Current challenges and frontiers for the EHD simulation of journal bearings: a review

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Abstract

Elastohydrodynamic (EHD)-simulation is a widely applied simulation technique that is used in a very diverse field of applications ranging from the study of vibroacoustics to the calculation of friction power losses in lubricated contacts. In particular, but not limited to, the automotive industry, technical advances and new requirements put current EHD simulation methodology under test. Ongoing trends like downsizing, downspeeding, start-stop and the continuing demand for increasing fuel efficiency impose new demands and challenges also on the simulation methodology. Increasing computational capabilities enable new simulation opportunities on the other hand.

In the following, an overview is given on the current state of the art and today's challenges for the elastohydrodynamic simulation of journal bearings and their wide range of applications from highly loaded main bearings supporting the crank shaft in the ICE to high speed turbocharger bearings.

The topics addressed in this work span a wide range from mixed lubrication, low viscosity lubricants, polymer coatings, microstructured surfaces to the simulation of the thermal behavior and the simulation of hydraulically coupled journal bearings as they are commonly present in internal combustion engines.

Introduction

The review concentrates on conformal contacts, which are characterized by equally signed curvature of the contact surfaces (differently to non-conformal contacts). Oil lubricated conformal contacts can be separated into hydrostatic/hydrodynamic (HD), elastohydrodynamic (EHD) and thermoelastohydrodynamic (TEHD) lubrication models. HD contacts consider large contact areas and therefore low lubricant pressure. EHD and TEHD contacts have to represent locally high pressures up to 3 GPa as well as large material load.

Due to the large number of available works, the paper excludes simplified early works as for instance those, which combine two-dimensional solid models with one-dimensional (short bearing) fluid film models. Included are, however, simple approaches, which are derived from more advanced EHD models. In the following, the terms journal bearing and radial slider bearing are used interchangeably.

The first part of this work discusses today's and tomorrow's applications for modern journal bearing simulation methodology. Mainly, these are new modes of engine operation and constructions, new coatings and surface textures but also new classes of lubricants with complex properties. The second part maps these application demands on requirements for journal bearing simulation tools.

Requirements for journal bearings

New engine operation modes and constructions

One of the most widely used possibilities to improve fuel economy especially for inner city driving, is the employment of start-stop systems (Bishop07, Silva09). By switching off the engine instead of running it inefficiently at idle during waiting times at crossings etc., significant fuel savings can be obtained (Silva09, Fonseca11). More recently, this procedure was even extended to shut down the engine also during driving for very low loads, which is commonly called engine coasting. These systems have quickly become widely available, but despite their apparent simplicity the repeated stopping and starting of the engine still can be a source of problems. The reason for this is that during this stopping and starting procedure the journal bearings experience mixed lubrication as their lubrication follows the entire Stribeck-curve (Sander16). While during stopping the journal bearings are still well lubricated, lubricant continuously leaves the journal bearing during the stopped engine period until the poorly lubricated journal bearing has to endure the next starting process (Mokhtar77). It is commonly understood that journal bearings operate almost exclusively in the purely full film lubricated regime. Even if the possibility of metal-to-metal contact is taken into account in the simulation by employing a contact model, the accuracy of the used contact model was of less importance to date. As only little mixed lubrication was allowed to occur under normal operating conditions, the repeated stopping and starting requires a sophisticated description of metal-to-metal contact. An accurate description of metal-to-metal contact is not only needed to predict friction properly, but – even more importantly – to predict journal bearing wear realistically (Sander15-2, Gudin13).

A further widely used possibility to improve fuel economy as the result of more efficient thermodynamics is the cylinder de-activation (CDT) technology, where commonly half of the available cylinders are disabled for low load operation. Further benefits are the higher exhaust temperatures under partial loading, which yields an improved aftertreatment efficiency for Diesel engines by supporting the three-way catalyst technology (Wilcutts13). Friction power losses but also shaft bending and misalignment are topics of bearing simulation for this CDT-technology. In addition to gather a basic understanding of above effects, this new mode of engine operation has inherent requirements that need to be considered in the engine design (Mohmamadpour14, Mohmamadpour14-2, Shahmohamadi15).

