PRESSURE AND CAPACITY CHANGES IN SLIDE JOURNAL BEARING FOR LAMINAR UNSTEADY OIL FLOW

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Abstract

This paper presents numerical solutions of the modified Reynolds equation of laminar unsteady lubrication in a cylindrical slide journal bearing. Laminar, unsteady oil flow is performed during periodic and unperiodic perturbations of bearing load or is caused by the changes of gap height in the time. Above perturbations occur during the starting and stopping of machine. The particular solutions are limited to the isothermal models of bearing with infinite length and lubricated by Newtonian oil with the dynamic viscosity dependent on pressure. The disturbances are related to the unsteady velocity of oil flow on the sleeve and on the surface Diagram shows the results of hydrodynamic pressure and capacity forces in the dimensionless form in time intervals of displacement duration.

Keywords: *journal bearing, unsteady laminar lubrication, pressure distribution, capacity forces*

1. Introduction

This article refers to the unsteady, laminar flows issue, in which [4,5] modified Reynolds number $Re^*=Re\psi$ is smaller than 2 or Taylor number $Ty=Re\sqrt{\psi}$ is smaller than 41,1. Laminar, unsteady oil flow is performed during periodic and unperiodic perturbations of bearing load or is caused by the changes of gap height in the time. In this article following problems were also mentioned: lubricated oil disturbance velocity on the pin and on the bearing shell. Velocity perturbations of oil flow on the pin, are caused by torsion vibrations during the rotary movement of the shaft. Perturbations are proportional to torsional vibration amplitude and to pin radius of the shaft. Oil velocity perturbations on the shell surface can be caused by rotary vibration of the shell together with bearing casing. This movement can be consider as kinematic constraint for whole bearing friction node. Isothermal bearing model can be approximate to bearing operation in friction node under steady-state thermal load conditions for example bearing in generating set on ship. In considered model flow [1,5,6] we assume small unsteady disturbances and accordingly to the laminar flow, oil velocity V_i^* and pressure p_1^* are sum of time dependent quantities $\tilde{V}_i$; $\tilde{p}_1$ and time independent quantities $V_i; p_1$ from time [1,2,3] in following form:

$$V_i^* = V_i + \tilde{V}_i \qquad i = 1,2,3$$
$$p_1^* = p_1 + \tilde{p} \tag{1}$$

Unsteady components of dimensionless oil velocity and pressure we show [1,5] in following form of infinite series :

$$\tilde{V}_i(\varphi;r_1;z_1;t_1) = \sum_{k=1}^{\infty} V_i^{(k)}(\varphi;r_1;z_1)\exp(j\,k\omega_0\,t_0 t_1)$$

$$\tilde{p}_1(\varphi;z_1;t_1) = \sum_{k=1}^{\infty} p_1^{(k)}(\varphi;z_1)\exp(j\,k\omega_0\,t_0 t_1) \qquad i=1,2,3 \qquad (2)$$

where:

ω_0 – angular velocity perturbations in unsteady flow,
j - imaginary unit: $j=\sqrt{-1}$.

Reynolds equation describing total dimensionless pressure p_1^* (sum steady and unsteady components) in oil journal bearing gap [1,2] by unsteady, laminar, isotherm Newtonian flow along with disturbances of peripheral velocity V_{10} on the journal and V_{1h} on the sleeve and disturbances of velocity on journal length V_{30} on the journal and V_{3h} on the sleeve has following form:

$$\frac{\partial}{\partial\varphi}\left\{\frac{(h_1)^3}{e^{Kp_1}}\left[\frac{\partial p_1^*}{\partial\varphi} - K\left(p_1^* - p_1\right)\frac{\partial p_1}{\partial\varphi}\right]\right\} + \frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left\{\frac{(h_1)^3}{e^{Kp_1}}\left[\frac{\partial p_1^*}{\partial z_1} - K\left(p_1^* - p_1\right)\frac{\partial p_1}{\partial z_1}\right]\right\} =$$
$$= 6\frac{\partial h_1}{\partial\varphi} + \frac{1}{2}\frac{\rho_1}{\eta_1}\mathrm{Re}^*\,n\left\{\frac{\partial}{\partial\varphi}\left[h_1^3\left(V_{10}+V_{1h}\right)\right] + \frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left[h_1^3\left(V_{30}+V_{3h}\right)\right]\right\}\sum_{k=1}^{\infty}A_k + \qquad (3)$$
$$-6\left\{\frac{\partial}{\partial\varphi}\left[h_1\left(V_{10}+V_{1h}\right)\right] + \frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left[h_1\left(V_{30}+V_{3h}\right)\right] - 2\left(V_{1h}\frac{\partial h_1}{\partial\varphi} + \frac{1}{L_1^2}V_{3h}\frac{\partial h_1}{\partial z_1}\right)\right\}\sum_{k=1}^{\infty}B_k$$

