Lubricants **2015**, *3*, 27-53; doi:10.3390/lubricants3010027

OPEN ACCESS

lubricants

ISSN 2075-4442

www.mdpi.com/journal/lubricants

Review

On the Characteristics of Misaligned Journal Bearings

Joon Young Jang and Michael M. Khonsari *

Department of Mechanical Engineering, Louisiana State University,
Baton Rouge, LA 70803, USA; E-Mail: jyjang@me.lsu.edu

* Author to whom correspondence should be addressed; E-Mail: khonsari@me.lsu.edu;
Tel.: +1-225-578-9192; Fax: +1-225-578-5924.

Academic Editors: Romeo P. Glovnea and Michel Fillon

Received: 29 January 2015 / Accepted: 3 March 2015 / Published: 16 March 2015

Abstract: Journal bearing misalignment arise generally from the shaft deformation under load, deflection of the shaft, manufacturing and assembly errors, improper installation, and asymmetric loading. During operations, misalignment has a considerable effect on the static and dynamic performances. It could cause wear, vibration and even system failure. In this article, a literature review of misalignment of the journal bearings is presented. The basic theory for the misalignment and some results for the circular journal bearing are also presented to show the general trends of the misalignment.

Keywords: misalignment; journal bearing; thermohydrodynamic analysis; mass conservation; cavitation

1. Introduction

Relative sliding or squeeze motion between two surfaces separated by a thin film of fluid, results in the generation of hydrodynamic pressure and load-carrying capacity. This is referred to as the hydrodynamic lubrication. Among all different types of bearings, radially-loaded hydrodynamic journal bearings are the most common. They are used to support rotating shafts in machines such as compressors, turbo-generators, pumps, and internal combustion engines, *etc.* Ideally, the axes of both shaft and bush are parallel when installed and remain so during the operation under the imposed load and speed. Nevertheless, in practice, this ideal condition rarely exists, and the shaft tends to suffer from some degree of misalignment while rotating inside its bushing. The primary effect is considerable reduction in the minimum thickness of the lubricant film, which is responsible for protecting the surfaces from direct contact.

Substantially, reduced minimum film thickness also alters the entire pressure and temperature fields~~.~~ In fact, the maximum values of pressure and temperature in misaligned bearings are much higher than those in the aligned ones. To counteract this effect a slight defect in the minimum film thickness area is sometimes added to the bearing geometry to prevent contact. Misalignment can also result from an individual or combination of the following factors:

a. Asymmetric bearing loading;
b. Elastic deflection of the shaft, under an imposed under load or its own weight;
c. Thermal distortion of the shaft;
d. Distortions caused by bearing housing supports;
e. Manufacturing tolerances (due to inaccurate machining, casting and forging); and
f. Errors due to installation and assembly defects.

Misalignment is known to have a deleterious effect on the steady-state performance of a journal bearing. However, general characterization of the behavior of misaligned bearings is very complex. The objective of this article is to examine the state of the art on the subject of misalignment in journal bearings and to guide the interested readers to the appropriate publications.

1.1. Types of Misalignment and Coupling Effects

Misalignment occurs where the driving shaft and the driven machine are not concentric. There are two types of misalignment: parallel and angular misalignment. In parallel misalignment, the center lines of both shafts are parallel but they are offset. In angular misalignment, the shafts are at an angle to each other. Parallel misalignment can be further divided up in horizontal and vertical misalignment, depending on whether the shaft's misalignment position in the horizontal or vertical planes. Coupling misalignment can be one of—or commonly a combination of—parallel misalignment or angular misalignment. The shaft misalignment can also occur because of vibration due to the reaction forces generated in the shaft couplings.

1.2. Film Thickness

The most important effect of bearing misalignment is the drastic alteration of the protective film thickness in both the circumferential and axial directions. Therefore, the Reynolds equation should be modified to allow the variation of film thickness in both directions. This needs to be carefully taken into consideration in the analysis.

Figure 1 shows the schematic geometry of a misaligned journal bearing system. Referring to Figure 1, the film thickness in journal bearing with provision for misalignment can be expressed as:

$$h = C + e_o \cos\left(\theta - \varphi_o\right) + e'\left(\frac{z}{L} - \frac{1}{2}\right)\cos\left(\theta - \alpha - \varphi_o\right) \tag{1}$$

where C is the clearance, R is the journal radius and L is the bearing width, φ_o is the attitude angle between the vertical line (Y-axis) and the line of centers (Y'-axis), and e_o is the eccentricity at the bearing mid-plane. The parameter e' is the magnitude of the projection of the axis of the misaligned journal on the mid-plane of the bearing. The misalignment angle α is the angle between the line of centers and the rear center of the misaligned journal (see Figure 1). The misalignment eccentricity ratio is $\varepsilon' = e'/C = D_M \varepsilon'_{\max}$, where D_M is the degree of misalignment in value from 0 to 1. $\varepsilon'_{\max}$ is the maximum possible ε' [1]. The position of the journal specified by means of its eccentricity ratio and attitude angle at its mid-plane and by two important parameters, the degree of misalignment D_M and misalignment angle α, defining the extent and direction of the misalignment.

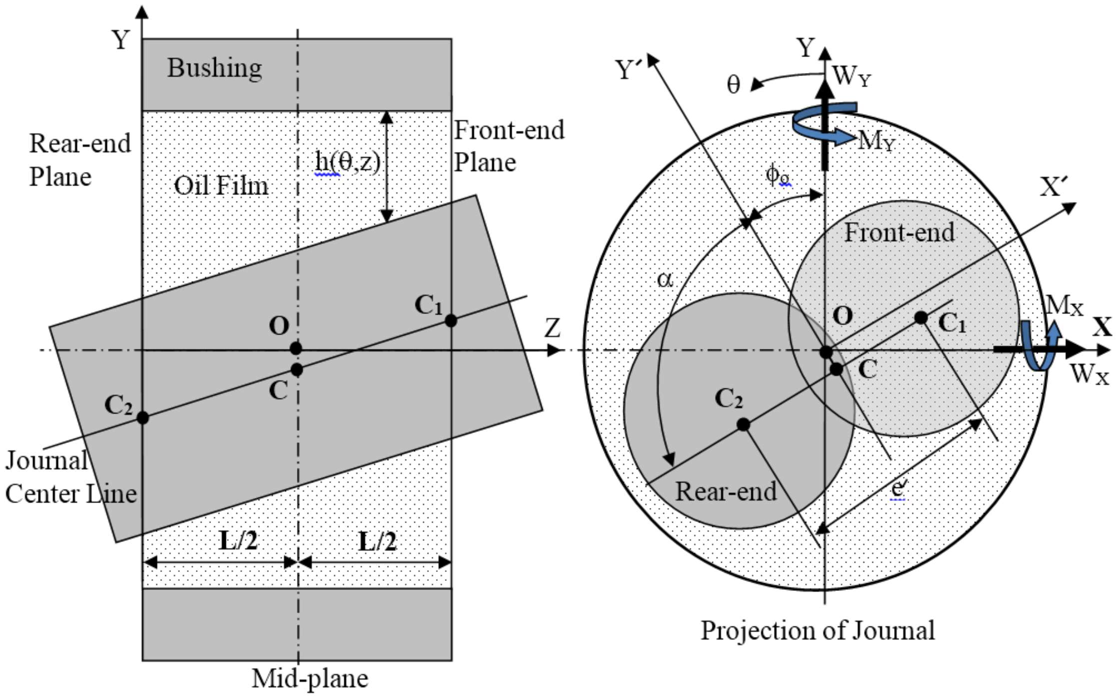

Figure 1. Nomenclature of misaligned journal bearing.

1.3. Pressure Field

The generalized Reynolds equation with provision for variation of viscosity across the film thickness is used for the prediction for the pressure profile. Based on the assumptions that the axial movement of the journal is neglected, the bushing is stationary, and the density remains constant across the film thickness the generalized Reynolds equation is:

$$\frac{\partial}{\partial x}\left[\rho F_2\left(\frac{\partial P}{\partial x}\right)\right]+\frac{\partial}{\partial z}\left[\rho F_2\left(\frac{\partial P}{\partial z}\right)\right]=\frac{\partial}{\partial x}\left[\rho u_s\left(h-\frac{F_1}{F_0}\right)\right]+\frac{\partial}{\partial t}(\rho h) \tag{2}$$

where

$$F_0=\int_0^h \frac{1}{\eta}dy \;\; F_1=\int_0^h \frac{y}{\eta}dy \text{ and } F_2=\int_0^h\left[\frac{1}{\eta}\left(y^2-\frac{F_1}{F_0}y\right)\right]dy \tag{3}$$

For the steady-state condition, the last term in Equation (2) disappears. The boundary conditions are

$$P = P_{\text{Supply}} \text{ at Groove} \tag{4}$$

$$P = P_{\text{ambient}} \text{ at both ends}$$

$$\frac{\partial P}{\partial \theta} = 0 \text{ and } P = P_{\text{cavitation}} \text{ at } \theta = \theta_{\text{cavitation}}$$

The third boundary condition represents the Swift-Stieber boundary condition which can be easily adopted in the numerical scheme. However, this boundary condition does not satisfy the continuity equation near the cavitation boundary. While this set of boundary conditions generally suffices for perfectly-aligned, steadily-loaded bearings, their use becomes questionable when one deals with misaligned bearings because of the complications associated with cavitation. This calls for implementation of a more realistic cavitation boundary condition known as Jakobsson-Floberg-Olsson (JFO) condition.

To implement the JFO boundary conditions, Elrod [2] introduced a new parameter called the fractional film content, $\Theta = \rho/\rho_c$, formulated the corresponding pressure $P = P_c + \beta \ln \Theta$, and derived a form of the Reynolds equation where the unknown is the fractional film content instead of the pressure. The result is a mass-conservative form of the Reynolds equation that takes on the following form:

$$\frac{\partial}{\partial x}\left[g\beta F_2 \left(\frac{\partial \Theta}{\partial x}\right)\right] + \frac{\partial}{\partial z}\left[g\beta F_2 \left(\frac{\partial \Theta}{\partial z}\right)\right] = \frac{\partial}{\partial x}\left[u_S \Theta \left(h - \frac{F_1}{F_0}\right)\right] + \frac{\partial}{\partial t}(\Theta h) \tag{5}$$

where ρ_c is the fluid density at the cavitation pressure P_c, β is the bulk modulus and u_s is the speed of the shaft. The cavitation switch function g(Θ) is zero within the cavitation, and unity elsewhere.

$$\begin{aligned} g &= 0 \text{ when } \Theta < 1 \\ g &= 1 \text{ when } \Theta \geq 1 \end{aligned} \tag{6}$$

Equation (5) satisfies the pressure conditions at the full film and cavitation region and also automatically implements the film rupture and reformation. The pressure-induced flow terms exist in the full region and disappear in the cavitation region. Therefore, in the full film region, the Reynolds equation is an elliptic partial differential equation and the shear flow term (source term) should be centrally differenced. In the cavitation region, the Reynolds equation is hyperbolic and the upwind differencing should be used for the shear flow term. Elrod [2] modified the finite difference expressions to allow the variation of the film thickness in the axial direction. The Elrod's mass-conservative cavitation algorithm has been widely used in many researches. However, this cavitation algorithm is susceptible to numerical instability in many cases when the switch function is varying its value between 0 and 1 near the cavitation boundary during the iterations. The so-called p-Θ formulation by Elrod and Adams [3] reduces the numerical instability since the switch function is not used in this formula.