Today, turbocharging the internal combustion engines of passenger cars (Porzig14) is one of the most promising technologies to increase the power output, improve the engine efficiency and keep the emissions within the legal regulatory limits. But also engines of light duty trucks, city buses and even tractors and harvesters use turbochargers (Smolik15). The rotors of turbochargers are typically supported by two journal bearings of the floating ring type, which consist of a short cylindrical bushing placed between the shaft and the turbocharger housing forming two thin fluid films in series. The two thin fluid films may interact via radial drillings in the bushing. This bearing system is lubricated with conventional internal combustion engine oil and has proven to be robust, reliable and very cost effective due to its simplicity and ease of manufacturing. Different designs as for instance semi-floating ring bearings, where the rotation of the ring is inhibited by an axial pin, and full-floating bearings are in practical use. Turbochargers operate at high speeds on the order of hundreds of thousands revolutions per minute and tend to be prone to friction, fatigue and instability (Porzig14).

New classes of coatings and surface textures

To both increase durability and reduce the friction losses in journal bearings new techniques become increasingly popular in the manufacturing of journal bearings. While still two and three layer journal bearings dominate the automotive industry, new development trends bring polymer coatings, diamond like carbon (DLC) coatings and refined surface structuring techniques. These bring classical EHD-simulation methodology to its limits and poses new challenges.

Recently it was shown that DLC coatings also have a potential for friction reduction for the journal bearings of the crank train (Bobzin15). DLC-coatings are already commonly used in engines today, like for piston pins or cam tappets, which are highly loaded contacts operating in dominantly mixed lubrication. In the journal bearings of the crank train, dominantly hydrodynamic lubrication is present, which makes this finding quite remarkable. Aside from potential benefits of DLC coatings under mixed lubrication in the journal bearings, apparently under elastohydrodynamical lubrication conditions the DLC-coating provides a thermally insulating layer that leads to higher lubricant temperatures in the contact. As lubricant viscosity decreases with temperature, the friction losses in the oil film are reduced as a consequence. State of the art elastohydrodynamic simulation methodology utilizes the bearing back temperature to predict with great accuracy the friction power losses in the lubricant film (Allmaier11, Sander15). To extend this approach to these DLC-coatings with their distinct thermally insulating behavior will be a new challenge, if the same accuracy of the simulation results is to be expected.

The running-in and wear behavior of standard journal bearings is quite well understood and can also be described by simulation very closely (Allmaier15, Sander15-2). Polymer coatings, however, are increasingly wide spread as these can offer better durability in start-stop applications (George12, Ferreira14). As these are non-metallic materials, their behavior in terms of wear and plastic deformation is distinctly different and requires new approaches to describe them realistically in the simulation (Offner15, Gudin13). Despite the use of polymer materials in mass produced engines, the very scarce number of publications shows that there still needs to be a lot of work done to better understand this new class of materials.

However, surface coatings are not the only way to improve the properties of journal bearings. Surface texturing, that is the intentional modification of the journal bearing surface (Costa15) with regular patterns (dimples, microgrooves) was also shown to have the potential to increase the load carrying capacity. Such surface texturing methods themselves are not new and are, for example, widely used in other parts of the internal combustion engine, namely as honing of the cylinder liners. For cylinder liners, the main purpose of this particular surface texture is to retain lubricant on an otherwise poorly lubricated surface. The situation for journal bearings is very different in this regard as these are by design supplied with sufficient lubricant. It was shown experimentally that surface textures can affect the tribological properties of the journal bearing (Lu07, Henry15) as well as the damping properties (Dadouche11) and the compatibility of the journal bearing with contaminants (Dadouche16). The theoretical understanding of the underlying effects of surface textures is still quite limited (Gropper16). So far it is commonly accepted that cavitation (Braun10) plays a crucial role for this effect (Zhang12, Qiu11) as well that surfaces textures have an influence on local microhydrodynamics (Gropper16). The consideration of such textures in the EHD-simulation poses new challenges: a direct approach including these structures requires a correspondingly fine hydrodynamic mesh size as these surface structures are typically only a few millimeters in size. Therefore, the numerical demand becomes the limiting factor (Woloszynski15). For indirect methods that describe the influence of these surface structures only in a statistical sense, e.g. by flow factors, entirely new approaches are required. Both, the optimum geometry as well as the optimum surface area for such surface textures (partial vs. full texturing) and their interaction with other journal bearing properties like acoustical damping (Meng15) are very active research areas where still a lot of questions need to be answered.