where $0 \le \varphi \le \varphi_e$; $0 \le r_1 \le h_{p1}$; $-1 \le z_1 \le 1$; $0 \le t_1 \le t_k$; $p_1^* = p_1^*(\varphi; z_1; t_1)$

Components of oil velocity V_φ, V_r, V_z in cylindrical co-ordinates r,φ,z have presented as V_1, V_2, V_3 in dimensionless form:

$$V_\varphi = UV_1^* \qquad V_r = \psi UV_2^* \qquad V_z = \frac{U}{L_1}V_3^* \qquad (4)$$

Dynamic oil viscosity η is depended on pressure by Barrus formula [1] and has following form:

$$\eta = \eta_0 e^{\alpha(p-p_a)} \approx \eta_0 e^{\alpha p} = \eta_0\eta_1 \qquad (5)$$

where:

η_o- the dynamic oil viscosity for atmospheric pressure $p = p_a \approx 0$
η – the dynamic oil viscosity function,
α – the pressure influence piesocoefficient of the oil viscosity ,
η_1 – dimensionless dynamic viscosity depending on pressure $\eta_1=\exp(\alpha p)$.

Parameter K characterize dimensionless oil dynamic viscosity change caused by pressure(if K=0 dynamic oil viscosity is constant and independent from pressure):

$$K = \alpha\,p_0 \qquad p_0 = \frac{U\eta_0}{\psi^2 R} \qquad (6)$$

where:

p_0 - characteristic value of pressure

Sum for series $\sum_{k=1}^{\infty} A_k$ and $\sum_{k=1}^{\infty} B_k$ in right hand side of Reynolds equation (3) are results from conservation of the momentum solutions and were define in work [1,2]. Nomenclature has been placed in the end of the paper. Specific explanation to above Reynolds equation were define in works [1,2,3]. Quantity of oil peripheral velocity perturbations on the whirling pin surface with ω velocity caused by forced torsion vibration of shaft with the angular velocity ω_o and angular amplitude φ_0 can be present in the following dimensionless form:

$$V_{10} = \varphi_0 n \ , \qquad n = \frac{\omega_0}{\omega} , \tag{7}$$

In this torsion vibrations case of engine, n number depend on the cylinder number c and on engine type: two-stroke (s=2) or four-stroke (s=4):

$$n = \begin{cases} c & s = 2 \\ \dfrac{c}{2} & s = 4 \end{cases} , \tag{8}$$

Rest of the symbols and quantities which apply to Reynolds equation (3) have been precisely define and describe in work [1,2,3].

2. Pressure distributions

The equation solution (3) for bearing with infinity length determine total dimensionless hydrodynamic pressure function in following [2,3] (K>0) form:

$$\begin{aligned} p_1^*(\varphi) = \frac{1}{1 - Kp_{10}} \Bigg[& -\frac{1 - Kp_{10}}{K} \ln\left|1 - Kp_{10}\right| + p_{10}\left(V_{1h} - V_{10}\right)\sum_{k=1}^{\infty} B_k + \\ & + \frac{1}{2}\rho_1 \operatorname{Re}^* n\left(V_{10} + V_{1h}\right)\left(\varphi - h_{1e}^3 \int_0^{\varphi} \frac{d\varphi}{h_1^3} - K\int_0^{\varphi} p_{10} d\varphi\right)\sum_{k=1}^{\infty} A_k \Bigg] \end{aligned} , \tag{9}$$

$$\text{for} \qquad 0 \le \varphi \le \varphi_e , \quad 0 \le t_1 \le t_k , \quad p_1^* = p_1^*(\varphi; t_1)$$

Pressure p_{10} is located in the oil gap by steady flow and by constant oil dynamic viscosity (K=0):

$$p_{10}(\varphi) = 6\int_0^{\varphi} \frac{h_1(\varphi) - h_{1e}}{h_1^3(\varphi)} d\varphi \ , \qquad \text{for} \quad 0 \le \varphi \le \varphi_e \tag{10}$$