Figure 2 shows the typical isothermal pressure distributions of the misaligned journal bearing considering the Elrod cavitation algorithm. Figure 2a represents the aligned journal bearing since $D_M = 0$ and, therefore, the maximum pressure is located at the mid-plane and the pressure is symmetric about the mid-plane. At $D_M = 0.6$, it is clearly shown that the maximum pressure is shifted to the rear end. The location of the maximum pressure is influenced by the orientation of the misalignment, *i.e.*, misalignment angle, α. The maximum pressure is greater than that for the aligned bearing due to the misalignment. Further, an increase in the degree of misalignment (D_M = 0.88 and 0.9) yields two peak values in the pressure axially near both ends.

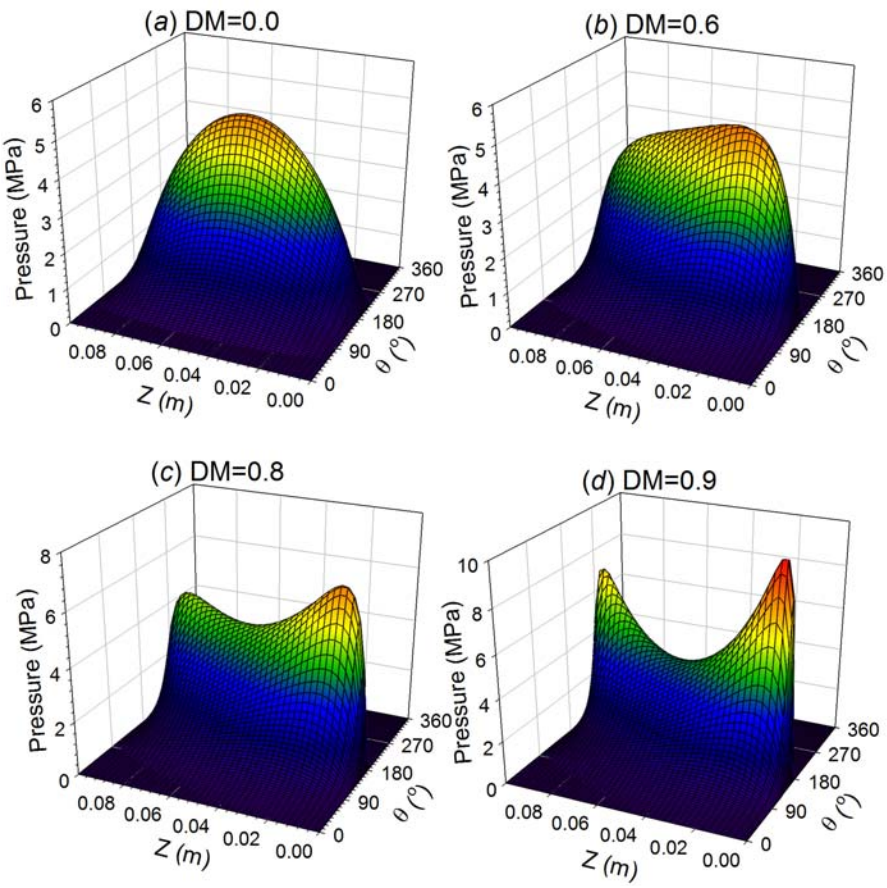

Figure 2. Isothermal Pressures of Misaligned Journal Bearing at N = 3000 rpm, ε = 0.6, Λ = 1 and α = 90° (**a**) when degree of misalignment D_M = 0; (**b**) D_M = 0.6; (**c**) D_M = 0.8; (**d**) D_M = 0.9.

1.4. Temperature Field

With the recent trend in limit design, bearings are expected to function satisfactorily when subjected to higher loads and speeds. As the bearing operational envelope is pushed to its limit, heat effects become the major bottleneck that must be accounted for at the design stage. This presents a formidable challenge in practice since bearing misalignment is nearly unavoidable. Severe operating conditions generate significant heat that must be properly dissipated otherwise bearings become failure prone. The bushing of many turbine bearings are lined with a protective soft metal such as Babbitt. Babbitt is known to undergo plastic flow at 150 °C [4–6] under pressure. This is directly related to the bearing maximum temperature. The bearing maximum temperature becomes more severe when the bearing is misaligned. Large temperatures leading to scuffing failure is also possible in the presence of scuffing contamination [7]. To this end, it is particularly important to develop techniques capable of realistically predicting the bearing maximum temperature, a parameter known to be an important design parameter [6,8,9].

The maximum temperature increases with the increase in the degree of misalignment. Whereas in stationary-loaded aligned journal bearings the maximum temperature occurs in the bearing mid-plane in the vicinity of the minimum film thickness, the situation is different in misaligned bearings: the location of the maximum temperature shifts away from the central plane. Extensive experimental tests with aligned bearings clearly indicate that the shaft temperature is uniform circumferentially and to a great extent in the axial direction [10], so that the rotating shaft can be practically treated as an isothermal element. This experimental observation was, in fact, used as a boundary condition to analytically predict the temperature of an aligned shaft with great accuracy when compared to experimental measurements. In a perfectly aligned, stationary-loaded bearing a two-dimensional heat transfer analysis generally provides satisfactory results, for thermohydrodynamic analyses show that, indeed, the predictions closely match the experimental results. The interested reader is referred to analyses [6,8,11–15] where comparison with experimental results of references [16–18] are shown to verify the predictions. These analyses also provide a generalized procedure for predicting the effective and maximum bearing temperature as reported in references [9,19,20]. For dynamically-loaded, engine bearings, Allmaier *et al.* [21] finds that only about 10% of the heat enters the journal or the supporting structure and finds 2D solution reliable. In contrast, in a misaligned bearing, the journal temperature varies considerably along the axial direction. Accordingly, heat flows into the bushing not only radially but also circumferentially and axially. In the case of misaligned bearing, extensive simulations show that the pressure, temperature, and cavitation field can be significantly affected [22]. It, therefore, follows that a full three-dimensional heat transfer consideration is necessary to accurately characterize the thermal field in a misaligned bearing. As we shall discuss later in this article, the results of many reports attest to the fact that the temperature at the bush-oil interface increases with increase in misalignment at all load values, and that it varies considerably in both the circumferential and axial directions.

An analysis that fully considers the fluid and solid thermal field with the non-linear coupling of the Reynolds and energy equation through the viscosity-temperature relationship (e.g., $\mu = \mu^* e^{(T-T^*)}$, where T^* and μ^* are the reference temperature and its corresponding viscosity) is known as thermohydrodynamic (THD) analysis. THD analyses almost always require numerical treatment of partial differential equations using either the finite difference [8,11,23–25] or finite element methods [26]. If the pressure and/or the temperature field become severe enough to warrant consideration of the deformation of the surfaces due to load or thermal expansion, then the elasticity equations also come in to play. These problems fall into the category of thermoelastohydrodynamic (TEHD) analysis [6,27–29]. The thermohydrodynamic (THD) formulation involves treating the energy equation to obtain the temperature profile in the fluid film.

$$\rho c\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\eta\left[\left(\frac{\partial u}{\partial y}\right)^2+\left(\frac{\partial w}{\partial y}\right)^2\right] \tag{7}$$

The term on the left-hand side represents the energy transported by convection. The terms on the right-hand side are the heat conduction and the dissipation, respectively. The conduction terms in the axial and circumferential directions are negligible in comparison to the conduction across the film for the small clearance. For most conventional lubricants, the viscosity decreases exponentially with increasing temperature. The relationship between the viscosity and temperature couples the Reynolds equation and the energy equation.

The temperature field in the bushing and journal can be computed by solving the heat conduction equation in cylindrical coordinates.

$$\rho_I c_I \omega_I \frac{\partial T_I}{\partial \theta} = k_I \left(\frac{\partial^2 T_I}{\partial r_I^2} + \frac{1}{r_B} \frac{\partial T_I}{\partial r_I} + \frac{1}{r_I^2} \frac{\partial^2 T_I}{\partial \theta^2} + \frac{\partial^2 T_I}{\partial z_I^2} \right) \tag{8}$$

where subscript I represents either the bush or the journal. Parameter ω_I is the rotating speed and the rotating speed of the bush is generally zero. A condition requiring the continuity of heat flux is applied at the interfaces. When misalignment exists, the bearing temperature field is no longer symmetric about the mid-plane and the heat flux in the shaft is more complicated. Appropriate treatment of the heat conduction in the rotating shaft is required for determining the temperature field in misaligned bearing.

The temperature field in the bush and journal can be computed by solving the heat conduction equation in cylindrical coordinates.

$$\rho_I c_I \omega_I \frac{\partial T_I}{\partial \theta} = k_I \left(\frac{\partial^2 T_I}{\partial r_I^2} + \frac{1}{r_B} \frac{\partial T_I}{\partial r_I} + \frac{1}{r_I^2} \frac{\partial^2 T_I}{\partial \theta^2} + \frac{\partial^2 T_I}{\partial z_I^2} \right) \tag{9}$$

where subscript I represents either the bush or the journal. Parameter ω_I is the rotating speed and the rotating speed of the bush is generally zero. A condition requiring the continuity of heat flux is applied at the interfaces. When misalignment exists, the bearing temperature field is no longer symmetric about the mid-plane and the heat flux in the shaft is more complicated. Appropriate treatment of the heat conduction in the rotating shaft is required for determining the temperature field in misaligned bearing.