New ultralow viscosity lubricants

The usage of low viscosity lubricants in the automotive sector is certainly not new. Increasingly strict legislation forces the automotive industry to increase the engine efficiency and few ways to achieve this are as economic as the usage of a lower viscosity lubricant, which is considered to be a very effective measure for the whole industry (Holmberg14). However, the current trend goes to lubricants with drastically reduced viscosities and new standards had to be defined accordingly (Covitch10). With the new SAE standards 0W16, 0W12, 0W10 (Covitch10) and even lower classes being targeted, pure hydrodynamic lubrication conditions will more and more diminish and be replaced by mixed lubrication near the EHD minimum (Sander15, Ligier15).

It is well known that modern multi-grade lubricants show a non-Newtonian behavior. It is highly relevant in practical applications and its assessment was, therefore, standardized for one relevant set of conditions which is defined as high temperature high shear viscosity (HTHS). It was shown that the HTHS-viscosity directly affects the mechanical efficiency of the engine in fleet tests (Macian14, Macian16), which demonstrates the necessity to realistically consider the non-Newtonian behavior of the lubricant in the simulation. There exists a further lubricant property counteracting the non-Newtonian behavior in journal bearings, which increases the lubricant viscosity locally under a load. It is called the piezoviscous effect and is present in both mono and multi-grade lubricants (Allmaier12).

Therefore, simulations assessing the lubrication performance of such contacts need to be very accurate and consider extensively the rheological properties of the lubricant. It was shown that both the consideration of the non-Newtonian behavior as well as of the piezoviscous effect in the EHD simulation yield accurate predictions of the lubrication conditions. Notably the friction power losses in journal bearings can be predicted within the measurement uncertainty of a dedicated journal bearing test rigs (Allmaier11, Sander15). With this powerful methodology, the lubrication performance of ultralow viscosity lubricants can be investigated in great detail for internal combustion engines and the onset of mixed lubrication can be predicted (Knauder15, Carden12, Carden13).

While all of this certainly appears promising, still a lot of work needs to be done in this area. Currently the required detailed lubricant data are not contained in the lubricant datasheets, as such measurements are not part of common standards. Therefore, such rheological data are only sparse. Also there exist an interdependency of the piezoviscous effect and the non-Newtonian behavior, which makes it difficult not only to measure them, but also to address these properties separate from each other in simulation. Currently, common approaches use exponential functions like e.g. the Barus-equation (Barus93) and the Cross-equation (Cross65) to consider these effects in simulation separately and independent of the lubricant temperature. While these are useful approximations of the reality and considerably increase the quality of the results, the true lubricant behavior is still considerably more complex (Bair01, Bair02, Bair07).

Improving the thermal behavior

The lubricant viscosity is central to the lubrication of journal bearings; it limits the load carrying capacity and is directly related to the friction power losses. Despite all work done by the lubricant manufacturers during the last decades, modern lubricants still show a more than exponential dependence of the lubricant on temperature. Consequently, the lubricant viscosity is extremely large at cold temperatures, which causes very high friction and pumping power losses for the cold engine. The automotive OEMs are well aware of this issue and work continually on the improvement of the thermal management. However, in comparison to the coolant system the oil takes considerably longer to warm up (Roberts14, Addison15), so work needs to be done to improve the thermal behavior of journal bearings. In addition to constructive measures like to preheat the oil supplied to the journal bearings (Zammit12, Zammit12-2) or to reduce the oil flow (Honda14, Estupinan12) to increase the bearing temperature, also a more detailed understanding of the thermal behavior of journal bearings is needed. So far the published works on the thermal properties of journal bearings focused on stabilized thermal conditions (Piffeteau99, Wang02, Allmaier13, Lorenz15, Shahmohamadi15), but not on the transient warming-up process.

