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Dynamics of a Journal Bearing During Sudden Rotor Stoppage Considering Inertial Properties and Pressure-Dependent Lubricant Viscosity

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Abstract

The dynamic behavior of a journal bearing during sudden rotor stoppage is investigated using a mathematical model that accounts for the inertial properties of the rotor and the pressure dependence of lubricant viscosity. The lubricant viscosity is described by the Barus law, and the pressure distribution in the lubricant film is determined from the Reynolds equation for one-dimensional flow with variable viscosity. The resultant hydrodynamic force is incorporated into the second-order equation of motion of the rotor–bearing system. A sinusoidal half-wave is used to represent the external impact load, and the resulting nonlinear differential equation is solved numerically. The transient variation of lubricant-film thickness and maximum lubricant-film pressure is analyzed during the impact and after removal of the load. For the investigated conditions, the maximum pressure in the lubricant film reaches approximately 74 MPa and subsequently decreases to low-amplitude oscillations around a mean value of about 14 MPa. The lubricant-film thickness remains sufficient to preserve film integrity throughout the considered transient process. Comparison with the approximate approach shows that the latter overestimates the displacement, velocity, and pressure responses. The proposed model can be used to assess the dynamic behavior and impact resistance of journal bearings under sudden rotor stoppage and impact loading

Keywords: journal bearing, sudden rotor stoppage, impact loading, rotor inertia, lubricant viscosity, pressure dependence, lubricant film, dynamic analysis, impact resistance.

Introduction

Rapid stoppage of rotating machinery can generate transient impact loads in journal bearings and cause a substantial reduction in the hydrodynamic load-carrying capacity of the lubricant film. The resulting transient motion of the rotor relative to the bearing may lead to a significant decrease in lubricant-film thickness and a corresponding increase in pressure within the lubricating layer. Therefore, the dynamic behavior of journal bearings during sudden rotor stoppage is an important factor in assessing their reliability and resistance to impact loading. The analysis of transient processes in journal bearings traditionally relies on hydrodynamic lubrication theory based on the Reynolds equation. In many engineering calculations, the lubricant viscosity is assumed to be constant and the inertia of the rotor is neglected. Such assumptions simplify the governing equations but may limit the accuracy of the predicted transient response, particularly under relatively high impact loads.

An important factor affecting the dynamic characteristics of the lubricant film is the pressure dependence of lubricant viscosity. This effect can be described using the Barus relation, according to which lubricant viscosity increases with pressure. Accounting for this dependence makes it possible to obtain a more realistic description of the pressure distribution and load-carrying capacity of the lubricant film under transient conditions. The influence of lubricant viscosity on the impact resistance of journal bearings has been considered in previous studies, including approaches based on approximate solutions.

At the same time, the inertia of the rotor may play a significant role in the transient response of the rotor–bearing system. When rotor acceleration is neglected, the governing equation is reduced to a first-order nonlinear differential equation, whose numerical implementation may be associated with computational instability. Including the inertial force of the rotor leads to a second-order equation of motion that provides a direct description of the dynamic process and is convenient for numerical solution.

Literature review

The performance of journal bearings is strongly influenced by lubricant properties, which affect the load-carrying capacity, minimum lubricant-film thickness, maximum pressure, and dynamic characteristics of the bearing [1]. These relationships provide a theoretical basis for determining the initial operating conditions of a journal bearing and remain widely used in engineering analysis.

The dynamic response of journal bearings under rapidly changing loads has subsequently attracted considerable attention. Transient behavior under dynamic and suddenly applied loads has been investigated, with particular attention to journal misalignment, lubricant-film thickness, maximum film pressure, and journal motion [2]. The dependence of lubricant viscosity on pressure has also been considered, and an approximate solution for the transient problem has been obtained [3]. These studies established the basis for assessing the influence of lubricant properties on the impact resistance of journal bearings.