The flow chart for THD solution of the misaligned bearing is shown in Figure 3. It shows that the pressure field and temperature field are coupled. Generally, there are 4 unknowns, the attitude angle, eccentricity ratio, degree of misalignment and misalignment angle. The attitude angle and the eccentricity ratio can be determined from the load vector, and the degree of misalignment and the misalignment angle can be determined from the moment vector. Error between two successive iterations can be computed using the following expression

$$E_\Psi = \frac{1}{N} \sum_{i=1}^{N} \frac{\Psi^{now} - \Psi^{old}}{\Psi^{now}} \tag{10}$$

where E_Ψ is the error, Ψ can be the temperature or fractional film content to be solved and N is the total number of nodes.

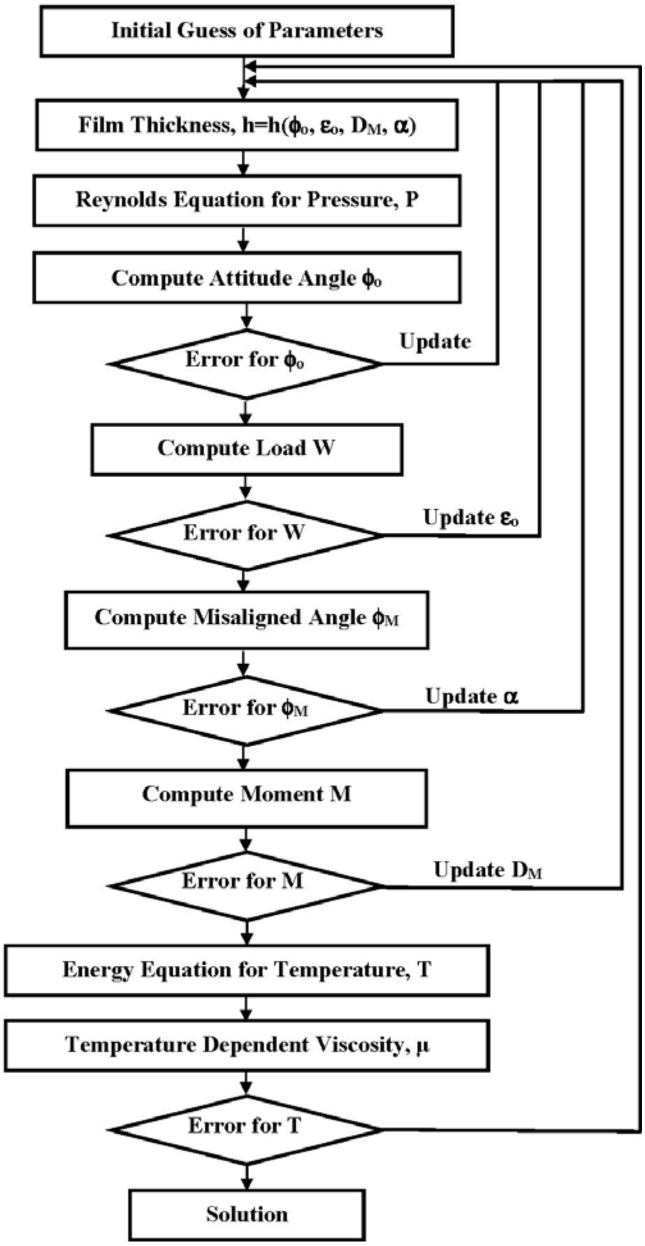

Figure 3. Flow Chart for thermohydrodynamic (THD) solution of the misaligned bearing.

Having described the appropriate equations that govern the fluid mechanics and heat transfer, we now proceed to describe the computational results as well as experimental findings reported in the literature. The summary of literatures is listed in Table 1.

2. Literature Review

2.1. General Effects

The performance characteristics of misaligned journal bearings are generally functions of load. The bearing characteristics are normally classified as static and dynamic characteristics. The static characteristics include eccentricity ratio, attitude angle, Sommerfeld number, friction force, maximum hydrodynamic pressure, minimum film thickness, leakage flow-rate, mean and maximum temperatures. The dynamic characteristics normally include stiffness coefficients, damping coefficients, and stability threshold, *etc.* Misalignment causes an appreciable reduction in the load-carrying capacity. Misalignment also reduces the minimum film thickness drastically, and makes the pressure distribution asymmetric. Misalignment reduces the leakage flow-rate at higher loads. Somewhat surprisingly, however, it has been shown that misalignment significantly improves the stability margin of all journal bearing systems. Experimental results verify that journal bearings are relatively more stable under misalignment [30] and the stability is even better when the load is higher [31]. Noting that power loss induced by bearing misalignment is greater than an aligned bearing, Safar [32] in 1984 found that an optimum position of misalignment is almost along the axial plane containing the load vector due to lower frictional losses and lubricant leakage.

2.2. Early Discoveries

Experimental observations. One of the first documented investigations on journal bearing misalignment is reported by McKee and McKee [33] in 1932 who experimentally observed its influence on the axial symmetry of hydrodynamic pressure and noting that the location of the peak hydrodynamic pressure moves from the axial center of towards the bearing ends. Later in 1940, Cowlin [34] identified the misalignment as one of seven major causes that provides the discrepancies between numerical simulations and the observed bearing performances. Pigott [35] showed that a 40% reduction in bearing load capacity was induced by a 0.0002 radian misalignment. These observations clearly revealed the importance of misalignment on the bearing performances.

Further progress was reported by DuBois *et al.* [36] in 1951 who showed that the pressure distribution of a misaligned bearing is not symmetric, and reported that the maximum pressure was located at the bearing ends. They observed that when a bearing is subjected to severe misalignment, the maximum pressure increases and the bearing performance deteriorates due to the permanent deformation at bearing ends. Subsequently, in 1957, DuBois *et al*. [37] introduced a new parameter called the degree of misalignment for quantifying the severity of misalignment. This parameter is equal to one when the contact occurs between the shaft and the housing.

Analytical treatments. The next early progress was in analytical studies. Ausman [38] in 1960 used the perturbation method and obtained the torque generated by a misaligned gas-lubricated journal bearing. They presented results graphically in terms of non-dimensional parameters. Galletti-Manacorda and Capriz [39] in 1965 obtained the torque generated in a misaligned short cylindrical bearing. The torque was derived in the special case of a journal “whirling and rocking” in the neighborhood of its steady-state position under load. Rice [40] in 1965 computed the misalignment torque of gas lubricated bearings, where the pressure distribution is solved from the Reynolds equation using the finite difference method. They provided design curves of the misalignment torques at different aspect ratios and the eccentricity ratios. Smalley and McCallion [41] in 1966 investigated the misalignment effect of ungrooved journal bearings on bearing performances at two aspect ratios under steady running conditions. They presented their results in the form of non-dimensionalized graphs, showing the variation of direct load and misalignment couple with the misalignment parameters, and validated their predictions by comparing results with the experimental measurements reported by Ocvirk [42].

2.3. Application to Different Types of Bearings

Stockley and Donaldson [43] in 1969 presented the widely-used Raymondi and Boyd diagrams for the centrally loaded 180° partial hydrodynamic journal bearings with provision for shaft misalignment. They showed simulation results for aspect ratios of 1, 0.5, and 0.25 and a range of tilt angles are presented in design charts and tables.

Pinkus and Bupara [44] in 1979 presented a comprehensive isothermal solution for a two-groove finite bearing in the form of charts and tables. The equations are valid for finite bearings with grooves at any angular position where the misalignment can vary in both magnitude and direction with respect to the bearing boundaries. However, the application of their data is inconvenient since the load direction varies depending on the eccentricity and the misalignment. Mokhtar *et al.* [45] in 1983 showed that the misalignment of journal bearings with both *axial and spiral feeding* has a pronounced effect on bearing performance, particularly when the minimum film thickness approaches zero. They showed that bearings with axial feeding produce higher loads and smaller friction coefficient than bearings with spiral feeding. Safar *et al.* [46] in 1985 obtain an adiabatic solution of the misaligned finite journal bearings considering the both axial and spiral oil inlet conditions. They showed that thermal effects are more pronounced for bearings with axial rather than spiral oil inlet grooves.

Ikeuchi *et al.* [47] in 1985 investigated the effect of misalignment on the static performances of bearings with a circumferential groove. A modified Reynolds equation was used to take into account the local proportion of fluid. They concluded that the minimal film thickness was reduced by the misalignment at one end of the bearing leading to the reduction of the load capacity. Wang *et al.* [48] in 2006 developed a mixed EHL model to investigate the effects of misalignment and elastic deformations on the performance of coupled journal-thrust bearing system. They found the elastic deformation reduces the load capacity in both journal and thrust bearing. Howard [49] in 2009 studied the misaligned gas foil journal bearing. He presented a preliminary experimental investigation of the misalignment tolerance of gas foil journal bearing systems. Their experimental results indicate that gas foil bearings are quite tolerant of high levels of misalignment.

2.4. Consideration of Cavitation Effects

Vijayaraghavan and Keith [1] in 1989 analyzed the effect of cavitation on the performance of a line-grooved misaligned bearing for both flooded and starved inlet conditions. They used the mass-conserving cavitation algorithm in their analysis. They took into account the lubricant rupture and the reformation phenomena. One year later, Vijayaraghavan and Keith [50] showed that at the higher degrees of misalignment, the performance characteristics of the bearing are significantly different from those for an aligned journal bearing. Pierre *et al.* [51] in 2004 studied the three-dimensional THD behavior of misaligned plain journal bearings with theoretical and experimental approaches under steady-state conditions. Their model includes lubricant film rupture and reformation phenomena by mass conservation. They also investigated the influence of the misalignment direction both numerically and experimentally. Jang and Khonsari [22] in 2010 developed a three-dimensional THD model considering the film rupture and reformation for the misaligned journal bearing using the mass conservative Elrod algorithm. They provided an extensive set of numerical solutions to closely examine the effects of misalignment in journal bearings.

2.5. Dynamic Characteristics

Singh and Sinhasan [52] in 1989 studied the effects of small misalignment on the dynamic characteristics of an ungrooved big end bearing using the finite element method. They found that the minimum film thickness is about 28% smaller than that obtained in the parallel axes bearing. Choy *et al.* [53] in 1992 analyzed the nonlinear behavior of the bearing stiffness and damping of a misaligned hydrodynamic cryogenic journal bearing by means of a finite difference solution technique. They solved the coupled Reynolds equation and liquid oxygen property equation for fluid film pressure iteratively. They studied parametrically the effects of input variables such as the inlet temperature, applied load, rotational speed, and misalignment on the axial and circumferential pressure distribution and the non-linear behavior of the bearing stiffness and damping. Nikolakopoulos and Papadopoulos [54] in 1994 presented an analysis of misaligned journal bearing operating considering both the linear and nonlinear plain journal bearing characteristics. The finite element method was used to solve the Reynolds equation. They calculated the linear and nonlinear dynamic properties for the misaligned bearing depending on the developed forces and moments as functions of the displacements and misalignment angles.