Also in turbochargers the bearing system is exposed to a difficult thermal environment. Besides the shear driven frictional losses within the fluid films acting as heat sources, the turbine wheel is also exposed to hot exhaust gas from the engine, which reaches temperatures of 600°C in modern Diesel engines and higher temperatures in modern Otto-cycle engines. This results in non-uniform temperature distributions in the shaft, the turbocharger housing and the bearing system itself, which is complicated to analyze because of the multiplicity of different heat flow paths (Pozig14).

Requirements for simulation tools

After this introduction to the recent application challenges, the consequent requirements for the journal bearing simulation methodology are discussed in detail.

There exist a number of review papers, which deal with EHD lubrication analysis of conformal contacts. Examples are Goenka92, Xu97, Tanaka97, Goodwin03 and finally Booker10. The last one, which is also the most recent one, will serve as starting point for the present work.

Booker10 lists in his review a number of basic topics, which are (a) mass-conservative cavitation; (b) thermal effects; (c) surface roughness effects; (d) fluid pressure-viscosity effects; (e) fluid non-Newtonian effects; (f) structural inertia and (g) experimental methods. From the current perspective all requirements (a)-(g) are still valid. However, above discussed applications imply the need to extend these requirements further by:

- The consideration of accurate structural properties is essential for the accuracy of the obtained results. Both structure details and its dynamic properties are relevant. In particular, dynamic properties are even more relevant in the case of high speed applications as for instance in turbochargers.

- A main influence of the performance of radial slider bearings originates in the amount of available lubricant and its distribution within the bearing (Offner13). Grooves and bores serve as oil supplies for a radial slider bearing, hence define transient boundary conditions. In ICEs, the bores of different radial slider bearings may be connected via drillings. Examples are drillings in a crankshaft, which connect big end bearings to main bearings. Another example are drillings in the floating bushing of a turbocharger, which connect the outer and the inner EHD contact. The task of these drillings is to transfer lubricant between connected bearings. Furthermore, due to relative motion of different boundary conditions within a bearing, these may overlap geometrically so that several neighboring bearings are interacting adaptively at certain times. This results in the requirement to consider the drillings within a hydraulic network approach with adaptive connectivity. Furthermore, the flow in the lines may be influenced by the motion of the embedding component.

- Strong coupling of structural dynamics and fluid dynamics (Arnold08) is mandatory. In case of weak coupling the physics is solved sequentially. Data exchange/update happens only between the time steps.

- Common approaches consider only laminar fluid properties in the Reynolds equation. However, in some cases this assumption is not valid. Examples are bearings with deep grooves or turbocharger applications. In these cases at least a simplified turbulence approach is required.

Based on above requirements the following text discusses different approaches, which are currently available in literature and industry. The discussion is subdivided into component modeling (structural dynamics, large rotational journal speeds, displacement representation), mixed lubricated contact modeling (simplified fluid dynamics, viscosity, density, cavitation, mixed friction asperity contacts and micro-hydrodynamics, turbulent flow conditions, thermal fluid representation, 3D fluid dynamics modeling) and coupling methods (Coupling between structure dynamics and fluid dynamics, hydraulic coupling between radial slider bearings).

COMPONENT MODELING

Structural dynamics

Typical example applications involve several components with many contacts between each other. Due to the complexity of each system, it is broken down into coupled sub-systems. Components like engine block, crankshaft, pistons, piston-pins, connecting rods, journals, bushings, housings, etc. have to be considered in case of an internal combustion engine.

There exist different ways to model a component. Depending on the application, a component may be rigid or flexible. For instance Kim10 and Ozdemir14 consider both journal and shell to be rigid. Andres12 and Andres04 model both the shaft and the floating rings as rigid components, considering radial gross motion.

A component may require the consideration of gross motions in addition to the flexibility. According to Schwertassek99, two basic strategies for handling the flexibility used in literature. The first strategy is not to split the overall displacement of a point situated on the flexible component into gross motion and elastic deformation. This strategy is called “Absolute Nodal Coordinate Formulation” and leads to non-linear systems for the elastic forces but linear inertia terms. Due to the large component models that have to be represented, this non-linearity is expensive from a CPU time point of view. Therefore, the second strategy is common – the “Floating Frame of Reference Formulation”. This strategy splits gross motions and deformations. A moved reference coordinate system is introduced, which moves in accordance with the gross motions of the elastic component. Elastic deformations are measured relative to the reference frame. Due to involved materials as steel, iron, aluminum etc. the assumption of linear elasticity is made. The derived equations can be applied to differently modeled components. Besides beam-mass models and structured models, also representations taken from FEM (which may have reduced degrees of freedom, also called condensed models), are common (Offner11).

