference in the neighbourhood of the oil groove (Fig. 4). This stresses the need for a more adequate description of the phenomena occurring in the oil grooves.

Calculations for other cases of load and rotary velocities of experimental bearings described in [2, 3] have been also performed. The temperature curves and the differences were similar to those obtained in cases described above and that is why, in order to make this work more concise, they are not presented here in details.

Table 2

Calculated basic static characteristic of the experimental bearing (bearing geometry acc. to [3])

$P_{ot} = 6000$ N ($p_{av} = 7.5 \times 10^5$ N/m²), $n = 4000$ rev/min, $D = 0.1$ m, $L/D = 0.8$, $\Delta r = 152 \times 10^{-6}$ m

Nº of case	Busch model in calculations	α_{ot} $\frac{kg}{s^3 K}$	k_p $\frac{kg\,m}{s^3 K}$	t_c K	ε –	γ °	q_c m³/s ×10⁻³	q_w m³/s ×10⁻³	q_M kW	h_{min} m ×10⁻⁶	$(t_p)_{max}$ K	$(t_p)_{min}$ K	$(t_p)_{max}-(t_p)_{min}$ theory K	$(t_p)_{max}-(t_p)_{min}$ experiment K
1	stiff	80	25	316.8	0.545	327.4	0.137	0.0033	2.09	69.3	330.2	319.2	11.0	
2	-II-	80	250	316.8	0.54	327.1	0.140	0.0033	2.08	70.0	327.6	321.7	5.9	
3	thermo-elastic	80	25	316.8	0.575	335.4	0.154	0.0036	2.02	68.9	329.7	318.6	11.1	16.2
4	-II-	80	250	316.8	0.56	336.1	0.160	0.0036	2.01	70.0	327.0	321.1	5.9	
5	stiff with „barrier"	80	25	316.8	0.54	328.3	0.128	0.0029	2.08	70.1	329.3	313.4	15.9	
6	-II-	80	250	316.8	0.535	327.7	0.128	0.0029	2.07	70.8	326.5	314.6	11.9	

The calculations, which have been compared with the experiment, have been performed basing on a too small number of analysed cases. Furtheron, one could feel the lack of a number of fundamental experimental data connected with the static characteristics. Thus, the complete verification of the theoretical model was not possible. The character of the discrepancies between the theoretical and experimental data has been different for the two cases analysed in the paper. This emphasises the complex nature of the phenomena occurring in slider bearing and the difficulties in their mathematical description.

However, basing on the performed investigations one could conclude, that the assumption of a relatively simple heat transfer conditions (e.g. a model of the bush in form of a ring without oil grooves, feeding channels and dividing planes) can lead to relatively big differences in the temperature distributions on the internal surface of the bush. This can be confirmed by the "heat barrier" calculations, or by the relatively high temperature differences around the axial oil grooves. The heat transfer process in the axial oil grooves and particularly the mixing of the hot and cold oil certainly need a more detailed investigation.

On the other hand, the performed calculations proved that even relatively big differences in the temperature distributions on the internal surface of the bush do not cause significant changes in the basic static characteristics of the bearing oil flow, friction power, minimum thickness of the oil film. Much more evident is the influence of the journal temperature t_c. This fact is rather inconvenient since a precise determination of the temperature of the journal is still very difficult.

2. Theoretical investigations

As it follows from the considerations presented in part I [1] the basic static and dynamic characteristics of the bearing depend on a large number of non-dimensional parameters. Thus, a general theoretical analysis, performed only on the basis of the fundamental similiarity numbers (e.g., criteria numbers such as Sommerfeld, Nusselt, Peclet or $\bar{E}, \alpha_t^*, T_0, \Pi_z$ etc.) is an arduous task. For further consideration let us take a given theoretical bearing which can be a representative for the whole series of type of bearings of diameter $D=0.125$ m$\div$0.450 m used in large machines and power devices. The features of the chosen theoretical bearings are as follows:

- journal diameter $D=0.125$ m and $D=0.450$ m,
- rotation speed $n=3000$ rev/min,
- average load $p_{av}=P_{st}/(LD)=40\times10^5$ N/m^2,
- relative clearance $\Delta r/r_w=1.55\times10^{-3}$ (cylindrical clearance),
- ratio of the width and diameter of the bearing $L/D=0.8$,
- two axial oil grooves (circumferential span – 60°, width – 0,75),
- feeding presure $p_0=1\times10^5$ N/m^2,
- different variants of bush fixing in bearings supports.