2.6. Experimental Tests

Asanabe *et al.* [55] in 1972 studied the misalignment of a grooved bearing, limiting their consideration to the misalignment in the vertical plane only. They noted that the pressure distribution was seriously affected by the misalignment. Nicolas and Frene [56] in 1973 compared theory and experimentation for a bearing subjected to a central load and a misalignment torque, with an axial groove. They showed that plain bearings have a weak resistance against misalignment. Tieu and Qiu [57,58] in 1996 built an experimental device to measure the film thickness, the hydrodynamic pressure and the temperature in the misaligned bearing for static and dynamic approaches. The relationships of eccentricity, attitude angle, and side flow to the Sommerfeld number are experimentally determined. Arumugam *et al.* [59] in 1997 conducted experiments to investigate the effects of journal misalignment on the performance of three-lobe journal bearings. They analyzed the dynamic characteristics of a bearing under the horizontal misalignment, and showed that the system damping increases as the misalignment increases.

Huber *et al.* [60] in 1998 investigated the misaligned plain circular bearing under a static load experimentally and theoretically. In the theoretical investigation, the adiabatic oil film and static equilibrium position of the journal were assumed. The pressure, oil flow and power loss were compared with the results observed from experimental investigations.

Bouyer and Fillon [61] in 2002 experimentally studied the hydrodynamic plain journal bearing submitted to a misalignment torque. Hydrodynamic pressure and temperature in the mid-plane of the bearing, temperatures in two axial directions, oil flow rate, and minimum film thickness were measured for various operating conditions and misalignment torques. They showed that the maximum pressure in the mid-plane decreased by 20% at their maximum torque of 70 Nm while the minimum film thickness was reduced by 80%. The misalignment caused more significant changes in bearing performance when the rotational speed or load was low.

2.7. Surface Roughness

The effect of surface roughness on the performance characteristics of bearings has long been of interest in tribology and has intensified in recent years. The effects of surface roughness on the hydrodynamic pressure are included using either stochastic model [62] or the average flow model proposed by Patir and Cheng [63,64]. The stochastic model is limited to one dimensional ridges oriented either longitudinally or transversely. The average flow model modifies the Reynolds equation by using the pressure and shear flow factors to adjust the amount of the flow-rate. The pressure and shear flow factors vary with the standard deviation of the surface roughness (rms) and the surface pattern. It is shown that the load capacity decreases and the attitude angle increases with an increase in the roughness parameter. The leakage flow-rate increases with an increase in the roughness parameter at higher values of the eccentricity ratio for a certain value of the degree of misalignment. Note that these results are under the operating conditions where the metal-to-metal contact does not occur. If the metal-to-metal contact is included, then contact pressure is generated at the asperity level, which could contribute to supporting a portion of the applied load. The contact pressure can be determined from the elastic contact models [65,66] or the elastic-plastic/plastic contact models [67,68].

Abdel-Latif and Mokhtar [69] in 1988 investigated misalignment in full finite journal bearings with rough surfaces. The roughness function was considered to be a randomly varying quantity of zero mean, measured from the nominal level. They concluded that the combined influence of bearing surface roughness and misalignment is more pronounced at higher eccentricities. Guha [70] in 2000 investigated the steady-state characteristics of finite hydrodynamic journal bearings considering two types of misalignment—axial (vertical displacement) and twisting (horizontal displacement)—with isotropic surface roughness. They obtained the steady-state performance characteristics for various values of isotropic roughness parameter, eccentricity ratio and degree of misalignment. Sharma *et al.* [71] in 2002 studied the combined effect of journal misalignment and surface roughness on the performance of hybrid journal bearing. They showed that the surface roughness compensated the decrease in minimum film thickness caused by shaft misalignment. Sun *et al.* [72] in 2010 studied the effects of journal misalignment on the bearing characteristics based on the THD model considering the roughness. The temperature of the oil film was investigated under different misalignment angles and surface roughness. Sun *et al.* [73] in 2014 analyzed the lubrication characteristics of misaligned journal bearings considering the viscosity-pressure effect of the oil, the surface roughness and the elastic deformation at the same time. They found that the oil viscosity-pressure relationship and the deformation of the bearing have a significant effect on the lubrication of misaligned journal bearings. They also showed that surface roughness affects the lubrication of misaligned journal bearings when both the eccentricity ratio and misalignment angle are large.

2.8. Structural Effects

Sun and Gui [74] in 2004 studied the effect of journal misalignment caused by shaft deformation. The pressure, load-carrying capacity, attitude angle, end leakage flow-rate, frictional coefficient, and misalignment moment were calculated for different values of misalignment degree and eccentricity ratio. The results showed that there are obvious changes in pressure distribution, the maximum pressure, film thickness, the minimum film thickness, and the misalignment moment when misalignment takes place. Later Sun *et al.* [75,76] in 2005 investigated the effects of journal misalignment caused by shaft deformation under static and rotary loads. The results showed that the higher the load on the shaft, the larger is the journal misalignment resulted from shaft deformation, and the more obvious becomes the effect on lubrication performance of journal bearing. The main effects are: The maximum pressure increases markedly, the minimum film thickness reduces, the leakage flow-rate increases, and the friction coefficient changes slightly. Ebrat *et al.* [77] in 2004 developed a new model based on a local perturbation of the oil film that captures the effects of misalignment and bearing structural deformation in rotor dynamics and engine NVH (Noise, Vibration and Harshness) applications.

2.9. Thermal Effects

Many different types, shapes and journal bearing configurations exist that defer in the film thickness geometries, bushing configuration, and the lubricant supply ports and delivery system. Plain bearings, axially- or radially-grooved, circular, elliptical, multi-lobe, and tilting-pad bearings are only a few examples. The THD analysis of each specific bearing with appropriate boundary conditions is needed for assessing their performance due to misalignment, particularly when the load is heavy and the degree of misalignment is large.

Misalignment in journal bearings causes the maximum temperature to increase and minimum film thickness to decrease. Research shows that the performance characteristics of the bearing could be significantly improved by adding either a local or a global defect in the bearing geometry when it is subjected to extreme operating conditions to, in effect, locally increase the film thickness. Bouyer and Fillon [25] in 2003 showed that that the performance characteristics of the bearing could be significantly improved by adding a defect in the bearing surface. However, because of the complexity of the THD analysis, the optimal location and depth of the defect varies depending on the specific bearing geometry and the operating conditions for each bearing. Thus, developing a complete TEHD analysis the designer is in a position to improve bearing performance by adjusting the film thickness to counterbalance the deterioration caused by misalignment. When the lubricant is subjected to high pressures such as in EHL, the viscosity is strongly dependent upon the pressure, which is so called piezoviscous lubrication. Piezovisous effects should be considered in calculations coupled with any non-Newtonian effects such as shear thinning. The interested reader can refer to Allmaier *et al.* [78] within the context of EHL [79–82].

Mokhtar *et al.* [83] in 1985 obtained an adiabatic solution for a misaligned journal bearing. They showed that the misalignment effects on the bearing performances are more significant when thermal effects are considered, as expected. Medwell and Gethin [23] in 1986 observed that the misalignment torque intensity induces lubricant overheating in two-lobe bearings, and noted that the misalignment reduces the rupture zone. Banwait *et al.* [31] in 1998 conducted a thermohydrodynamic (THD) analysis of a misaligned circular plain journal bearing. They investigated the influence of the misalignment on the pressure and temperature and the minimum film thickness.

Pierre *et al.* [84] in 2002 compared the experimental data and theoretical results of three-dimensional THD study of a misaligned plain journal bearing. They concluded that the temperature increases with increasing misalignment. He *et al.* [85] in 2012 investigated the effect of the journal misalignment caused by an asymmetric rotor structure. They simplified the rotor system as a step shaft problem based on the structure stiffness equivalent characteristic. Their results indicate that bearing performances are greatly affected by misalignment caused by the asymmetric structure. A simple stepped shaft can effectively represent a misaligned journal bearing in a rotor-bearing system. Thomsen and Klit [29] in 2012 proposed a TEHD flexure journal bearing model that improves the hydrodynamic performance compared to the traditional bearing when misaligned. Furthermore, they investigated the influence of a compliant bearing liner and found that it increases the hydrodynamic performance when applied to a stiff bearing, whereas the liner has practically no influence on the flexure journal bearing's performance.

2.10. Thermally-Induced Bearing Failure

Khonsari and Kim [86] in 1989 examined the influence of misalignment on the bearing seizure during start up. They showed that the rapid thermal expansion of the shaft during the start-up period can lead to a complete loss of bearing clearance and seizure in the absence of lubrication, and the presence of shaft misalignment can significantly influence the time of seizure. Wang [87] gives a review of relevant literature to this type of a failure which is more likely to occur in bearings that have been out of service for an extended period of time so that the system runs dry at the first usage. Similar type of a problem can occur if the oil supply is interrupted during the operation and the bearing runs dry. Of course, the thermally-induced seizure is not confined to start up condition in misaligned bearings. It can occur in perfectly aligned bearings with dry contact [88–92] as well as lubricated bearings with or without radial load [93–95]. Unloaded journal bearings widely used in vertical pumps, agitators and mixers and special care must be taken in their design to avoid failure.

2.11. Tilting-Pad Journal Bearings

Monmousseau and Fillon [27] in 1999 analyzed the static and dynamic misalignment effects on tilting-pad journal bearings. They investigated the influence of thermal and deformation effects on bearing performances and concluded that the bearing characteristics are influenced significantly by the misalignment amplitude and dynamic load, and three-dimensional thermoelastohydrodynamic (TEHD) analysis is necessary under severe misalignment. When misalignment is small, the decrease in oil film thickness can be compensated by an increase in the oil film thickness due to elastic and thermal distortions of solid surfaces. For tilting-pad journal bearings, it may be more efficient to select the pad material with a certain amount of elasticity to compensate misalignment. El-Butch and Ashour [28] in 2005 analyzed the performance of a misaligned tilting-pad journal bearing under transient loading condition. They considered both the elastic and thermal distortions of the pad. They showed that the TEHD distortion improves the bearing performance in the case of misalignment shaft. They also showed that at a low shaft misalignment, the decrease in oil film thickness due to shaft misalignment is compensated by the increase in oil film thickness due to elastic and thermal distortions.