In some works the gross motion is computed using a multi-body system (MBS) approach. Knoll10 uses a condensed FEM model to represent the flexible shaft. The gross motion is calculated by using Newton’s equations. Lang14 also considers local elastic surface deformations similar to Knoll10. Smolik15 and SmolikRendlova15 consider a combination of flexible and rigid components. Both the housing and the bushings are considered to be rigid whereas the shaft is flexible. Both the shaft and the bushings perform gross motions in angular but also in radial direction. Offner13, Offner13-2, Bukovnik07, Offner15 and Lorenz15 consider both the journal and the shell as flexible. The shell is modeled in FEM and the journal is represented as beam mass model. In Offner15 the authors additionally consider plastic deformation by using an additional contact surface layer.

Vetter14 considers a rotor model, which consists of a flexible shaft, modeled by FEM using Timoshenko beam theory. Compressor and turbine wheel as well as the floating rings are assumed to be rigid.

A dynamic strong coupling by integrating the elastic structures in parallel to hydrodynamic equations is most accurate even if CPU time consuming. However, a simplified static approach uses compliance matrices related to the lubrication gap, respectively, which are set up in a pre-processing step. The matrix is derived from FEM by loading the discrete nodes on the surface of the plain bearing one after another in radial direction and then calculating the elastic deformation for all nodes on this surface from the resulting single-force elastic deformation. Commercial FEM tools may be used for that purpose. With the aid of the compliance matrix calculated by this means it is possible to calculate the elastic deformations vector of the lubrication gap contour for a discrete hydrodynamic pressure vector. However, the static support of the component is essential in this case. Stefani04, Bobach08, Bartel10, Steinbatz10, Bartel12, Profito15 and Illner15 use a compliance matrix approach to model the flexibility of the shell. The journal is assumed to be rigid in these works. Gentre01 considers elastic and thermal deformations. Elastic deformations are obtained using a compliance matrix obtained from FEM. Thermal deformations are modeled using thermal matrices (Khonsari91) related to journal and shell, respectively.

Mohammadpour14 considers a simple two degree of freedom (DOF) system to compute bearing loads as the result of varying in-cycle piston kinematics and the combustion pressure in the discussed big end bearing application. The clearance gap is computed based on the lateral excursions of the journal center and a pressure dependent radial deflection of the bearing bushing is considered (Rahnejat00) assuming a static elliptic bore shape.

In contrast to many works which consider homogeneous material properties Krampl12 presents a method that considers also heterogeneous material properties. Stresses result because of asperity contact and hydrodynamic contact pressure, but also due to the heterogeneous material itself. In case of mixed lubrication adhesion and abrasion of the surfaces (see for instance Offner15) may result. Even if local stress peaks may occur, the usage of heterogeneous materials has been proven practically to be beneficial. With respect to sliding applications Schratt98 distinguishes four materials: (1) hard matrix with soft phases, (2) soft matrix and hard inclusions, (3) combinations of (1) and (2) and finally (4) compound structures (including hardened surfaces). Krampl11, Grün12 and Krampl12 show investigations of above variants. Based on microscope images as well as artificially generated stochastic and regular phase distributions, the method allows an investigation of local stress concentrations in materials intended for use in tribological applications.

Large rotational journal speeds

A very specific requirement for turbocharger simulation compared to common crank train simulations are the involved high rotational speeds. Turbochargers are typically operating in the speed interval 20 – 200 krpm or even higher. Consequently, relative speeds of the order of 50 m/s have to be dealt with in contrast to about 8m/s or less which are common for crank train journal bearings. Nominal speeds of 120 krpm (SmolikRendlova15) or 130 krpm (Smolik15) are usual. Andres04 and Andres12 consider rotational speeds up to 120/240 krpm, respectively. Smolik15, SmolikRendlova15 and Knoll10 investigate up to 140/200/240 krpm. Porzig14 and Vetter14 investigate up to 190/210 krpm, respectively. In particular the accurate representation of inertia properties in relation to mass properties of a shaft is relevant in case of large rotational speeds. In particular Smolik15 discusses the influence of the increased mass properties.