More recent studies have significantly expanded the range of physical effects considered in transient journal-bearing models. An analytical solution for the dynamic forces generated by lubricated long journal bearings under dynamic loading has been developed. By analytically integrating the Reynolds equation using Sommerfeld variables, expressions for dynamic lubricant forces have been obtained and squeeze-film and hydrodynamic contributions have been separated, reducing the dependence on classical approximate solutions [4].

Particular attention in recent research has been given to the coupled dynamics of the rotor and bearing under nonstationary loading. A dynamic misalignment model coupled with rotor dynamics and transient mixed elastohydrodynamic lubrication has been proposed. The results demonstrated that dynamic misalignment can substantially influence the transient lubrication characteristics, particularly under nonstationary operating conditions. The model incorporated journal-center translation, surface roughness, cavitation, and elastic deformation of the bearing [5]. The influence of bearing deformation under impact loading has also been investigated. A transient mixed-lubrication model accounting for the coupled effects of journal misalignment and bearing viscoelastic deformation was developed in [6]. The results showed that the effects of deformation become particularly important at high rotational speeds, heavy loads, and rapidly changing impact loads, where neglecting the compliance of the bearing structure may lead to less realistic predictions of lubrication performance. For marine applications, the nonlinear tribo-dynamic behavior and transient stability of offset-halves journal bearings under sudden loading have been investigated [7]. The thermo-elasto-hydrodynamic model considered load amplitude, bearing geometry, and lubricant properties and demonstrated that these parameters significantly affect the transient response and stability of the bearing–rotor system. Recent research has also demonstrated the importance of considering thermal effects and time-varying operating conditions. A transient thermal elastohydrodynamic model of a journal-thrust coupled bearing subjected to time-varying loads was developed [8]. The study showed that the coupled thermal-pressure effects can significantly influence transient lubrication characteristics and therefore should not be neglected when analyzing bearings under realistic dynamic operating conditions. In addition, transient lubrication models are increasingly being coupled with wear and surface-contact phenomena. This development is associated with the fact that the operating conditions of journal bearings during start-up and other nonstationary regimes are accompanied by significant changes in the lubricant-film thickness, local contact conditions, and hydrodynamic pressure distribution. Under such conditions, the interaction between the bearing surfaces may become more pronounced, and the simultaneous consideration of lubrication, mechanical contact, and wear becomes important for a reliable assessment of bearing performance. A tribo-dynamic model of a hydrodynamic bearing–rotor system under start-up conditions has been developed, accounting for misalignment, wear, rotor imbalance, surface contact, and elastic deformation [9]. The results demonstrate that these factors are mutually coupled and may considerably affect the transient response of the bearing–rotor system. In particular, the interaction between transient lubrication and rotor dynamics becomes important during nonstationary operating regimes, when the journal position, lubricant-film characteristics, and mechanical loading change continuously with time. These approaches indicate that the analysis of journal bearings under transient conditions should consider not only the hydrodynamic pressure field but also the changes in contact and surface conditions associated with wear and deformation. The influence of piezoviscous effects on journal-bearing characteristics has also been investigated for different bearing configurations. The pressure dependence of lubricant viscosity becomes especially important in regimes characterized by high local pressures, because an increase in pressure leads to a corresponding increase in lubricant viscosity and therefore changes the hydrodynamic resistance and pressure distribution within the lubricant film. This effect may alter the load-carrying capacity of the bearing and influence the dynamic characteristics of the lubricated contact. Modern lubrication models incorporating piezoviscous behavior have therefore been used to obtain a more realistic description of pressure generation in the lubricant layer under elevated-pressure conditions [10]. The results indicate that pressure-dependent viscosity affects lubricant-film pressure, stiffness, damping, and friction characteristics. Consequently, the assumption of constant lubricant viscosity may lead to deviations in the predicted dynamic response of a journal bearing, particularly when the bearing operates under high transient loads. This confirms the relevance of incorporating pressure-dependent viscosity into mathematical models intended for the analysis of short-duration dynamic and impact processes.