2.12. Non-Newtonian Lubricants

Application of non-Newtonian fluids to improve performance in due to the development of modern machine equipment is growing. Common non-Newtonian lubricants are the polymer-thickened oils, lubricants with additives or suspended lubricating particles, metal particles and dirt. Such non-Newtonian lubricants show that the stress is not directly proportional to the shear strain. Thus, the formulation of the governing equations becomes considerably more involved. Buckholz and Lin [96] in 1986 investigated the misalignment effects on load and cavitation for partial arc journal bearings lubricated with non-Newtonian fluid by using the matched asymptotic expansion under the isothermal condition. Jang and Chang [97] in 1987 presented an adiabatic solution for a finite hydrodynamic journal bearing with non-Newtonian lubricants obeying the power law model for misalignment angles of 0.0001 and 0.0002 radians. Their adiabatic solutions showed that the thermal effects are more pronounced for higher values of flow behavior index, higher eccentricity ratios and larger misalignment angles.

Boucherit *et al.* [98] in 2008 investigated the steady-state and dynamic analysis of misaligned compliant journal bearings considering the effects of couple stresses arising from the lubricant blended with polymer additives. They derived a modified form of the Reynolds equation based on the Stokes micro-continuum theory, and presented the effect of the misalignment and the couple stress parameters on static and dynamic performances. Osman [99] in 2001 developed a generalized Reynolds equation suitable for misaligned hydrodynamic journal bearings lubricated by either Newtonian or non-Newtonian ferrofluid. Ferrofluid lubrication is based on the possibility of adding magnetically active particles to both Newtonian and non-Newtonian fluids. He presented design charts for load capacity, attitude angle, side flow, friction force and misaligned moments under different misalignment conditions. He concluded that magnetic lubrication generally improves the bearing performance under misalignment.

Recently, many micro-continuum theories have been developed for describing the peculiar behavior of non-Newtonian fluids. In practice, *i.e.*, nuclear power, it is very difficult to remove dirt and metal particles from the lubricant and, therefore, the micro-polar lubricant model is helpful for analysis of the performance of bearings. The micro-polar fluid model predicts enhancement in the load-carrying capacity and improvement in the friction parameter compared to the Newtonian fluid when the bearing is subjected to misalignment. The micro-polar fluids generate more moment than a Newtonian fluid. Das *et al.* [100] in 2002 studied the performance of steady state misaligned journal bearing lubricated with micro-polar lubrication theory considering axial and twisting misalignment. The load-carrying capacity, misalignment moment and friction parameter are presented at various values of eccentricity ratio, degree of misalignment and micro-polar fluid characteristic parameters. They noted that, under misalignment conditions, micro-polar fluids exhibit better performance than Newtonian fluids. The gas lubricated journal bearings were also investigated by some researchers [37,38,40].

2.13. Turbulent Effects

In the original derivation of the Reynolds equation, the flow is assumed to be laminar. However, the flow can become turbulent for high speed bearings particularly when operating with low viscosity lubricant such as water or cryogenic fluids. Safar and Riad [101] in 1988 studied the influence of journal misalignment on the performance characteristics of journal bearings operating in the turbulent regime. They concluded that the percentage increase in the friction coefficient due to the misalignment reduces with the increase of the load capacity, as a result of the increase of the eccentricity ratio. Safar *et al.* [102] in1989 further investigated the bearing characteristics of the misaligned journal bearings operating in turbulence. They showed that the misalignment significantly influences the bearing characteristics especially at lower eccentricity ratios.

Shenoy and Pai [103] in 2009 investigated theoretically the effect of turbulence and misalignment on the performance characteristics of a centrally loaded 120° single pad externally adjustable bearing under the steady-state operation. They found that for a given eccentricity, adjustment configuration and degree of misalignment with unidirectional load increases the friction variable and shaft attitude angle. Abass and Sahib [104] in 2013 studied the effect of bearing compliance on THD lubrication of high-speed, misaligned journal bearing with a compliant liner lubricated with bubbly oil. Their model included the oil film turbulence and the liner elastic deformation. Ram and Sharma [105] in 2014 studied the influence of wear on the performance of capillary-compensated hole-entry hybrid misaligned journal bearings under a turbulent regime. The wear due to start/stop operations using Dufrane's wear model is included in their model. It was found that when a worn bearing operates in laminar/turbulent regimes, the reduction in the minimum fluid film thickness is more due to misalignment as compared to the aligned bearing operates in laminar regime. Xu *et al.* [106] in 2014 investigated the static and dynamic characteristics of the misaligned journal bearing under the turbulence. Their results showed that both the turbulent and thermal effects will obviously affect the lubrication of misaligned journal bearings when the eccentricity or misalignment ratio is large.

2.14. Dynamic Loading and Engine Bearings

Dynamic load arises from the shaft orbital motion in the oil film clearance due to imbalance, misalignment, and other non-static forces. The misalignment and bearing structural deformation become especially important since they greatly affect the oil film distribution and, therefore, the bearing dynamic characteristics.

Of particular interest is the study of internal combustion engine crankshaft bearings which is a complex problem when misalignment is involved because it requires not only the determination of the eccentricity ratio and the attitude angle from the load but also the determination of the misalignment parameters (D_M and α) from the moment acting on the shaft. In such bearings, the misalignment is essentially due to the elastic deflection of the crankshaft. Misalignment parameters vary during an engine cycle since the bearing load and its direction vary. The failure of engine bearings in the crankshaft can be very costly to repair.

Goenka [107] in 1984 studied a misaligned big-end connecting-rod bearing under the dynamic loading using the finite element technique. His formulation can be used for various type of bearing configurations and grooves. Lahmar *et al.* [108] in 2002 analyzed the effect of journal misalignment on the hydrodynamic performance of a main crankshaft bearing with mobility method of Booker [109]. They showed that for small misalignment of the main journal, the minimum film thickness and the journal center trajectories differ greatly from that of an aligned journal. Furthermore, the use of the circumferential feed grooves can cause a destructive metal-to-metal contact due to a significant reduction in the minimum film thickness.

Misalignment defect in internal combustion engine bearings, particularly the main crankshaft bearings, must be considered as an important parameter at the design stage. Boedo and Booker [110] in 2004 comprehensively researched the steady-state and transient performances of misaligned bearings. They found that misaligned bearings have infinite load and moment capacity as the end-plane minimum film thickness approaches zero under transient journal squeeze motion and under steady load and speed conditions. These results differ markedly from finite capacity trends reported previously in both numerical and experimental studies. Li *et al.* [111] in 2012 studied the effects of journal misalignment on the transient flow of a finite grooved journal bearing in the flexible rotor-bearing systems. They indicated that bearing performance is greatly affected by misalignment caused by unbalanced excitation, and the new CFD method based on the interaction between transient fluid dynamics and rotor dynamics can effectively predict the transient flow field of a misaligned journal bearing in a rotor-bearing system.

For heavily loaded applications, distortion of the bearing components is significant and cannot be ignored. The distortion arises due to applied load and thermal deflection. Large crankshafts are most commonly used on marine diesel engines and they should remain straight during operation. The crankshaft consists of crank webs, crank pins and journals along its length. The misalignment of the crankshaft is critical due to its length, leading to serious damage on the engines. Crankshaft web deflection measurements have been used to evaluate crankshaft alignment and as criteria for the realignment. The main bearing center lines can be calculated from the web deflection data analytically. This analytical process for the misalignment profile of the main journals can reduce the huge cost of the alignment correction prior to tearing down engines.

2.15. Wear Effects

Wear is progressive loss of material from the surfaces in contact as the result of relative motion between them. Wear is the most influential factor which shortens the effective life of machine components. The main purpose of a lubricant (liquid or solid) is to minimize the friction and reduce the wear. The types of wear are the abrasive wear, erosive wear, adhesive wear, surface fatigue, and corrosive wear. Landheer *et al.* [112] in 1990 provided transition diagrams for plain journal bearings operating under conditions of moving contact on the basis of experimental data in terms of Stribeck curves and a few simple principles from lubrication and heat transfer theories. These diagrams may serve as wear mechanism maps. Ligterink *et al.* [113] in 1996 derived the wear equation for the calculation of the specific wear rate of the bearing material as a function of the wear depth based on the Holm/Archard wear law.

The shaft misalignment in bearings is one of the most common causes of wear. Wear can be observed in both the dry and lubricated bearings which are affected by speed, load, and temperature and working time. Bearings are particularly susceptible to wear during the startup operation, before the shaft lifts off and prior to the generation of hydrodynamic pressure to protect the surfaces from coming into contact. A similar situation also exists during the coast- down operation when the engine is shut down and the speed drop to zero. Thus, stop/start is responsible for significant wear during the life of a bearing for both aligned and misaligned bearings. During the starting and stopping periods, the lubricant film thickness is not adequate enough to separate the surfaces and lift off does not occur [114]. Below the lift-off speed, mixed lubrication prevails and the surface interaction at the asperity level should be taken into consideration. For this purpose, the load-sharing concept can be employed. Progress toward this for EHL line contact has been reported in references [115–117]. Progress in engine bearing has been reported by Priestner *et al.* [118]. These papers have dealt with aligned configuration. Extension to misaligned configurations requires further research. The operation of the bearing with misalignment is more severe because, additionally, it can cause load concentration on the bearing edges, leading to the mixed lubrication, unstable operation and intensive wear of mating parts.

Nikolakopoulos and Papadopoulos [119] in 2008 studied the friction coefficient of worn misaligned journal bearings operating under severe hydrodynamic lubrication. They found that the friction coefficient generally increases with increasing wear depth as well as misalignment and Sommerfeld number. Ram and Sharma [120] in 2013 studied the combined influence of journal misalignment and wear on the performance of a hole-entry hybrid journal bearing system under turbulence theoretically. They solved modified Reynolds equation based on the Constantinescu lubrication theory by using the finite element method together with orifice and capillary restrictors flow equations as a constraint. They found that for a symmetric hole-entry journal bearing the minimum film thickness is more for the bearing compensated by orifice restrictor in comparison to the capillary restrictor when bearing operates in turbulent regime under worn/unworn conditions.

2.16. Hydrostatic Bearings

Andres [121] in 1993 studied the hydrostatic journal bearing operating in the turbulent regime including journal misalignment effects. He showed that the journal misalignment causes a reduction in the load carrying capacity due to the loss in fluid film thickness, increases the lubricant flow-rate and produces significant misaligning moments. Jain *et al.* [122] in 1997 studied the static and dynamic performance characteristics of a capillary-compensated hole-entry hybrid misaligned journal bearing. Their study suggested that the misalignment significantly affects the performance of the hole-entry journal bearing, and need to be included for more accurate predictions.