Dimensionless total pressure by disturb flow and by constant oil dynamic viscosity independent from pressure (K=0):

$$p_1^*(\varphi) = p_{10} + \frac{1}{2}\rho_1 \operatorname{Re}^* n\left(V_{10} + V_{1h}\right)\left(\varphi - h_{1e}^3 \int_0^{\varphi} \frac{d\varphi}{h_1^3}\right)\sum_{k=1}^{\infty} A_k + p_{10}\left(V_{1h} - V_{10}\right)\sum_{k=1}^{\infty} B_k \ , \tag{11}$$

Disturbance pressure in unsteady flow part can be presented with common formula for constant (K=0) and variable dynamic viscosity $(K \ge 0)$:

$$\tilde{p}_1 = \frac{1}{1-Kp_{10}}\left[\frac{1}{2}\rho_1 \mathrm{Re}^* n(V_{10}+V_{1h})\left(\varphi - h_{1e}^3\int_0^{\varphi}\frac{d\varphi}{h_1^3} - K\int_0^{\varphi} p_{10}d\varphi\right)\sum_{k=1}^{\infty} A_k + p_{10}(V_{1h}-V_{10})\sum_{k=1}^{\infty} B_k\right], \quad (12)$$

Presented equations (9),(11),(12), which describe total pressure course and perturbation pressure course, have following conclusions: pressure in dependence on velocity perturbations quantity V_1 and on direction with relation to pin peripheral velocity U. In this equations, two components occur in dependence on the sum and on the difference of velocity perturbations on the pin and on the bearing shell. That is way in presented graphs characteristic cases of perturbations with different and equal values were shown.

Numerical calculation results are presented by following tangential velocity perturbations:

1. velocity perturbations on the journal V_{10}=0,05 and on the sleeve V_{1h}=0,
2. velocity perturbations on the journal V_{10}=0,05 and on the sleeve V_{1h}=0,025,
3. velocity perturbations on the journal V_{10}=0,05 and on the sleeve V_{1h}=0,05,
4. velocity perturbations on the journal V_{10}=0,05 and on the sleeve V_{1h}=-0,05

In numerical calculation example, oil with constant density was assume, equivalent to quantity ρ_1=1. In presented calculation way an expression value is assumed $n\rho_1\mathrm{Re}^*$ = 12, what is approximately equivalent to force over first frequency torsion vibrations force of six cylinder engine shaft. This take place by laminar unsteady flow. Time of reference t_o is a period of velocity disturbances dispersion. Examples apply to bearing with constant dependent eccentricity λ=0,6.