Displacement representation

To couple structural dynamics (for instance of a journal or a shell) with any kind of contact force computation (for instance EHD), it is necessary to know the component motions so that the corresponding forces can be computed based on these. Traditionally either a modal or a nodal approach is used for that purpose (Booker10). The modal (or mode based) approach is less widespread in the literature. According to this method the nodal displacements are computed by adopting a linear combination of particular mode shapes determined from the linear elastic solution of the journal bearing structure. On the other hand the nodal (or node based) approach is most predominant in publications. In this case the nodal displacements are directly determined from the dynamics equations of the components, see for instance Offner11, or from the direct relation between the nodal forces vector and the linear compliance matrix of the elastic structure.

For a coupled FEM model, typically only the degrees of freedom of nodes placed on the internal bearing surface are retained for the EHD solutions. Journals are typically represented as beam mass elements (as for instance in case of a Timoshenko or a Bernoulli approach). Thus the journal is assumed to be very stiff and its radial deformation is consequently assumed to be negligible in this case. However, in case of low radial stiffness, as for instance in case of hollow journals, the consideration of radial flexibility may be required (Offner13).

LUBRICATED CONTACTS MODELING

Simplified fluid dynamics

In this section a number of simplified approaches are discussed. Specific applications might require special boundary conditions and certain simplifications. Further, some methods are discussed which use at first an advanced Reynolds equation to consequently employ a parameterized, simplified approach.

Due to large computation times and for numerical stability reasons Vetter14 use a simplified approach based on the Reynolds equation. As the rotational speeds are very high in case of the discussed turbo charger application it is assumed that the cavitation zone of the film is equal to the one determined for a certain rotor position. Consequently, the Reynolds equation becomes linear and is split into two equations. The forces resulting from the two pressure distributions of the two separate oil films are superimposed in order to predict the fluid film force. Similarly, Mohammadpour14 introduces application specific assumptions to be able to integrate the Reynolds equation twice. Further, zero pressure boundary conditions are employed at the bearing boundaries so that a simplified pressure calculation formula is obtained. Another simplified approach can be found in Stefani04. The authors outline a 2D approach assuming the clearance height to be constant in axial direction and assume further a constant angular velocity of the journal and a constant time step size.

Illner15 outlines an analytical method to calculate the minimum film thickness and the transition speed in journal bearings taking into account elastic bearing deformations. The method requires elasticity factors, which are a priori derived by an advanced bearing simulation. Ebrat03 also uses a perturbation approach on the static Reynolds equation in order to calculate the bearing's dynamic characteristics in terms of stiffness and damping matrices.

Lubricant rheology

Despite many work in literature still describe the lubricant in journal bearings using a single value for viscosity, the rheological properties of automotive lubricants are considerably more complex. In addition to the well known stronger than exponential temperature dependence of the viscosity on temperature, also pressure and shear rates have a strong impact. The increase of viscosity under a load is commonly called the piezoviscous effect and is present in hydrocarbon based lubricants. For automotive lubricants which are a complicated and secret mixture of a hydrocarbon based basis oil with many different additives, also a distinct Non-Newtonian behavior can be observed. Simply put, the presence of a pressure and/or shear stress/rate can affect the lubricant viscosity in the contact significantly (Bair01, Bair02, Bair07).

A number of different models for describing viscosity are available and used in literature. The two works Smolik15 and SmolikRendlova15 use a constant viscosity and show that for turbocharger applications at least the consideration of a temperature dependent viscosity is required. Beyond the assumption of a constant viscosity (also assumed for instance in Steinbatz10, Ebrat04, Kim10, Offner11, Krampl12), other approaches are the temperature and pressure dependent model by Houpert85, the temperature dependent model by Vogel, the pressure dependent models by Barus (Barus93) and Roelands (Roelands66) and the shear rate description of Cross (Cross65). Allmaier11, Allmaier12, Allmaier15, Sander15, Sander15-2, and Knauder15 consider both the piezoviscous effect as well as the non-Newtonian behaviour for multi-grade lubricants. Allmaier12, Sander15, Bukovnik07 and Lorenz15 use a combined Vogel-Barus and a combined Vogel-Barus-Cross approach and compare the proposed approaches.