For calculations it has been also assumed: $t_0=313$ K, $t_z=293$ K, $\mu_0=0.05$ Ns/m^2, $b_0=0.045$ 1/K, $\rho=900$ kg/m^3, $C_{wt}=2000$ m^2/s^2 K, $k_{ol}=0.135$ kgm/s^3 K, $k_p=50$ kg m/s^3 K, $\alpha_{0t}=80$ kg/s^3 K, $E=2.1\times10^{11}$ N/m^2, $\alpha_t=11.5\times10^{-6}$ 1/K and the function for oil viscosity *vs.* temperature $\mu/\mu_0=\exp(-b_0(t-t_0))$. The journal temperature t_c has been taken as the mean of the temperature distribution on the internal surface of the bush and in the axial oil grooves.

Thus, a bearing with a cylindrical clearance and a steel bush fixed in different ways has been chosen for the investigation. The dynamical properties of the bearing have been calculated basing on the assumptions presented in Part I of the present paper. This means that the linear relations connecting the forces with the displacements and the simple model of the rotor have been used. Thus one can gauge the dynamical properties of a bearing by means of its stable operation limit ω_{gr}/ω_0, the damping factor for the system u/ω_0, and the self-frequencies ϑ/ω_0. Typical values of the material coefficients for a steel bush and motor oil were used in the calculations.

The value of $So_0=0.6$ has been taken for bearing of $D=0.125$ m diameter and $So_0=1.12$ for the bearing of $D=0.450$ m diameter.

It may be interesting, particularly from the technological point of view to find an answer to the following question: what is the influence of the bush fixing on the static and dynamic properties of a rotor-bearing system, when one assumes the above defined geometrical and operational features of the sliding bearings? Furtheron, one may ask what happen for a given bearing load on stability reserve of the system. Or in other words, what is the boundary rotational velocity of the rotor and how could the damping of the system change in accordance with a change of the rotational velocity or the mode of the bush fixing.

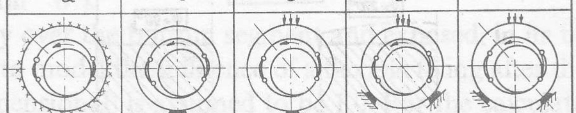
The assumption of a simple model of the rotor (stiff, single-mass, symmetrical rotor) allows one to confine the considerations to the properties of the oil film only. Answers to the above stated question are the subject of the theoretical investigations which will be presented in the next sections.

2.1. Influence of the bush fixing mode

The bush of a bearing is considered as a symmetrical thick-walled ring without dividing planes, bearings and oil channels. The model of the bush in the assumed system of coordinates X_p, Y_p, Z_p together with finite element digitization is presented in Fig. 3 of the first part of this paper [1].

Table 3

Influence of bush fixing mode. Basic static and dynamic characteristics

		$n = 3000$ rev/min ($\omega = 314.16$ rd/s)						$p_0 = 1 \times 10^5$ N/m^2			
		1 – $So_0 = 0.6$ $D = 0.125$ m						$p_{av} = 40 \times 10^5$ N/m^2			
		2 – $So_0 = 1.12$ $D = 0.450$ m									
		Thermoelastic bush model								Elastic model	
		a		b		c		d		e	
											