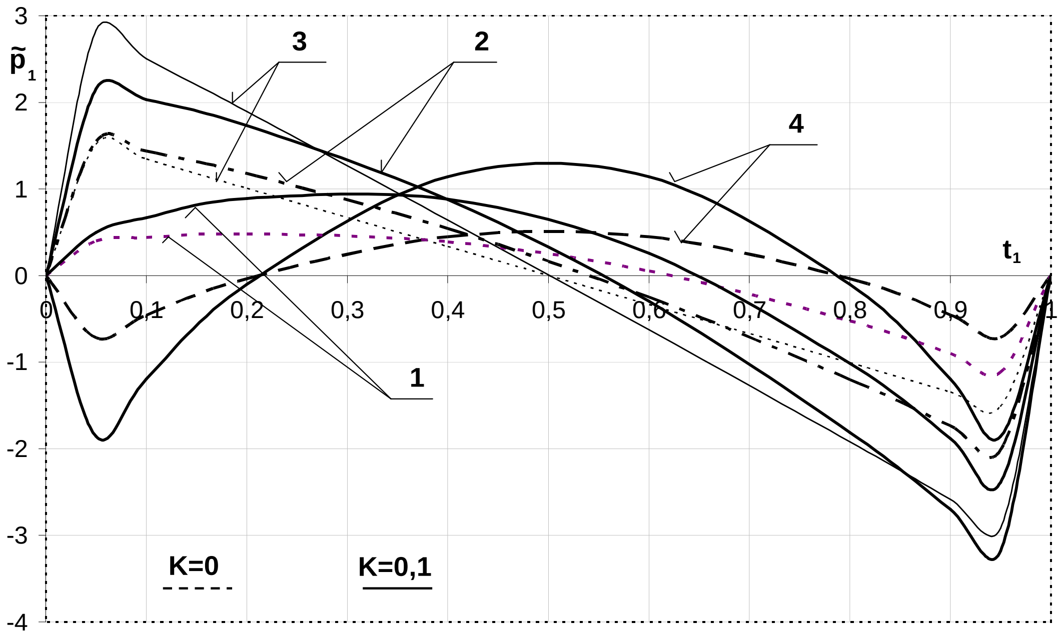

Fig.1. Pressure distributions $\tilde{p}_1$ in place φ=145° in the time t_1 for constant oil viscosity (K=0) and for oil viscosity in dependence on pressure (K=0,1) by velocity perturbations: 1)V_{10}=0,05; V_{1h}=0; 2) V_{10}=0,05; V_{1h}=0,025; 3) V_{10}=0,05; V_{1h}=0,05; 4) V_{10}=0,05; V_{1h}=-0,05

Unsteady pressure is changing due velocity perturbations time and it is in function of time and position on the journal. It is a periodic function with the following lasting period of velocity perturbation. Pressure perturbation course in point φ=145° on the journal surface in dimensionless time function in case of velocity perturbation on the journal and on the sleeve is presented by four variant on Fig. 1. Above graphs are made for constant viscosity and for viscosity in dependence on pressure where K=0,1. When oil velocity perturbations on the journal are compatible to journal tangential velocity, the perturbation pressure increase, otherwise the pressure decrease. In this case decrease is considerably bigger than increase and it last shorter than half of perturbation period. In case of velocity perturbation on the sleeve it is opposite. There is a lack of graphs for this example. Periods of pressure increase and decrease are non-symmetrical in case of different perturbation velocity values (graph 2). When perturbations velocity values are equal and directions are the same or opposite then the perturbation pressure is symmetrical in time (graph 3 and 4, Fig.1). Pressure perturbation distribution by wrapping angle is changing in time, giving in different time periods maximal or minimal pressure. Maximal and minimal pressure distribution in considered velocity perturbation examples are presented on Fig.2. In order to compare influence of viscosity variable in dependence on pressure (graph b),pressure distribution for oil with constant viscosity (K=0) which is independent from pressure, were plotted (graph a). In case where velocity perturbations on the pin and on the bearing shell have the same signs and pressure, perturbations values are maximum (graph 3). When viscosity is in dependence of pressure it causes an increase of steady pressure and perturbation pressure on both maximal and minimal pressure sides. Steady pressure flow sum up with perturbation pressure and total distribution of maximal and minimal pressure by bearing wrapping angle is received. This are the border pressure distribution for given type of perturbation.