Hagemann13 and Porzig14 present works considering temperature dependent viscosity and Bartel10 uses the approach by Barus (Barus93). Illner15 considers pressure dependent viscosity and Carden12 consider temperature dependent viscosity and explicitly do not consider shear thinning. Mohammadpour14 consider temperature dependent viscosity according to an approach, discussed in Rahnejat98.

Furthermore, combinations of above methods are used. For instance Andres04 and Andres12 consider viscosity as a function of temperature and shear rate and combine the Vogel and the Cross approach. Grente01 consider a combination of the temperature dependent viscosity approach discussed in Rahnejat98 and a pressure dependent asymptotic viscosity term. Bode considers a pressure and temperature dependent viscosity model, which is based on the fluid's density (Bode89). Besides the Barus approach (Barus93), Bartel10 uses also the one proposed by Bode89 in some cases. Bosbach08 also uses the model according to Bode89.

Density

Different models exist to describe the density of an oil film. Besides the simplified approach of assuming constant density, works exist that consider its dependency on the pressure as well as on the temperature of the fluid (Allmaier13). Most common are the models by Dowson and Higginson (Dowson59). A further pressure dependent approach has been proposed by Cheng94. A temperature and pressure dependent approach has been presented by Bode89.

Most of the reviewed works do not consider density at all. Some consider it as constant (e. g. Bartel12, Grente01, Bukovnik07, Lorenz15). Hagemann13 considers pressure dependent density according to Cheng94. Illner15 also considers pressure dependent density. Bartel10 uses both the approach by Dowson59 but also the one proposed by Bode89 in some cases. Bosbach08 also uses the model according to Bode89.

Cavitation

Since the Reynolds equation describes a fully filled lubrication gap, it yields unphysical, negative film pressures in the area of the diverging clearance. As lubricants cannot transfer significant tensile stresses, cavitation occurs in these areas and, correspondingly, cavitation boundary conditions need to be considered. These boundary conditions must at least ensure that the oil film pressure does not fall below a defined cavitation pressure.

Several models that describe cavitation are available in literature. Basically these can be subdivided into two groups. The first group comprises models which do not ensure the continuity equation and, therefore, do not preserve mass. The clearance gap is assumed to be fully filled with oil. Examples are the Gümbel (Guembel25), the Reynolds (Reynolds27) and the half Sommerfeld (Sommerfeld04) boundary conditions. To distinguish between lubrication and cavitation domain the method by Murty (Murty74) is commonly used. The second group ensures mass preservation. Lubrication and cavitation areas are again distinguished and partial filling is considered in the cavitation areas. Examples are the model introduced by Jakobsson, Floberg and Olsson (Jakobsson57, Olsson65), which is commonly known as JFO model. For the purpose of an efficient numerical procedure Elrod developed a JFO based approach, which does not require solving the JFO boundary conditions (Elrod81). A further model is the exponential approach as suggested in Sahlin07. For more details see for instance Bobach08, Bartel10 and Krampl12.

In recent literature non-mass preserving approaches can still be found. For instance Ebrat04 considers the Reynolds cavitation approach and Ozdemir14 applies a half Sommerfeld boundary condition. Krampl12 investigates different cavitation models. In particular a number of non-mass preserving approaches are compared with the exponential approach of Sahlin07 and the importance of mass preserving approaches is shown. Furthermore, the exponential approach is proven to be a reasonable choice in particular when density is also considered. Bukovnik07, Offner11, Offner13, Offner13-2, Lorenz15, Offner15, Smolik15 and Profito15 consider the JFO model. Bartel12 and Porzig14 consider a mass conserving cavitation model based on JFO theory according to Elrod81. Stefani04 and Gentre01 consider Murty's algorithm stating no details on the used cavitation model. Bartel10 uses mass preserving models also without stating details on the used cavitation model. However, Kim10, Illner15, Mohammadpour14, Andres04 and Andres12 do not consider cavitation.