Recent tribo-dynamic models have also incorporated additional coupled effects, including journal axial motion and misalignment, to describe complex transient behavior under dynamic loading [11]. Such approaches extend conventional two-dimensional descriptions of journal-bearing operation by considering additional degrees of freedom of the journal and the possibility of simultaneous translational and axial motion. Misalignment may modify the local lubricant-film geometry and produce a nonuniform pressure distribution along the bearing width, while axial motion may further alter the temporal evolution of the lubricant-film state. Accounting for these effects allows a more comprehensive representation of the actual transient behavior of bearing systems subjected to dynamic loading. The results of such studies demonstrate that the response of a lubricated bearing may depend on the coupling between the journal motion, bearing geometry, and lubrication conditions, which should be taken into account when developing more general rotor–bearing dynamic models.

Wear-related effects are also important for assessing the long-term performance of sliding bearings. In addition to the instantaneous hydrodynamic response, the reliability of a sliding bearing is determined by the ability of the friction surfaces to maintain their functional characteristics during repeated loading and sliding. Wear changes the geometry of the contacting surfaces and may consequently alter the lubricant-film thickness, contact pressure, and operating conditions of the tribosystem. A calculation-experimental approach based on contact pressure and sliding velocity has been proposed for predicting wear and evaluating the wear resistance of cylindrical sliding bearings [12]. Such approaches make it possible to relate the kinematic and loading parameters of the tribosystem to the evolution of wear and provide additional information for assessing bearing durability. Although wear is not directly considered in the present transient model, the results of these studies indicate that the evolution of the friction surfaces is an important factor in the long-term assessment of sliding-bearing performance and should be considered in further development of dynamic models.

The wear resistance of friction elements is also affected by the conditions of surface interaction and lubrication. The state of the surface layer, the characteristics of the contact interaction, and the availability of lubricant may determine the intensity of material removal and the stability of the friction process. Experimental investigations of surface plastic deformation have demonstrated that the technological conditions of surface treatment can significantly influence the wear resistance of friction tools and contacting surfaces [13]. These results are relevant to the analysis of sliding bearings because the functional characteristics of the mating surfaces are directly related to the ability of the lubricant film to separate the surfaces during operation. Therefore, consideration of the condition and wear resistance of the friction surfaces complements the hydrodynamic analysis and provides a more comprehensive basis for evaluating the reliability of tribological systems.

Transient tribo-dynamic analysis has also been coupled with wear modeling to evaluate the evolution of lubricant-film and wear characteristics during nonstationary operation. Such models make it possible to analyze the mutual influence of the changing lubricant-film state and the gradual modification of the friction surfaces during transient processes. The influence of acceleration mode, acceleration time, external load, lubricant viscosity, and operating time on the transient response has been analyzed [14]. The results demonstrate that the transient behavior of a bearing is determined not only by the instantaneous value of the external load but also by the manner in which the loading is introduced and by the duration of the transient regime. These findings are particularly relevant to start-up and other rapidly changing operating conditions, where the lubricant film does not immediately reach a steady state and the bearing response is governed by coupled dynamic and tribological processes. Recent studies have further considered viscosity variation in rough and non-conventional bearing configurations, demonstrating the influence of viscosity variation on pressure distribution and load-carrying capacity [15]. Consideration of variable lubricant viscosity allows the influence of pressure and other physical effects on the formation of the lubricant film to be described more accurately than in models based on the constant-viscosity assumption. At the same time, bearing surface roughness and deviations from conventional geometry may alter the local film profile and pressure field, thereby affecting the overall load-carrying behavior of the lubricated contact. These results further confirm that the rheological properties of the lubricant and the geometric characteristics of the bearing should be considered jointly when analyzing bearing operation under nonstationary and elevated-pressure conditions.