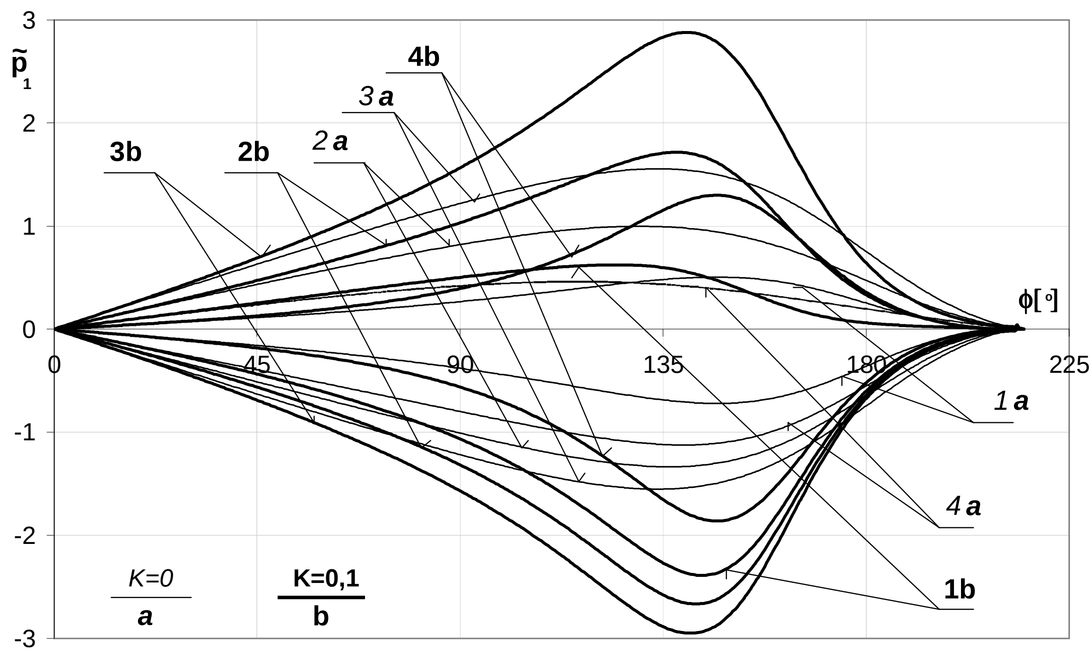

Fig.2. Unsteady part maximal and minimal pressure distributions $\tilde{p}_1$ in direction φ a) for constant oil viscosity (K=0), b) for oil viscosity in dependence on pressure (K=0,1) by velocity perturbations 1) V_{10}=0,05; V_{1h}=0; 2) V_{10}=0,05; V_{1h}=0,025. 3) V_{10}=0,05; V_{1h}=0,05;4) V_{10}=0,05; V_{1h}=-0,05

3. Capacity forces

Capacity force W for cylindrical slide journal bearing has following components W_x and W_y to be determined [2,5] in the local co-ordinate systems in Fig. 3. Thus dimensionless components W_{1x} and W_{1y} of capacity forces W_1 are as follows [2]:

$$W_{1x} = \frac{W_x}{W_0} = -\int_0^{\varphi_k} p_1 \cos\varphi d\varphi, \qquad W_{1y} = \frac{W_y}{W_0} = -\int_0^{\varphi_k} p_1 \sin\varphi d\varphi \ , \tag{13}$$

$$W_1 \equiv S_o = \sqrt{W_{1x}^2 + W_{1y}^2} = \frac{W}{W_0}$$

where:

W_0 - characteristic value of capacity force $W_0 \equiv 2Rbp_0$,
S_o – Sommerfeld Number for slide journal bearing.

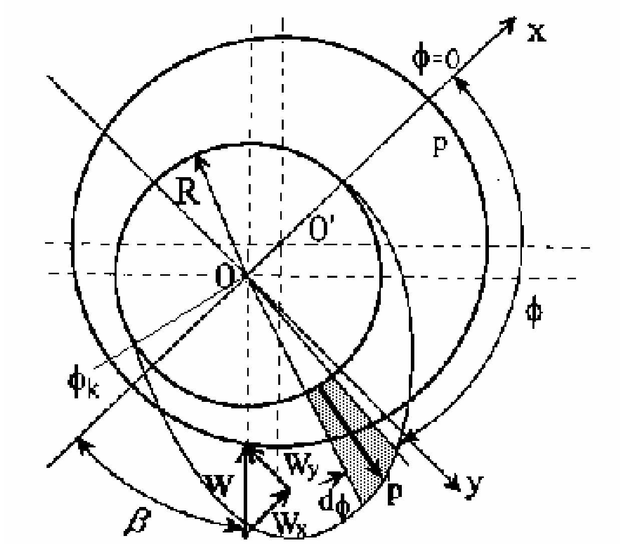

Fig. 3 Capacity force W and components W_x and W_y in the local co-ordinate system

Hydrodynamic capacity force change caused by the pressure perturbation is calculate from:

$$\tilde{W}_1 = W_1^* - W_1 \ , \qquad (14)$$

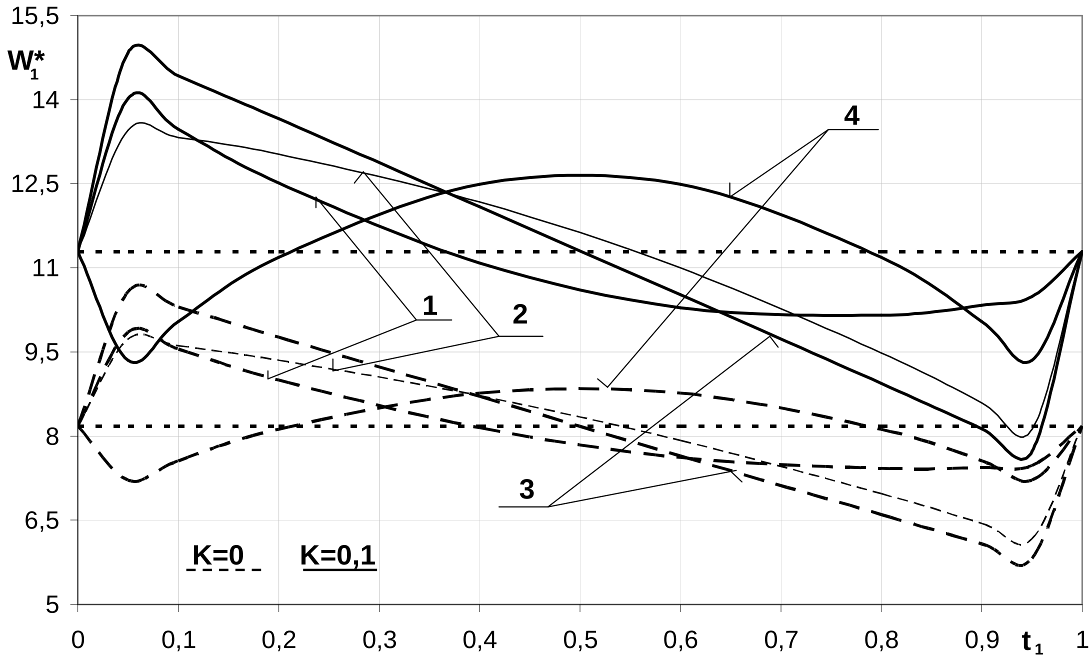

Fig.4. The dimensionless capacity forces W_1^* of slide journal bearing in the time t_1 by velocity perturbations:1) V_{10}=0,05; V_{1h}=0;2) V_{10}=0,05; V_{1h}=0,025;3) V_{10}=0,05; V_{1h}=0,05;4) V_{10}=0,05; V_{1h}=-0,05

Pressure in the bearing during the perturbation is a total of stationary flow pressure and perturbation pressure according to (1). According to mentioned equation (2) if we provide stationary flow pressure p_1 we will obtain capacity force W_1. On the other hand if we provide summary pressure p_1^* we will receive capacity force W_1^* as a result of this distribution.

Figure 4 presents hydrodynamic capacity W_1^* in the time function t_1 for perturbation velocities cases marked with graphs 1,2,3,4. Figure 4 also presents capacity calculation results for oil with constant viscosity (K=0). Capacity force in stationary flow is marked by horizontal lines with dots. Hydrodynamic capacity force W_1^* changes periodically with a period equal to perturbation velocity. In case of velocity perturbation in the bearing pin, increase of capacity force above the stationary condition value last no longer than half of the perturbation period and the increase of capacity force is bigger than the decrease in the remaining time. When perturbation of velocity on the bearing pin is in the same direction as a peripheral velocity of the pin it causes then bigger increase of capacity force than decrease. It is opposite in case of oil peripheral velocity perturbation on the shell surface, but his diagrams are not presented in his article. The case 2 effects are shown on the diagram 4. Capacity force course in time is not symmetrical for different perturbation of velocity quantities on the pin and on the shell. . Presented modification of dimensionless capacity force W_1^* illustrate also a change of Sommerfeld Number So in the bearing node model.