2.17. Porosity Effects

Elsharkawy [123] in 2003 investigated the effect of journal misalignment on the performance of flexible porous journal bearings. They showed that the effect of misalignment in porous journal bearings can be neglected when the permeability is high. Gulwadi and Shrimpling [124] in 2003 studied the effect of shaft tilting due to moment acting on it during an engine cycle. They described the features and capabilities of a computational tool ORBIT developed for analyzing journal bearings, *i.e.*, connecting rod bearings and crankshaft bearings. ORBIT is capable to consider: (i) non-circular journal bearing geometry; (ii) oil-feed holes/grooves; (iii) surface roughness; (iv) journal misalignment, (v) rise in oil temperature; and (vi) bearing elasticity effects (EHL) on bearing performance.

2.18. Variable Axial Profile

Heat elongation of bearing supports during the operation changes the static deflection line of the rotor determined during assembly and it causes an increase in the stresses on the bearing edges. To avoid the edge stresses, the variable axial profile (*e.g.*, hyperboloidal profile) can be applied to the bearings. The variable axial profile extends the bearing operation range without the stress concentration on the edges of bush. These bearings successfully carry the extreme load under the misalignment eliminating the necessity of using self-aligning bearings. Strzelecki [125] also in 2005 investigated the variable axial profile of the bearing. He showed the variable axial profile, *e.g.*, hyperboloidal convex profile, extends the bearing operation range without the stress concentration on the edges of bush.

The following summary table provides a convenient guide for the available literature on the subject of misaligned journal bearings.

Table 1. Summary.

Bearing Category	References
Adiabatic Solution	[46,83,97]
THD/TEHD Analysis	[4,12,15,19,20,23,25,27–29,31,51,72,82,84–86,104,106]
Transient Condition	[27,28,76,86,107–111]
Roughness	[69–73,117]
Turbulence	[101–106,120,121]
Non-Newtonian	[14,15,53,79,81,96–100,104]
Gas Bearing	[37,38,40,49]
Wear or Defect	[25,105,113,116,118–120]
Dynamic Characteristics	[52–54,58,77,98,106,122]
Experiment	[16–18,49,51,55–57,59–61,75,76,115]
Three-Lobe Journal Bearing	[59]
Tilting-Pad Journal Bearing	[20,27,28]
Hole-Entry Hybrid Journal Bearing	[71,105,120]
Axial and Spiral Oil Inlet	[45,46]
Circumferential Groove	[47,108]
Axial Groove	[1,22,25,29,44,50,51,56,77,104,111,125]

3. Conclusions

Hydrodynamic journal bearings are commonly used in various rotating machines such as pumps, compressors, fans, engines, turbines and generators. Hydrodynamic bearings quite often operate under some degree of misalignment. Investigations of misalignment of the journal bearings date back to the 1930s. It is shown that the static and dynamic characteristics are significantly influenced by misalignment, particularly when the load is heavy and misalignment is large. Generally, the misalignment is disadvantageous on the performance of the journal bearings, leading to the bearing failure. Of late, many influencing factors, such as bearing configuration, groove type and its location, thermal effect, surface roughness effect, elastic and thermal deformation, and non-Newtonian lubricants, *etc.*, are considered to more accurately predict the behavior of misaligned journal bearings. Significant progress has been made in the development of thermohydrodynamic analyses of misaligned journal bearings. The development of the appropriate boundary conditions for the heat transfer and cavitation remain to be active areas of research. Thus far, the majority of research activities have focused on stationary-loaded bearings and much less analyses are available on the TEHD analysis of dynamically loaded bearings with proper cavitation analysis. Another area where research is needed is the development of an analytical tool for prediction of wear in engine bearings, particularly for bearings with a protective over-layer and its wear as a function of time.

Acknowledgments

The authors wish to thank Greg Anderson of Naval Ship Systems Engineering Station, Naval Surface Warfare Center, Carderock Division (NAVSSES) for reviewing this article and making insightful comments.

Author contributions

Both authors contributed to the writing of this article.

Conflicts of Interest

The authors declare no conflict of interest.

References

1. Vijayaraghavan, D.; Keith, T.G. Effect of Cavitation on the Performance of a Grooved Misaligned Journal Bearing. *Wear* **1989**, *134*, 377–397.
2. Elod, H.G. A Cavitation Algorithm. *ASME J. Lubr. Tech.* **1981**, *103*, 350–354.
3. Elrod, H.G.; Adams, M. A Computer Program for Cavitation and Starvation Problems. In Proceedings of the 1st Leeds-Lyon Symposium on Cavitation and Related Phenomena in Lubrication, Leeds, UK, September 1974; pp. 37–41.
4. Booser, E.R.; Ryna, F.D. Linkinhoker, Maximum Temperature for Hydrodynamic Bearings under Steady Load. *Lubr. Eng.* **1970**, *26*, 226–235.
5. Khonsari, M.M.; Booser, E.R. In *Applied Tribology: Bearing Design and Lubrication*, 2nd ed.; John Wiley & Sons: West Sussex, UK, 2008.
6. Khonsari, M.M.; Wang, S.H. On the Maximum Temperature in Double-Layered Journal Bearings. *ASME J. Tribol.* **1991**, *113*, 464–469.
7. Khonsari, M.M.; Wang, S. On the Role of Particulate Contamination in Scuffing Failure. *Wear* **1990**, *137*, 51–62.
8. Khonsari, M.M.; Jang, J.Y.; Fillon, M. On the Generalization of Thermohydrodynamic Analyses for Journal Bearings. *ASME J. Tribol.* **1996**, *118*, 571–579.
9. Jang, J.Y.; Khonsari, M.M. Design of Bearings on the Basis of Thermohydrodynamic Analysis. *J. Eng. Tribol.* **2004**, *218*, 355–363.
10. Dyson, A.; Wilson, A.R. Film Thickness in Elastohydrodynamic Lubrication by Silicone Fluids. *J. Multi Body Dyn.* **1966**, *180*, 97–105.
11. Khonsari, M.M.; Beaman, J.J. Thermohydrodynamic Analysis of Laminar Incompressible Journal Bearing. *ASLE Trans.* **1986**, *29*, 141–150.
12. Khonsari, M.M.; Wang, S.H. On the Fluid-Solid Interaction in Reference to Thermoelastohydrodynamic Analysis of Journal Bearings. *ASME J. Tribol.* **1991**, *113*, 398–404.

13. Keogh, P.; Gomiciaga, R.; Khonsari, M.M. CFD Based Design Techniques for Thermal Prediction in a Generic Two-axial Groove Hydrodynamic Journal Bearing. *ASME J. Tribol.* **1997**, *119*, 426–436.
14. Jang, J.Y.; Khonsari, M.M. Performance Analysis of Grease-Lubricated Journal Bearings Including Thermal Effects. *ASME J. Tribol.* **1997**, *119*, 859–868.
15. Jang, J.Y.; Khonsari, M.M. Thermohydrodynamic Analysis of Journal Bearings Lubricated with Multigrade Oils. *Comput. Model. Eng. Sci.* **2002**, *3*, 455–464.
16. Dowson, D.; Hudson, J.D.; Hunter, B.; March, C.N. An Experimental Investigation of the Thermal Equilibrium of Steadily Loaded Journal Bearings. *Proc. Inst. Mech. Eng.* **1966**, *131*, 70–80.
17. Mitsui, J. A Study of Thermohydrodynamic Lubrication in a Circular Journal Bearing. *Tribol. Int.* **1987**, *20*, 331–341.
18. Ferron, J.; Frene, J.; Boncompain, R. A Study of Thermohydrodynamic Performance of a Plain Journal Bearing Comparison between Theory and Experiments. *ASME J. Tribol.* **1983**, *105*, 422–428.
19. Jang, J.Y.; Khonsari, M.M. Thermohydrodynamic Design Charts for Slider Bearings. *ASME J. Tribol.* **1997**, *119*, 733–740.
20. Fillon, M.; Khonsari, M.M. Thermohydrodynamic Design Charts for Tilting-Pad Journal Bearings. *ASME J. Tribol.* **1996**, *118*, 232–238.
21. Allmaier, H.; Priestner, C.; Reich, F.M.; Priebsch, H.H.; Novotny-Farkas, F. Predicting Friction Reliably and Accurately in Journal Bearings—Extending the EHD Simulation Model to TEHD. *Tribol. Int.* **2013**, *58*, 20–28.
22. Jang, J.Y.; Khonsari, M.M. On the Behavior of Misaligned Journal Bearings Based on Mass-Conservative Thermohydrodynamic Analysis. *ASME J. Tribol.* **2010**, *132*, 011702.
23. Medwell, J.O.; Gethin, D.T. Synthesis of Thermal Effects in Misaligned Hydrodynamic Journal Bearings. *Int. J. Num. Method Fluids* **1986**, *6*, 445–458.
24. Boncompain, R.; Fillon, M.; Frene, J. Analysis of Thermal Effects in Hydrodynamic Bearings. *ASME J. Tribol.* **1986**, *108*, 219–224.
25. Bouyer, J.; Fillon, M. Improvement of the THD Performance of a Misaligned Plain Journal Bearing. *ASME J. Tribol.* **2003**, *125*, 334–342.
26. Kucinschi, B.; Fillon, M.; Pascovici, M.; Frene, J. A Transient Thermoelastohydrodynamic Study of Steadily Loaded Plain Journal Bearings using Finite Element Method Analysis. *ASME J. Tribol.* **2000**, *122*, 219–226.
27. Monmousseau, P.; Fillon, M. Analysis of Static and Dynamic Misaligned Tilting-Pad Journal Bearings. *Proc. Inst. Mech. Eng.* **1999**, *213*, 253–261.
28. El-Butch, A.M.; Ashour, N.M. Transient Analysis of Misaligned Elastic Tilting-Pad Journal Bearing. *Tribol. Int.* **2005**, *38*, 41–48.
29. Thomsen, K.; Klit, P. Improvement of Journal Bearing Operation at Heavy Misalignment Using Bearing Flexibility and Compliant Liners. *J. Eng. Tribol.* **2012**, *226*, 651–660.
30. Tieu, A.K; Qiu, Z.L. Stability of Finite Journal Bearings—From Linear and Nonlinear Bearing Forces. *STLE Tribol. Trans.* **1995**, *38*, 627–635.