Despite these advances, the specific problem of sudden rotor stoppage in a journal bearing with simultaneous consideration of rotor inertia and pressure-dependent lubricant viscosity remains insufficiently investigated. Most recent models focus on misalignment, thermal effects, viscoelastic deformation, wear, or complex bearing geometries, whereas the combined influence of rotor inertial loading and piezoviscous effects during the short transient following sudden stoppage requires further study. In particular, the interaction between the rapidly changing motion of the journal and the pressure-dependent rheological response of the lubricant may substantially affect the evolution of the lubricant-film thickness and the maximum pressure during the impact event. This provides the motivation for the present work, which develops and numerically analyzes a dynamic model of a journal bearing during sudden rotor stoppage with pressure-dependent lubricant viscosity and rotor inertia.

Purpose

The purpose of this study is to develop and numerically investigate a mathematical model of the transient dynamics of a journal bearing during sudden rotor stoppage, taking into account the rotor inertial properties and the pressure dependence of lubricant viscosity. Particular attention is given to determining the transient variation of lubricant-film thickness and pressure during impact loading and after its removal, as well as to assessing the influence of the considered physical factors on the impact resistance and serviceability of the journal bearing.

Research methodology

The study is based on the hydrodynamic theory of lubrication and numerical analysis of the transient motion of a rotor–journal bearing system subjected to sudden rotor stoppage. An infinitely long journal bearing with a rigid bearing liner is considered. The lubricant viscosity is assumed to depend on pressure according to the Barus law:

$$\mu = \mu_0 e^{\alpha p},$$

where μ_0 – is the lubricant viscosity at atmospheric pressure, α – is the piezoviscosity coefficient, and p – is the local pressure in the lubricant film.

The pressure distribution in the lubricant film is determined from the one-dimensional Reynolds equation for a lubricant with variable viscosity. The equation is transformed from the linear coordinate to the angular coordinate of the journal, while the time-dependent lubricant-film thickness and the velocity of approach between the journal and bearing liner are taken into account. The resulting equation is integrated subject to the corresponding boundary conditions, yielding the pressure distribution in the lubricant film.

The resultant hydrodynamic force generated by the lubricant film is obtained by integrating the pressure distribution over the bearing surface. To describe the transient motion of the rotor, the inertial force of the rotor is included in the equation of motion. This results in a second-order differential equation governing the relative motion of the rotor and bearing liner. The rotor mass is expressed in terms of the initial mean pressure in the bearing and the gravitational acceleration.

The impact load is represented by a sinusoidal half-wave characterized by an overload factor k and an impact duration Δt. A dimensionless time variable is introduced to facilitate the numerical solution of the governing equation. The initial relative eccentricity is determined using relationships between the eccentricity ratio and the Sommerfeld number derived from classical hydrodynamic lubrication theory. Since sudden rotor stoppage produces a soft impact, the initial rate of change of relative eccentricity is taken to be zero.

The resulting nonlinear differential equation is solved numerically over a time interval covering both the impact stage and the subsequent post-impact response. The principal quantities evaluated are the time-dependent lubricant-film thickness and the maximum pressure in the lubricant film. The calculated results are then used to assess the transient behavior and impact resistance of the journal bearing.

Research results

The calculation scheme is shown in Fig. 1. Assume that the lubricant viscosity follows the Barus law.

$$\mu = \mu_0 \exp(\alpha_0 p), \tag{1}$$

where μ_0 – is the viscosity at atmospheric pressure; α_0 – is the piezoviscosity coefficient; p – is the pressure at an arbitrary point of the lubricant film.