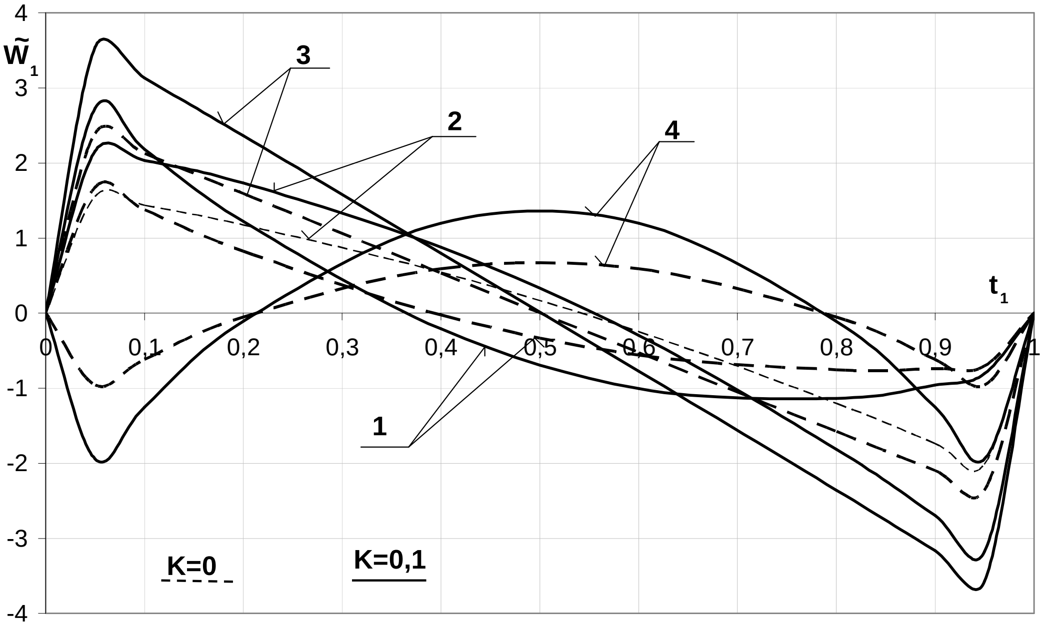

Fig.5. The dimensionless capacity forces $\tilde{W}_1$ of slide journal bearing in the time t_1 by velocity perturbations:1) V_{10}=0,05; V_{1h}=0; 2) V_{10}=0,05; V_{1h}=0,025 3) V_{10}=0,05;V_{1h}=0,05; 4) V_{10}=0,05; V_{1h}=-0,05

Capacity force W^* is situated in the co-ordinate angle φ_w from a angle φ=0 calculated for film origin (Fig. 3):

$$\varphi_w = \pi - \beta = \pi - arctg\left|\frac{W_{1y}^*}{W_{1x}^*}\right|, \tag{15}$$

Figure 6 presents contact angle of a bearing hydrodynamic capacity force in dimensionless time function t_1 in four considered perturbation of velocity cases marked same as before. By stationary flow angle is marked with dot line φ_w angle defines capacity force position changes periodically and the period is equal to perturbation of velocity period. In all considered cases of perturbation velocity the φ_w angle change in time is not bigger than four degrees. Only in case 4 (graph 4 Fig.6) for oil with constant viscosity the position angle is uniform (constant). Capacity force angle change for viscosity in dependence on pressure is insignificant comparing to change with constant viscosity and equals below one degree. Viscosity dependence on pressure causes angle φ_w increase for stationary flow.