ε	1	0.547		0.815		0.937		1.157		0.79	
	2	0.635		0.890		—		—		—	
γ	1	309.4		324.1		328.6		268.4		296.2	
	2	311.7		322.2		—		—		—	
Q_c	1	0.102		0.183		0.211		0.128		0.100	
	2	0.108		0.195		—		—		—	
Q_w	1	0.0159		0.0179		0.0190		0.0111		0.0149	
	2	0.0187		0.0209		—		—		—	
G_M	1	1.56		1.81		1.848		2.746		1.63	
	2	1.41		1.68		—		—		—	
H_{min}	1	0.219		0.224		0.196		0.086		0.270	
	2	0.146		0.146		—		—		—	
T_c	1	16.97		15.08		15.31		14.74		15.68	
	2	18.81		17.01		—		—		—	
$(T_p)_{max}$	1	21.69		20.79		21.01		21.30		21.00	
	2	30.67		30.26		—		—		—	
$(T_p)_{min}$	1	14.68		13.37		13.69		11.75		13.74	
	2	15.16		13.62		—		—		—	
$\frac{\omega_{gr}}{\omega}$	1	2.26		1.41		∞		5.85		2.56	
	2	∞		1.03		—		—		—	
$\frac{U_{1,2}}{\omega}$	1	0.247	—	0.051	—	0.0476	—	1.65	—	0.211	—
	2	0.048	1.503	0.0006	1.096	—	—	—	—	—	—
$\frac{\vartheta_{1,2}}{\omega}$	1	0.997	—	0.905	—	0.761	—	1.99	—	0.941	—
	2	0.645	0.915	0.537	1.138	—	—	—	—	—	—

A more difficult problem has emerged when the influence of external fixings of the bush was to be modelled. The simplest solution of the problem is to assume that the bush external surface is non-deformable i.e., an infinitely stiff fixing encloses bush over the whole perimeter. The opposite situation occurs when one assumes that the bush is fixed only by a small segment in the bottom part and the rest of the bush