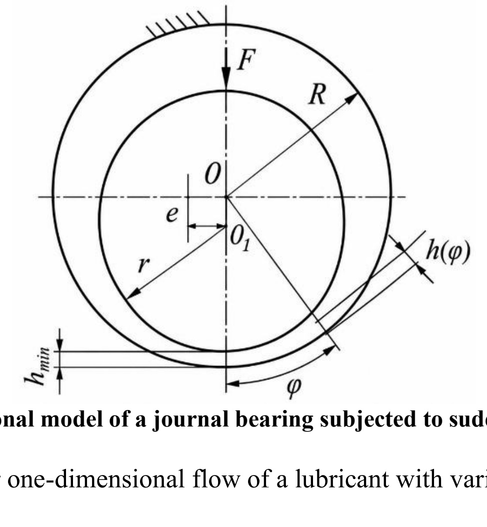

Fig. 1. Computational model of a journal bearing subjected to sudden rotor stoppage

The Reynolds equation for one-dimensional flow of a lubricant with variable viscosity has the form:

$$\frac{\partial}{\partial x} \cdot \left(\frac{h^3}{\mu} \cdot \frac{\partial \rho}{\partial x}\right) = 12 W_n,$$

where $h = h(\varphi)$ – is the lubricant-film thickness; $\frac{\partial \rho}{\partial x}$ – is the pressure gradient; $W_n = V \cos\varphi$ – is the journal velocity component directed normal to the surface at the given point; V – the velocity of approach of the journal to the bearing liner; x – is the linear coordinate.

Passing to the angular coordinate φ and substituting the expressions for the lubricant-film thickness and the approach velocity of the surfaces:

$$h = r\psi(1 - \chi\cos\varphi), \tag{2}$$
$$V = \frac{dh(0)}{dt} = -r\psi\frac{d\chi}{dt}, \tag{3}$$

where $\psi = (R - r)/r$ – is the dimensionless radial clearance; $\chi = e/(R - r)$ – is the dimensionless eccentricity; t – is time.

Integrating the resulting equation twice with respect to φ, taking into account the boundary conditions $\partial\rho/\partial\varphi = 0$ at $\varphi = 0$ and $p = 0$ at $\varphi = \pm\pi/2$. We obtain the pressure distribution function:

$$p(\varphi; t) = -\frac{1}{\alpha_0}ln\left\{1 - \frac{6\mu_0\alpha_0}{\psi^2\chi}\left[\frac{1}{(1-\chi\cos\varphi)^2} - 1\right]\frac{d\chi}{dt}\right\}.$$

Introduce a change of variable by defining the dimensionless time τ according to:

$$t = \frac{6\mu_0\alpha_0}{\psi^2}\tau, \tag{4}$$

then the pressure distribution function takes the following form:

$$p(\varphi, \tau) = -\frac{1}{\alpha_0}ln\left\{1 - \frac{\chi'}{\chi}\left[\frac{1}{(1-\chi\cos\varphi)^2} - 1\right]\right\}, \tag{5}$$

where $\chi' = d\chi/d\tau$.

The projections of the pressure forces onto the vertical axis produce the resultant force of the lubricant film, equal to

$$F_1(\tau) = 2rl\int_0^{\pi/2} p(\varphi, t)\cos\varphi\, d\varphi = -\frac{2rl}{\alpha_0}\int_0^{\chi/2} ln\left\{1 - \frac{\chi'}{\chi}\left[\frac{1}{(1-\chi\cos\varphi)^2} - 1\right]\right\}\cos\varphi\, d\varphi,$$

where l – is the bearing liner length.

The integral appearing on the right-hand side of the last expression can be found in analytical form. Thus, we obtain

$$F_1(\tau) = \frac{2rl}{\alpha_0}\mathcal{T}(\chi, \chi'), \tag{6}$$

where

$$\mathcal{T}(\chi, \chi') = \frac{2\sqrt{1-\chi^2}}{\chi}\left(\frac{\pi}{2} + arcsin\chi\right) - \frac{\sqrt{1-b_2^2}}{b^2}\left(\frac{\pi}{2} + arcsin\, b_2\right) - \frac{\sqrt{1-b_3^2}}{b_3}\left(\frac{\pi}{2} + arcsin\, b_3\right), \tag{7}$$
$$b_2 = \chi + \chi' - \sqrt{\chi'(\chi + \chi')}, \qquad b_3 = \chi + \chi' + \sqrt{\chi'(\chi + \chi')}.$$