31. Banwait, S.S.; Chandrawat, H.N.; Adithan, M. Thermohydrodynamic Analysis of Misaligned Plain Journal Bearing. In Proceedings of the 1st Asia International Conference on Tribology, Beijing, China, 12–15 October, 1998; pp. 35–40.
32. Safar, Z.S. Energy Loss due to Misalignment of the Journal Bearings. *Tribol. Int.* **1984**, *17*, 107–109.
33. McKee, S.A.; McKee, T.R. Pressure Distribution in Oil Films of Journal Bearings. *Trans. ASME* **1932**, *5*, 149–165.
34. Cowlin, F.J. The Lubrication of Steam-Turbine Driven Electric Generator. *J. Eng. Tribol.* **1940**, *143*, 83–100.
35. Pigott, R.J.S. Bearings and Lubrication—Bearing Troubles Traceable to Design Can Be Avoided by Engineering Study. *Mech. Eng.* **1942**, *64*, 259–269.
36. DuBois, G.B.; Mabie, H.H.; Ocvirk, F.W. Experimental Investigation of Oil Film Pressure Distribution for Misaligned Plain Bearings. National Advisory Committee for Aeronautics: Kitty Hawk, NC, USA, 1951.
37. DuBois, G.B.; Ocvirk, F.W.; Wehe, R.L. Properties of Misaligned Journal Bearings. *ASME J. Basic Eng.* **1957**, *79*, 1205–1212.
38. Ausman, J.S. Torque produced by Misalignment of Hydrodynamic Gas-Lubricated Journal Bearings. *ASME J. Fluids Eng.* **1960**, *82*, 335–340.
39. Galletti-Manacorda, L.; Capriz, G. Torque Produced by Misalignment in Short Lubricated Bearings. *ASME J. Fluids Eng.* **1965**, *87*, 847–849.
40. Rice, J.R. Misalignment Torques of Hydrodynamic Gas-Lubricated Journal Bearings. *ASME J. Fluids Eng.* **1965**, *87*, 193–198.
41. Smalley, A.J.; McCallion, H. The Effect of Journal Misalignment on the Performance of a Journal Bearing under Steady Running Conditions. *Proc. Inst. Mech. Eng.* **1966**, *181*, 45–54.
42. Ocvirk, F.W. Measured Oil Film Distribution in Misaligned Plain Bearings. *J. Lubr. Eng.* **1954**, *10*, 262–265.
43. Stockley, J.R.; Donaldson, R.R. Misalignment Effects in 180° Partial Journal Bearings. *ASLE Trans.* **1969**, *12*, 216–226.
44. Pinkus, O.; Bupara, S.S. Analysis of Misaligned Grooved Journal Bearings. *ASME J. Tribol.* **1979**, *101*, 503–509.
45. Mokhtar, M.O.A.; Rahman, M.A.A.; Safar, Z.S. An Analysis of Misaligned Journal Bearings with Axial and Spiral Feedings. *Wear* **1983**, *85*, 331–337.
46. Safar, Z.S.; Mokhtar, M.O.A.; Peekent, H.J. Thermal Characteristics of Misaligned Finite Journal Bearings. *Tribol. Int.* **1985**, *18*, 13–16.
47. Ikeuchi, K.; Katsuse, S.; Hamamura, K.; Mori, H. Effects of Misalignment in Full Journal Bearing with Circumferential Oil Groove. In Proceedings of the JSLE International Tribology Conference, Tokyo, Japan, 8–10 July, 1985; pp. 67–72.
48. Wang, Y.; Wang, Q.J.; Lin, C. A Mixed-EHL Analysis of Effects of Misalignment and Elastic Deformations on the Performance of a Coupled Journal-Thrust System. *Tribol. Int.* **2006**, *39*, 281–289.
49. Howard, S.A. Misalignment in Gas Foil Journal Bearings: An Experimental Study. *ASME J. Eng. Gas Turb. Power* **2009**, *131*, doi:10.1115/1.2966392.

50. Vijayaraghavan, D.; Keith, T.G. Analysis of a Finite Grooved Misaligned Journal Bearing Considering Cavitation and Starvation Effects. *ASME J. Tribol.* **1990**, *112*, 60–67.
51. Pierre, I.; Bouyer, J.; Fillon, M. Thermohydrodynamic Behavior of Misaligned Plain Journal Bearings: Theoretical and Experimental Approaches. *STLE Tribol. Trans.* **2004**, *47*, 594–604.
52. Singh, D.V.; Sinhasan, R. Performance Characteristics of an Ungrooved Big-End Bearing with Misalignment. *STLE Tribol. Trans.* **1989**, *32*, 234–238.
53. Choy, F.K.; Braun, M.J.; Hu, Y. Analytical and Numerical Study of a Misaligned Cavitating Hydrodynamic Cryogenic Journal Bearing. *Tribol. Int.* **1992**, *25*, 3–15.
54. Nikolakopoulos, P.G.; Papadopoulos, C.A. Non-Linearities in Misaligned Journal Bearings. *Tribol. Int.* **1994**, *27*, 243–257.
55. Asanabe, S.; Akahoshi, M.; Asai, R. Theoretical and Experimental Investigation of Misaligned Journal Bearing Performance. *Tribol. Conv. Inst. Mech. Eng.* **1972**, 1–8.
56. Nicolas, D.; Frene, J. Tilting Torque Permissible in Plain Bearings. Theory, Experimental Results and Application to Machine Design. In Proceedings of the First European Tribology Congress, London, UK, 25–27 September, 1973; pp. 353–360.
57. Tieu, A.K.; Qiu, Z.L. Experimental Study of Freely Alignable Journal Bearings—Part I: Static Characteristics. *ASME J. Tribol.* **1996**, *118*, 498–502.
58. Qiu, Z.L.; Tieu, A.K. Experimental Study of Freely Alignable Journal Bearings—Part II: Dynamic Characteristics. *ASME J. Tribol.* **1996**, *118*, 503–508.
59. Arumugam, P.; Swarnamani, S.; Prabhu, B.S. Effects of Journal Misalignment on the Performance Characteristics of Three-lobe Bearings. *Wear* **1997**, *206*, 122–129.
60. Huber, M.; Strzelecki, S.; Steinhilper, W. Theoretical and Experimental Determination of the Performances of Misaligned and Statically Loaded Cylindrical Journal Bearings. *Austrib Congress* **1998**, 243–248.
61. Bouyer, J.; Fillon, M. An Experimental Analysis of Misalignment Effects on Hydrodynamic Plain Journal Bearing Performances. *ASME J. Tribol.* **2002**, *124*, 313–319.
62. Christensen, H. Stochastic Models for Hydrodynamic Lubrication of Rough Surfaces. *Proc. Inst. Mech. Eng.* **1969**, *184*, 1013–1026.
63. Patir, N.; Cheng, H.S. Average Flow Model for Determining Effects of Three-Dimensional Roughness on Partial Hydrodynamic Lubrication. *ASME J. Tribol.* **1978**, *100*, 12–17.
64. Patir, N.; Cheng, H.S. Application of Average Flow Model to Lubrication between Rough Sliding Surfaces. *ASME J. Tribol.* **1979**, *100*, 12–17.
65. Greenwood, J.A.; Williamson, J.B.P. Contact of nominally Flat Surfaces. *Proc. R. Soc. Lond.* **1966**, *A295*, 300–319.
66. Greenwood, J.A.; Tripp, J.H. The Contact of Two Nominally Flat Rough Surfaces. *Proc. Inst. Mech. Eng.* **1970–1971**, *185*, 625–633.
67. Kogut, L.; Etsion, I. Elastic-Plastic Contact Analysis of a Sphere and a Rigid Flat. *ASME J. Appl. Mech.* **2002**, *69*, 657–662.
68. Kogut, L.; Etsion, I. A Finite Element Based Elastic-Plastic Model for the Contact of Rough Surfaces. *Tribol. Trans.* **2003**, *46*, 383–390.
69. Abdel-Latif, L.A.; Mokhtar, M.O.A. Misalignment Effects on Hydrodynamically Lubricated Journal Bearings with Rough Surfaces. *Wear* **1988**, *128*, 225–237.

70. Guha, S.K. Analysis of Steady-state Characteristics of Misaligned Hydrodynamic Journal Bearings with Isotropic Roughness Effect. *Tribol. Int.* **2000**, *33*, 1–12.
71. Sharma, S.C.; Jain, S.C.; Nagaraju, T. Combined Influence of Journal Misalignment and Surface Roughness on the Performance of an Orifice Compensated Non-Recessed Hybrid Journal Bearing. *STLE Tribol. Trans.* **2002**, *45*, 457–463.
72. Sun, J.; Deng, M.; Fu, Y.; Gui, C. Thermohydrodynamic Lubrication Analysis of Misaligned Plain Journal Bearing with Rough Surface. *ASME J. Tribol.* **2010**, *132*, doi:10.1115/1.4000515.
73. Sun, J.; Zhu, X.; Zhang, L.; Wang, X.; Wang, C.; Wang, H.; Zhao, X. Effect of Surface Roughness, Viscosity-Pressure Relationship and Elastic Deformation on Lubrication Performance of Misaligned Journal Bearings. *Ind. Lubr. Tribol.* **2014**, *66*, 337–345.
74. Sun, J.; Gui, C. Hydrodynamic Lubrication Analysis of Journal Bearing Considering Misalignment Caused by Shaft Deformation. *Tribol. Int.* **2004**, *37*, 841–848.
75. Sun, J.; Gui, C.; Li, Z. An Experimental Study of Journal Bearing Lubrication Effected by Journal Misalignment as a Result of Shaft Deformation under Load. *ASME J. Tribol.* **2005**, *127*, 813–819.
76. Sun, J.; Gui, C.; Li, Z.; Li, Z. Influence of Journal Misalignment Caused by Shaft Deformation under Rotational Load on Performance of Journal Bearing. *J. Eng. Tribol.* **2005**, *219*, 275–283.
77. Ebrat, O.; Mourelatos, Z.P.; Vlahopoulos, N.; Vaidyanathan, K. Calculation of Journal Bearing Dynamic Characteristics Including Journal Misalignment and Bearing Structural Deformation. *STLE Tribol. Trans.* **2004**, *47*, 94–102.
78. Allmaier, H.; Priestner, C.; Reich, F.M.; Priebsch, H.H.; Forstner, C.; Novotny-Farkas, F. Predicting Friction Reliably and Accurately in Journal Bearings—The Importance of Extensive Oil-Models. *Tribol. Int.* **2012**, *48*, 93–101.
79. Jang, J.Y.; Khonsari, M.M.; Bair, S. On the Elastohydrodynamic Analysis of Shear-Thinning Fluids. *Proc. R. Soc. Lond. Ser. A* **2007**, *463*, 3271–3290.
80. Kumar, P.; Khonsari, M.M. Combined Effects of Shear Thinning and Viscous Heating on EHL Characteristics of Rolling/Sliding Line Contacts. *ASME J. Tribol.* **2008**, *130*, doi:10.1115/1.2959111.
81. Kumar, P.; Khonsari, M.M.; Bair, S. Full EHL Simulations using the Actual Ree-Eyring Model for Shear Thinning Lubricants. *ASME J. Tribol.* **2009**, *131*, doi:10.1115/1.3002328.
82. Kumar, P.; Khonsari, M.M. Traction in EHL Line Contacts Using Free Volume Pressure-Viscosity Relationship with Thermal and Shear Thinning Effects. *ASME J. Tribol.* **2009**, *131*, doi:10.1115/1.3002331.
83. Mokhtar, M.O.A.; Safar, Z.S.; Abd-Ei-Rahman, M.A.M. An Adiabatic Solution of Misalignment Journal Bearings. *ASME J. Tribol.* **1985**, *107*, 263–267.
84. Pierre, I.; Fillon, M.; Bouyer, J. Thermohydrodynamic Study of Misaligned Plain Journal Bearings—Comparison between Experimental Data and Theoretical Results. *Int. J. Appl. Mech. Eng.* **2002**, *7*, 949–960.
85. He, Z.P.; Xie, W.; Zhang, J.H.; Xie, W.S.; Li, Z.Y.; Zhang, G.C. Misalignment Analysis of Journal Bearing Influenced by Asymmetric Deflection, Based on a Simple Stepped Shaft Model. *J. Zheijang Univ. Sci. A Appl. Phys. Eng.* **2012**, *13*, 647–664.