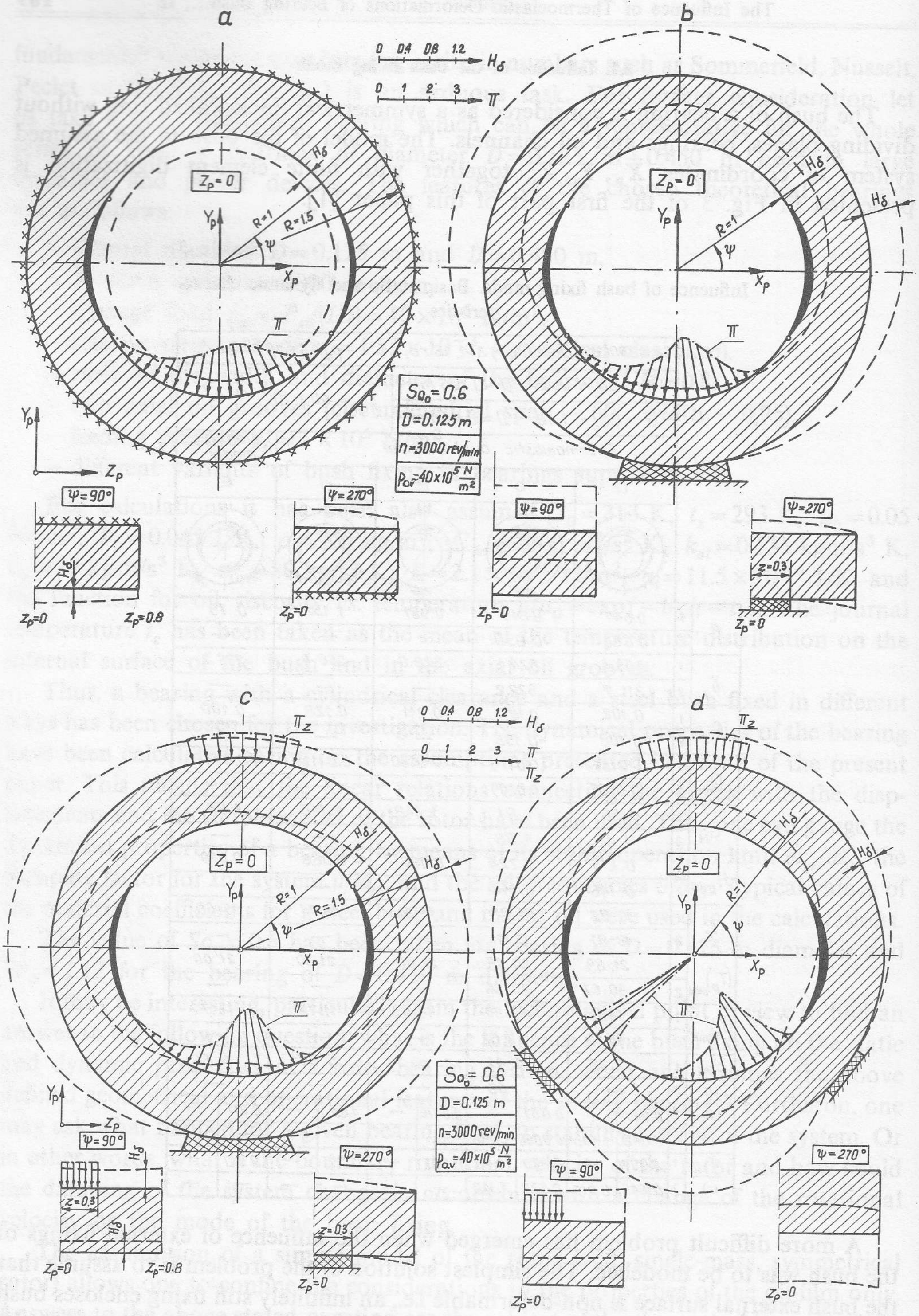


Fig. 6. The distribution of the deformations of the bush and the distribution of the pressure of the oil film depending on bush fixing mode (the numbering of the cases according to Table 3)

external surface can deform freely. One may assume that the influence of fixing elements equals to the reaction to some external pressures Π_z. However, the external surface of the bush can translate within fixing range. Finally, one can assume that some of the fixing elements are stiff (for example, the bottom ones forming thus a natural reference base) whereas the other ones are flexible.

All the assumptions mentioned above give only an approximate description of the conditions in a real bush fixing. Since the fixing elements can also deform in a way which is very difficult to determine, theoretical modelling of a bush fixing is particularly difficult. Thus, the determination of the value of Π_z depends on the initial clamp of the bush and in this way on the conditions of assembling.

Due to of the lack of more precise data four simple theoretical models have been selected the further consideration:

– Bush fixed stiffly over the whole external surface (version "a").

– Bush fixed stiffly only over the bottom segment of the external surface 30° span and 0.3 width (version "b").

– Bush fixed stiffly over the bottom segment and exposed, in its upper part to an external pressure Π_z (to model the influence of a flexible fixing after the initial clamp). The value Π_z (nondimensional) is assumed to be 1.0. For the calculated casses this is around 1/3 of the maximum value of the hydrodynamic pressures $\Pi_{\max}$. It has been assumed that the pressure Π_z acts on the upper segment of the external surface of the bush of the span of 30° and the width of 0.3 (version "c").

– Bush fixed stiffly over two bottom segments of 15° span and 0.3 width, each symmetrically divided from the vertical line at angle 52.5° instead of lying on the vertical axis of the bearing. In its upper part, as in the version "c", the bush is under pressures $\Pi_z=1$ (version "d").