If, as is customary in practical calculations, acceleration and, consequently, the inertial load of the rotor are neglected, the resultant force $F_1(\tau)$ of the lubricant film should be equated to the external dynamic load $F(\tau)$. As a result, a first-order differential equation is obtained that is not solved explicitly for the derivative χ'. In most practically important cases, it can be solved with respect to the independent variable τ and therefore admits a parametric representation of the solution:

$$\begin{cases} \tau = f_1(\chi); \\ \chi' = f_2(\chi). \end{cases}$$

It should be noted, however, that the numerical implementation of this solution method is complicated by the instability of the computational process, requiring, at a minimum, calculations with double precision. On the other hand, the equation accounting for the inertial force of the rotor, although it is of second order, can be solved on a computer without any difficulty. Taking the above into account, write the equation in the following form:

$$-m\frac{d^2h(0)}{dt^2} + \frac{2rl}{\alpha_0}\mathcal{T}(\chi, \chi') = F(t), \tag{8}$$

where m – is the rotor mass; $F(t)$ – is the prescribed external excitation.

Passing to dimensionless time according to relation (4) and taking equation (2) into account, we obtainτ

$$h(0) = r\psi(1 - \chi),$$
$$\frac{d^2h(0)}{dt^2} = -r\psi\frac{d^2x}{dt^2} = -\frac{r\psi^5}{36\mu_0^2\alpha_0^2}\chi", \tag{9}$$

where $\chi" = d^2\chi/d\tau^2$.

Let the rotor mass be represented as

$$m = G/g = p_0\, 2r\, l/g, \qquad (10)$$

where p_0 – is the initial, at $\tau = 0$, mean pressure on the bearing liner; g = 9.81 ms^{-2} – is the gravitational acceleration.

Substituting expressions (9) and (10) into equation (8), we obtain

$$\frac{\rho_0 r \psi^5}{36\mu_0^2 \alpha_0 g} \cdot \frac{2rl}{\alpha_0} \chi'' + \frac{2rl}{\alpha_0} \mathcal{T}(\chi, \chi') = F(\tau),$$

or

$$\frac{\rho_0 r \psi^5}{36\mu_0^2 \alpha_0 g} \chi'' + \mathcal{T}(\chi, \chi') = f(\tau), \qquad (11)$$

where $f(\tau) = \alpha_0 F(\tau)/(2rl)$.

If, for example, the external impact $F(t)$ is specified in the form of a sinusoidal half-wave

$$F(t) = \begin{cases} G\left(1 + k \sin\dfrac{\pi}{\Delta t} t\right) & \text{при } 0 \le t \le \Delta t, \\ 0 & \text{при } t > \Delta t, \end{cases}$$

where k – is the overload factor, Δt – is the duration of the impact load; then the function $f(\tau)$ takes the corresponding form

$$f(\tau) = \begin{cases} p_0 \alpha_0 \left(1 + k \sin\dfrac{\tau}{T}\right) & \text{при } 0 \le \tau \le T, \\ \rho_0 \alpha_0 & \text{при } \tau > T, \end{cases}$$

where

$$T = \frac{\psi^2 \Delta t}{6\pi \mu_0 \alpha_0}.$$

Then the equation of motion (11) can be written in the following form:

$$\chi'' + \frac{36\mu_0^2 \alpha_0 g}{\rho_0 r \psi^5} \mathcal{T}(\chi, \chi') = \frac{36\mu_0^2 \alpha_0^2 g}{r \psi^5} \left(1 + k \sin\frac{\tau}{T}\right), \qquad (12)$$

where as at $\tau > T$ one should set $k = 0$.

It remains to determine the initial conditions for integration of equation (12). The initial value of the relative eccentricity can be obtained from the classical hydrodynamic theory of lubrication of a rotating rotor. In practice, the values χ_0 are determined from tables, which is inconvenient when solving problems on a computer.