86. Khonsari, M.; Kim, H. On Thermally Induced Seizure in Journal Bearings. *ASME J. Tribol.* **1989**, *111*, 661–667.
87. Wang, Q. Seizure Failure of Journal-Bearing Conformal Contacts. *Wear* **1997**, *210*, 8–16.
88. Dufrane, K.F.; Kannel, J.W. Thermally Induced Seizure of Journal Bearings. *ASME J. Tribol.* **1989**, *111*, 288–292.
89. Hazlett, T.L.; Khonsari, M.M. Finite Element Model of Journal Bearings Undergoing Rapid Thermally Induced Seizure. *Tribol. Int.* **1992**, *25*, 177–182.
90. Hazlett, T.L.; Khonsari, M.M. Thermoelastic Behaviors of Journal Bearings Undergoing Seizure. *Tribol. Int.* **1992**, *25*, 183–187.
91. Krishivasan, R.; Khonsari, M.M. Thermally Induced Seizure in Journal Bearings during Start up and Transient Flow Disturbance. *ASME J. Tribol.* **2003**, *125*, 833–841.
92. Krishivasan, R.; Khonsari, M.M. Thermomechanical Analysis of Oscillatory Pin-Bushing Performance. *Rev. Eur. Élém. Finis* **2005**, *14*, 255–269.
93. Pascovici, M.; Khonsari, M.M.; Jang, J.Y. On the Modeling of a Thermomechanical Seizure. *ASME J. Tribol.* **1995**, *117*, 744–747.
94. Jang, J.Y.; Khonsari, M.M.; Pascovici, M. Thermohydrodynamic Seizure: Experimental and Theoretical Evaluation. *ASME J. Tribol.* **1998**, *120*, 8–15.
95. Jang, J.Y.; Khonsari, M.M.; Pascovici, M. Modeling Aspect of a Rate-controlled Seizure in an Unloaded Journal Bearing. *STLE Tribol. Trans.* **1998**, *41*, 481–488.
96. Buckholz, R.H.; Lin, J.F. The Effect of Journal Bearing Misalignment on Load and Cavitation for non-Newtonian Lubricants. *ASME J. Tribol.* **1986**, *108*, 645–654.
97. Jang, J.Y.; Chang, C.C. Adiabatic Solution for a Misaligned Journal Bearing with Non-Newtonian Lubricants. *Tribol. Int.* **1987**, *20*, 267–275.
98. Boucherit, H.; Lahmar, M.; Bou-Said, B. Misalignment Effects on Steady-State and Dynamic Behavior of Compliant Journal Bearings Lubricated with Couple Stress Fluids. *Lubr. Sci.* **2008**, *20*, 241–268.
99. Osman, T.A. Misalignment Effect on the Static Characteristics of Magnetized Journal Bearing Lubricated with Ferrofluid. *Tribol. Lett.* **2001**, *11*, 195–203.
100. Das, S.; Guha, S.K.; Chattopadhyay, A.K. On the Steady-State Performance of Misaligned Hydrodynamic Journal Bearings Lubricated with Micropolar Fluids. *Tribol. Int.* **2002**, *35*, 201–210.
101. Safar, Z.S.; Riad, M.S.M. Prediction of the Coefficient of Friction of a Misaligned Turbulent Flow Journal Bearings. *Tribol. Int.* **1988**, *21*, 15–19.
102. Safar, Z.S.; El-Kotb, M.M.; Mokhtar, D.M. Analysis of Misaligned Journal Bearings Operating in Turbulent Regime. *ASME J. Tribol.* **1989**, *111*, 215–219.
103. Shenoy, S.B.; Pai, R. Theoretical Investigations on the Performance of an Externally Adjustable Fluid-film Bearing Including Misalignment and Turbulence Effects. *Tribol. Int.* **2009**, *42*, 1088–1100.
104. Abass, B.A.; Sahib, M.M. Effect of Bearing Compliance on Thermo-hydrodynamic Lubrication of High Speed Misaligned Journal Bearing Lubricated with Bubbly Oil. *Ind. Eng. Lett.* **2013**, *3*, 48–60.

105. Ram, N.; Sharma, S.C. Influence of Wear on the Performance of Hole-Entry Hybrid Misaligned Journal Bearing in Turbulent Regime. *Ind. Lubr. Tribol.* **2014**, *66*, 509–519.
106. Xu, G.; Zhou, J.; Geng, H.; Lu, M.; Yang, L.; Yu, L. Research on the Static and Dynamic Characteristics of Misaligned Journal Bearing Considering the Turbulent and Thermo-hydrodynamic Effects. *J. Tribol.* **2015**, *137*, doi:10.1115/1.4029333.
107. Goenka, P.K. Dynamically Loaded Journal Bearings: Finite Element Method Analysis. *ASME J. Tribol.* **1984**, *106*, 429–437.
108. Lahmar, M.; Frihi, D.; Nicolas, D. The Effect of Misalignment on Performance Characteristics of Engine Main Crankshaft Bearings. *Eur. J. Mech. A* **2002**, *21*, 703–714.
109. Booker, J.F. Dynamically Loaded Journal Bearings: Numerical Application of the Mobility Method. *ASME J. Tribol.* **1971**, *93*, 168–174.
110. Boedo, S.; Booker, J.F. Classic Bearing Misalignment and Edge Loading: A Numerical Study of Limiting Cases. *ASME J. Tribol.* **2004**, *126*, 535–541.
111. Li, Q.; Liu, S.L.; Pan, X.H.; Zheng, S.Y. A New Method for Studying the 3D Transient Flow of Misaligned Journal Bearings in Flexible Rotor-Bearing System. *J. Zhejiang Univ. Sci. A Appl. Phys. Eng.* **2012**, *13*, 293–310.
112. Landheer, D.; Faessen, J.P.M.; de Gee, A.W.J. A Transition Diagram for Plain Journal Bearings. *Tribol. Trans.* **1990**, *33*, 418–424.
113. Ligterink, D.J.; de Gee, A.W.J. Measurements of Wear in Radial Journal Bearings. *Tribotest J.* **1996**, *3*, 45–54.
114. Lu, X.; Khonsari, M.M. On the Lift-off Speed in Journal Bearings. *Tribol. Lett.* **2005**, *20*, 299–305.
115. Lu, X.; Khonsari, M.M.; Gelnick, E.R.M. The Stribeck Curve: Experimental Results and Theoretical Prediction. *ASME J. Tribol.* **2006**, *128*, 789–794.
116. Akbarzadeh, S.; Khonsari, M.M. Prediction of Steady State Adhesive Wear in Spur Gears Using the EHL Load Sharing Concept. *ASME J. Tribol.* **2009**, *131*, doi:10.1115/1.3075859.
117. Akbarzadeh, S.; Khonsari, M.M. Performance of Spur Gears Considering Surface Roughness and Shear Thinning Lubricant. *ASME J. Tribol.* **2008**, *130*, doi:10.1115/1.2805431.
118. Priestner, C.; Allmaier, H.; Priebsch, H.H.; Forstner, C. Refined Simulation of Friction Power Loss in Crank Shaft Slider Bearings Considering Wear in the Mixed Lubrication Regime. *Tribol. Int.* **2012**, *46*, 200–207.
119. Nikolakopoulos, P.G.; Papadopoulos, C.A. A Study of Friction in Worn Misaligned Journal Bearings under Severe Hydrodynamic Lubrication. *Tribol. Int.* **2008**, *41*, 461–472.
120. Ram, N.; Sharma, S. A Study of Misaligned Hole-Entry Worn Journal Bearing Operating in Turbulent Regime. *Ind. Lubr. Tribol.* **2013**, *65*, 108–118.
121. Andres, L.S. The Effect of Journal Misalignment on the Operation of a Turbulent Flow Hydrostatic Bearing. *ASME J. Tribol.* **1993**, *115*, 355–363.
122. Jain, S.C.; Sharma, S.C.; Nagaraju, T. Misaligned Journal Effects in Liquid Hydrostatic Non-Recessed Journal Bearings. *Wear* **1997**, *210*, 67–75.
123. Elsharkawy, A.A. Effects of Misalignment on the Performance of Flexible Porous Journal Bearings. *STLE Tribol. Trans.* **2003**, *46*, 119–127.

124. Gulwadi, S.D.; Shrimpling, G. Journal Bearing Analysis in Engines Using Simulation Techniques. In *Society of Automotives Engineers Technical Paper Series*; Society of Automotive Engineers: Warrendale, PA, USA, 2003.
125. Strzelecki, S. Operating Characteristics of Heavy Loaded Cylindrical Journal Bearing with Variable Axial Profile. *Mater. Res.* **2005**, *8*, 481–486.