The results are presented in Table 3 and in Fig. 6. It is clearly visible that the basic static and dynamic characteristics vary within a wide range depending on the fixing variant. Particularly interesting is version "d". The deformations of the bush are so big that the hydrodynamic pressure zone is broken into two parts (Fig. 6d). Thus, the bearing becomes effectively the one with two convergent zones and of definitely different characteristics. In view of the static properties this version is the worst one. The very small value of $H_{\min}$ together with a great friction power G_M leads to a dangerous situation in which the bearing may seize. It is particularly evident in case of great bearing diameters. The value $H_{\min}$ is so small and the temperatures are so high that there are problems with a numerical computation of this variant of fixing with diameter $D=0.450$ m. That is the reason why Table 3 does include results for this case.

On the other hand, as far as the dynamic properties are concerned the version "d" is definitely the best one. The damping of the system is one order of magnitude greater than in the other variants and the free-vibration frequency ϑ_1/ω is also high. Table 3 includes also the results of a calculation for the version "d" but without thermal deformations of the bush (elastic model). Comparing the results one may conclude that in the cases under consideration the most important are the thermal deformations of the bush. The same conclusion can be drawn after the analysis of the thermoelastic deformations of the bush in Fig. 6a-d.

2.2 Influence of rotational velocity of rotor

Fet us examine how the dynamical properties of the rotor-bearing system with a constant load $p_{av}=40\times10^5$ N/m^2, depend on the rotational velocity of the rotor. The investigations have been carried out for different theoretical models of the lubricating film (isothermal or diathermal model) and the external fixings of the bush. The damping and free vibration frequency curves obtained in this way determine the dynamic condition of the system and illustrate the influence of chosen geometrical properties of the bearing. Calculations have been performed for diameter $D=0.125$ m, $So_0=0.6$, two axial oil grooves and two versions of the external fixings of the bush (version "b" and "d" acc. to diagrams in Table 3). The results of the calculation are presented in Fig. 7. It is worth mentioning that the character of damping curve for version "d" of bush fixing is quantitatively different. Stability

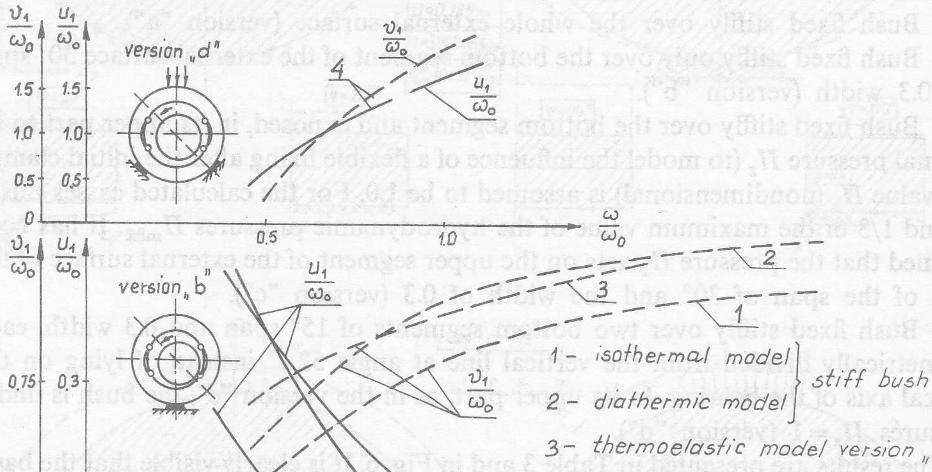


Fig. 7. The curves for the system damping and the frequency of free vibrations calculated for different models of oil film and different bush models.

limits, within the range of rotational velocity under consideration, do not exist and, what is more important, system damping u_1/ω_0 is very high (large reserve of stability). A bush fixing of the version "b" makes the dynamic properties of the system worse when one compares them with those for the stiff bush.