For practical calculations, approximate relationships can be used, since, within the considered range of relative eccentricities $0.3 \le \chi_0 \le 0.7$, a simple relationship exists between the values of χ_0 and the Sommerfeld numbers

$$S_0 = \frac{\rho_0 \psi^2}{\mu \omega},$$

where ω – is the angular velocity of the rotor before its sudden stoppage. For the specified range, the value χ_0 can be determined with sufficient accuracy from the formula

$$\chi_0 = a_0 + b_0 \ln S_0,$$

where the constants a_0 and b_0 depend only on the ratio of the bearing liner length to its diameter.

Let us determine the initial value of the derivative χ_0'. The sudden stoppage of the rotor causes the load-carrying capacity of the lubricant film associated with circumferential velocity to instantly decrease from a certain steady-state value to zero. A corresponding finite discontinuity is also experienced by the acceleration of the rotor center of gravity. Thus, a "soft" impact occurs, in which the velocity does not change discontinuously. Therefore, the initial value χ'_0 should be taken as zero.

The results of the numerical solution of the equation for the following parameter values are given below: $r = 0.075\, m, l/2r = 1, p = 6MP\text{a}, \mu_0 = 0.03Pa \cdot s, \alpha_0 = 0.14 \cdot 10^{-7} \text{Pa}^{-1}, \psi = 1 \cdot 10^{-3}, \omega = 314 s^{-1}, a_0 = 0.5329, b_0 = 0.25$.

The duration of the impact load was $\Delta t = 0.02$ s, the overload factor was $k = 5$, the initial value of the relative eccentricity $\chi_0 = 0.399$.

The calculation was performed for a time interval equal to $2\Delta t$ in order to investigate the behavior of the shaft–bearing dynamic system during the impact and after removal of the load. Simultaneously with determining the values of χ та χ' the maximum pressure in the lubricant film corresponding to these values was calculated $\varphi = 0$:

$$p_{max} = -\frac{1}{\alpha_0} \ln\left\{1 - \frac{\chi'}{\chi}\left[\frac{1}{(1-\chi)^2} - 1\right]\right\}.$$

The graphs $\chi(t), \chi'(t)$, i $p_{max}(t)$ are shown in Figs. 2 and 3. As can be seen from the figures, the lubricant-film thickness initially decreases rather rapidly, but then the rate of its change decreases sharply and, after the impact ceases, is $\chi' \approx 0.005$, which corresponds to a linear approach velocity of the shaft and bearing liner of less than 0.15 mm/s. At large values of t the lubricant-film thickness remains sufficient $(h_{min} > 10\ \mu m)$ to maintain its integrity, significantly exceeding the height of the surface roughness of the friction surfaces. The maximum pressure in the lubricant film initially increases to 74 MPa, then decreases sharply and, after removal of the impact, oscillates with a small amplitude around the mean value of 14 MPa.