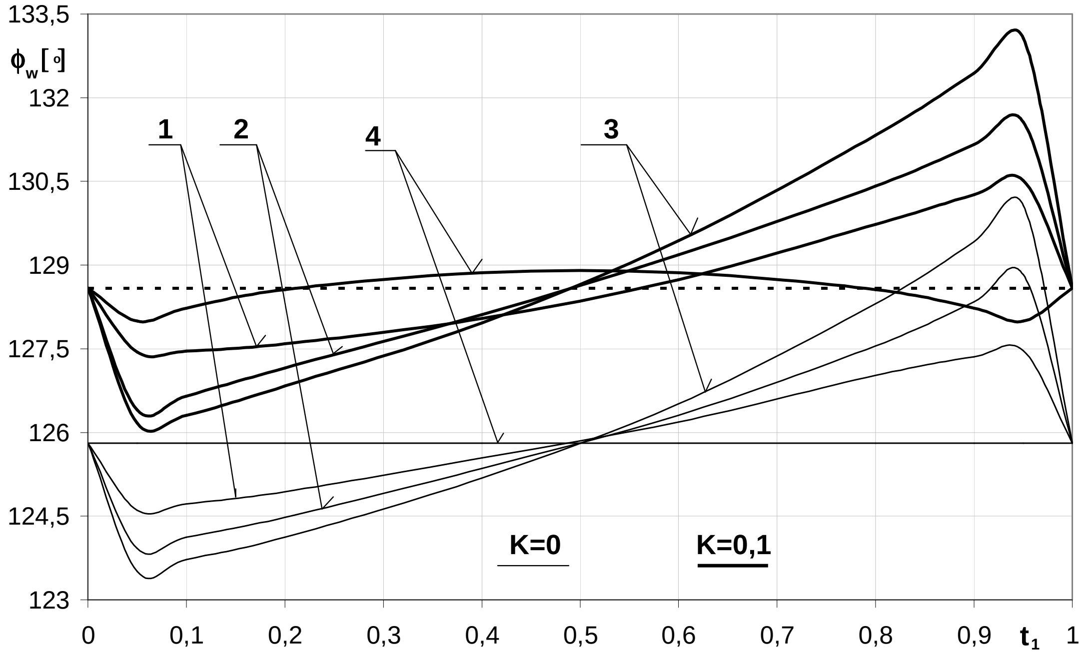

Fig. 6. Angle φ_w situated capacity force W_1^* in slide journal bearing in the time t_1 by velocity perturbations: 1) $V_{10}=0{,}05$; $V_{1h}=0$ 2) $V_{10}=0{,}05$; $V_{1h}=0{,}025$. 3) $V_{10}=0{,}05$; $V_{1h}=0{,}05$; 4) $V_{10}=0{,}05$; $V_{1h}=-0{,}05$

5. Conclusions

Unsteady perturbation of velocity on the journal and the sleeve have influence on the hydrodynamic pressure distribution and the hydrodynamic capacity forces in the lubricated gap. The influence is stronger when the oil viscosity depends stronger on pressure. Summary pressure, perturbation pressure change and capacity forces are periodical equal to perturbation of velocity period and the size of change depends on perturbation of velocity. When the perturbation of velocity on the journal has the same direction as peripheral velocity of the pin then the perturbation pressure is positive. When the peripheral velocity is in the opposite direction then the perturbation pressure is negative and it decrease summary pressure. Pressure increase and decrease is not symmetrical during the perturbation time. Although presented case consider isothermal bearing model with infinity width the results can be useful in preliminary pressure distribution estimation by laminar unsteady lubrication of cylindrical journal bearings with infinity width. Presented results will going to be used as a comparison quantities in case of laminar unsteady flow of Non-Newtonian fluids in cylindrical bearing gap.

Notation

b - length of the journal
ε - eccentricity
h - gap height $h = \varepsilon h_1$
h_1 - dimensionless gap height $h_1 = 1+ \lambda \cos\varphi$

L_1 - dimensionless bearing length $L_1 = \frac{b}{R}$

p - hydrodynamic pressure $p = p_0\, p_1$

p_o - characteristic value of pressure

p_1 - dimensionless hydrodynamic steady pressure

p_1^* - dimensionless hydrodynamic summary pressure

$\tilde{p}_1$ - dimensionless hydrodynamic unsteady pressure

r - radial co-ordinate $r = R(1+\psi r_1)$

r_1 - dimensionless radial co-ordinate

R - radius of the journal

Re - Reynolds Number $\mathrm{Re} = \frac{U\rho_0\varepsilon}{\eta_0 R}$

Re* - modified Reynolds Number $\mathrm{Re}^* = \mathrm{Re}\psi$

S_o - Sommerfeld Number for slide journal bearing

t - time $t = t_0\, t_1$

t_o - characteristic value of time

t_1 - dimensionless time

Ty - Taylor Number $\mathrm{Ty} = \mathrm{Re}\sqrt{\psi}$

U - peripheral journal velocity $U = \omega R$

V_{10} - dimensionless velocity of perturbation in direction φ on the journal

V_{1h} - dimensionless velocity of perturbation in direction φ on the sleeve

V_{30} - dimensionless velocity of perturbation in direction z on the journal

V_{3h} - dimensionless velocity of perturbation in direction z on the sleeve

W_0 - characteristic value of capacity force $W = 2Rbp_0$

W - capacity force $W = W_0\, W_1$

W_1 - dimensionless capacity force

W_x, W_y - components of capacity force W

W_{1x}, W_{1y} - dimensionless components of capacity force W_1

z - co-ordinate in length of the journal $z = b\, z_1$

z_1 - dimensionless co-ordinate in length of the journal

ε - radial clearance

η - dynamic oil viscosity $\eta = \eta_0\eta_1$

η_o - characteristic value of dynamic oil viscosity

η_1 - dimensionless dynamic oil viscosity

λ - dimensionless eccentricity ratio

ρ - oil density $\rho = \rho_0\rho_1$

ρ_o - characteristic oil density

ρ_1 - dimensionless oil density

φ_0 - the angular amplitude torsional forced vibrations of the shaft

φ_e - the angular co-ordinate for the film end

φ_w - the angular co-ordinate situated capacity force

ψ - dimensionless radial clearance $\psi = \frac{\varepsilon}{R}$; $10^{-4} \le \psi \le 10^{-3}$

ω - angular journal velocity

ω_o - angular velocity of perturbations

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