For a given operational angular velocity of the rotor (for example: $\omega_r=314.16$ rd/s$\rightarrow\omega_r/\omega_0=0.987$) one can easily determine from Fig. 7 the "distance" from the boundary angular velocity ω_{gr}/ω_0. For a bush fixing of the version "b" $\omega_{gr}/\omega_0\cong1.33$, i.e. this boundary lies dangerously close to the operational velocity ω_r. However, for a bush fixing of the version "d" stability limits lie far behind the range of the rotor velocity under consideration. These limits exist only from the theoretical point of view since in reality the bearing may seize much earlier due to the high temperature and small value of $H_{\min}$. It is worth mentioning here, that values of ω_{gr}/ω presented in Tables 3, do not equal the ultimate rotor speed for a given bearing load and are meant as the stability limits for an unchanged shape of the lubricating gap. The values of ω_{gr}/ω presented in tables should be under-stood as the stability boundary for such an increase of the load of the bearing (for the increase of rotor velocity) that the values of ε and of γ remain unchanged. It is a case rarely taking place a technical application despite of its significance from the point of view of the oil film properties. A direct examination of the distance from the stability limits is possible only with the aid of damping curves such as the ones in Fig. 7.

3. Conclusions

The results presented in this paper, obtained for an examplary series of types of bearings, indicate that bush thermoelastic deformations and its fixing mode have great influence on the basic static and dynamic characteristics of a bearing. Particularly interesting is the influence of the external fixings of the bush on the dynamic properties of the bearing, especially in view of the small amount of data in the literature referring to this problem. The mode of the bush fixing can, in some cases, definitely change the dynamic properties of the whole rotor-bearing system. The calculations performed indicate that the thermal deformations of the bush are the most important factor in the development of the deformations and the characteristics of the bearing.

The obtained results do not allow for more general conclusions about the optimum fixing mode of slider bearings. This has not been the aim of the present paper. The presented computational examples show the necessity of performing, for each particular case, a detailed analysis of the conditions for the assembling and the operation of a bearing.

The main purpose of the present work was to point out the possibilities and limits of a theoretical description of the phenomena which take place in a complex tribologic system like the rotor-lubricating film-bush-external fixings. The theoretical model presented in this paper, its experimental verification, computer program and discussion of the obtained results have been aimed at proving that despite the evident discrepancies between theoretical model, its mathematical description and reality the obtained data may be a source of valuable information for both investigators of the properties of the oil film and designers of bearings.

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Wpływ termosprężystych deformacji panwi oraz jej utwierdzeń zewnętrznych na statyczne i dynamiczne własności poprzecznych łożysk ślizgowych

Część II. Weryfikacja eksperymentalna i badania teoretyczne

Streszczenie

W pracy przedstawiono wyniki weryfikacji eksperymentalnej modelu teoretycznego łożyska przedstawionego w części I. Weryfikacja ta polega na porównaniu rozkładów temperatury na powierzchni panwi i ciśnieniu w filmie, obliczonych i pomierzonych na stanowisku doświadczalnym. Przedstawiono także wyniki badań teoretycznych przeprowadzonych na podstawie opracowanego modelu i programu komputerowego. Przeanalizowano wpływ deformacji sprężystych i termicznych panwi oraz sposobu jej utwierdzenia na podstawie charakterystyki statycznej i dynamicznej prostego układu wirnik – łożyska. Badania przeprowadzono na przykładzie typoszeregu łożysk turbinowych o średnicy ∅ 125÷450 mm.

Влияние термоупругих деформаций вкладыша и его внешних утверждений на статические и динамические свойства поперечных подшипников скольжения

Часть II. Экспериментальная проверка и теоретические исследования

Резюме

В работе представлены результаты экспериментальной проверки теоретической модели подшипника представленного в части I. Эта проверка полагалаць на сравнении распределений температуры на внутренней поверхности вкладыша и давления в масляной плёнке, вычислённых и измерённых на экспериментальном стенде. Представлены также результаты теоретических исследований проведённых на основе разработанных модели и вычислительной программы. Проанализировано влияние упругих и термических деформаций вкладыша и способа его утверждения на основе статической и динамической характеристик простой системы ротор – подшипники. Исследования произведены для примера типоряда турбинных подшипников диаметром 125 – 450 мм.