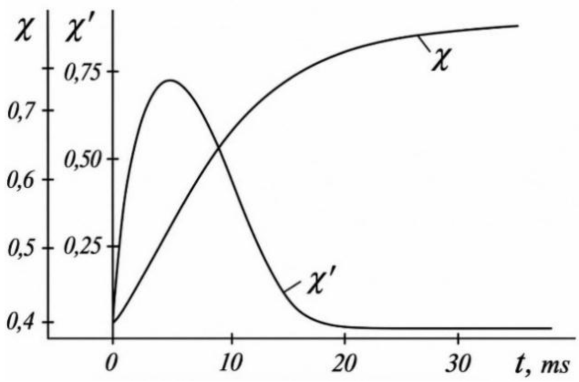

Fig. 2. Dependence of the lubricant film thickness on time under impact loading

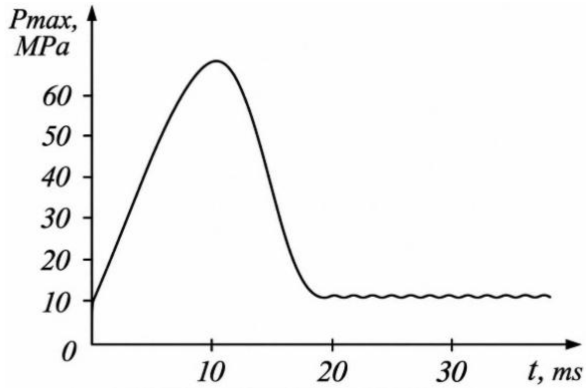

Fig. 3. Dependence of the maximum pressure in the lubricant film on time under impact loading

Comparison with calculations based on an approximate approach shows differences in the predicted displacements, velocities, and pressures, with the approximate approach providing higher values. Thus, the approximate approach can be used for assessing the serviceability of journal bearings under impact loading.

Finally, it should be noted that calculating the impact resistance of bearings at high overload factors requires taking the elasticity of the bearing liner into account. Such studies will be carried out in the future; however, it can already be noted that the compliance of the foundation leads to an increase in lubricant-film thickness and improves the impact resistance of bearings.

Conclusions

A mathematical model of the transient dynamics of a journal bearing during sudden rotor stoppage has been developed, accounting for rotor inertia and the pressure dependence of lubricant viscosity described by the Barus law.

The numerical results demonstrate a pronounced transient response of the rotor–bearing system after rotor stoppage. The lubricant-film thickness initially decreases rapidly, while the rate of its reduction subsequently decreases, allowing the lubricant film to remain continuous during the considered impact event.

Under the investigated impact conditions, the maximum lubricant-film pressure reaches approximately 74 MPa and subsequently decreases to low-amplitude oscillations around a mean value of about 14 MPa. These results indicate that the bearing retains its load-carrying capacity after the impact load is removed.

Comparison with the approximate approach shows that the approximate method overestimates the displacement, velocity, and pressure responses. Nevertheless, it can be used for preliminary engineering assessment of the serviceability and impact resistance of journal bearings. For higher overload factors, the model should additionally account for the elastic compliance of the bearing liner and its support.

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Марченко Д.Д., Матвєєва К.С. Динаміка підшипника ковзання при раптовій зупинці ротора з урахуванням інерційних властивостей та тискозалежної в’язкості мастила

Досліджено динамічну поведінку підшипника ковзання при раптовій зупинці ротора з урахуванням інерційних властивостей ротора та залежності в’язкості мастила від тиску. В’язкість мастила описано законом Баруса, а розподіл тиску в мастильному шарі визначено на основі рівняння Рейнольдса для одновимірного руху мастила зі змінною в’язкістю. Результуючу гідродинамічну силу враховано в рівнянні руху ротора–підшипника другого порядку. Зовнішню ударну дію задано у вигляді півхвилі синусоїди, а отримане нелінійне диференціальне рівняння розв’язано чисельно. Проаналізовано зміну товщини мастильного шару та максимального тиску в ньому під час ударної дії та після її припинення. За досліджуваних умов максимальний тиск у мастильному шарі досягає приблизно 74 МПа, після чого зменшується та здійснює коливання з малою амплітудою навколо середнього значення близько 14 МПа. Товщина мастильного шару протягом досліджуваного перехідного процесу залишається достатньою для збереження його цілісності. Порівняння з наближеним методом показало, що останній дає завищені значення переміщень, швидкостей і тисків. Запропонована модель може бути використана для оцінювання динамічної поведінки та ударостійкості підшипників ковзання при раптовій зупинці ротора та ударному навантаженні.

Ключові слова: підшипник ковзання, раптова зупинка ротора, ударне навантаження, інерція ротора, в’язкість мастила, залежність в’язкості від тиску, мастильний шар, динамічний аналіз; ударостійкість