Mixed friction asperity contacts and micro-hydrodynamics

Mixed friction refers to a hydrodynamic journal bearing in which both liquid- and boundary friction occur simultaneously. Mixed friction begins when the minimum thickness of the separating lubricant film falls below a critical film thickness. While journal bearings are typically designed in a such a way that no mixed lubrication occurs under normal operating conditions, new modes of engine operation like stopp-start or other especially severe operating conditions might lead to the occurrence of mixed lubrication. Then the first micro-contacts between the surface roughnesses of the shaft and bearing shells occur. Bartel10 and Offner11-2 describe methods, where representative sections of real surfaces are determined using a discrete elastic plastic solid contact models. In particular the model proposed in Bartel10 calculates distributions of contact pressure and deformation building upon the elastic half space theory. Simplified elastic-plastic deformation is incorporated with the aid of a linear elastic ideal plastic material model, whereby the maximum contact pressures are limited to a defined plastic flow pressure. The method distinguishes between elastic and plastic deformations parts. With the aid of the contact model, integral solid contact pressures are derived from local distributions as a function of the mean deformed gap height between rough surfaces.

At insufficiently low relative motion of the two contacting component surfaces mixed lubrication may occur. The local film thickness between the gliding surfaces drops down to a magnitude in the range of the mean average surface roughness. The short wave length of the roughness peaks would require a high resolution of the calculation grid in order to describe such local effects in the hydrodynamic calculation. As a consequence, high computational costs would arise. Thus, the contribution of asperity contact loads to the surface deformation is separated into two parts: (1) local asperity deformation and (2) global distortion of the two contacting component surfaces. The latter, second part describes the compliance of the contacting bodies and is determined in the overall elastic displacement calculation. The former, first local asperity deformation part is related to the local film thickness between the surfaces and its topography as well as the flexibility of the contacting components. The asperity load is a function of surface roughness and material properties. For an assumed Gaussian distribution of asperities the load supported by the asperity tips can be expressed according to Greenwood70. Based on his approach, refined approaches were presented by Gohar08 and Teodorescu05.

If the film thicknesses are of comparable magnitude to the roughness of the contact surfaces, the influences from micro-hydrodynamic effects on the hydrodynamic pressure development needs to be considered. Based on the studies by Patir78 the Reynolds equation has been extended by flow factors. Bartel10 presents a numerical pre-calculation on basis of real surfaces. Furthermore, the work discusses also the incorporation of the pre-calculated flow factors in the Reynolds equation. Bosbach08 uses the same approach.

Offner11, Offner13, Offner13-2, Lang14, Offner15 and Lorenz15 consider flow factors according to Patir78 and a surface contact model according to Greenwood70. In Offner15 flow factors are derived from measured surface roughness data. Furthermore, a generic friction modeling approach for radial slider bearings is used, which can be applied to lubricated contact regimes. In addition to viscous friction, the approach considers also boundary friction. The parameterization of the friction model is done using surface material and surface roughness measurement data. Furthermore, fluid properties depending on the involved oil additives are considered.

Illner15 considers flow factors and solid contact pressure curves determined based on 3D surface measurement of the real plain bearings surfaces.

The works Bartel12, Smolik15 and SmolikRendlova15 also consider rough surfaces and flow factors. Carden12 presents a work, where different bearings of an internal combustion engine are investigated using different approaches. Besides main and big end bearings for which simplified approaches are applied, the authors use the Reynolds equation to describe the small end bearing. Asperity contacts are calculated according to Greenwood70. Perfect alignment of bearings and perfectly cylindrical bearings and journals are assumed in this work.

Mohammadpour14 and Ozdemir14 do not consider flow factors in the Reynolds equation but consider asperity contact according to Greenwood70.

Bukovnik07, Kim10, Ebrat04, Steinbatz10, Stefani04, Andres04 and Andres12 assume ideally smooth surfaces.

Turbulent flow conditions

Most of the works which are available in literature assume laminar flow conditions of the lubricant in the journal bearing; examples are Andres12 and Grente01. However, an important aspect in the analysis of high speed journal bearings - as for instance those in turbochargers - is turbulence, which influences heat transfer, dissipation and the dynamic characteristics of the bearing. Due to the numerical complexity of the problem, these effects are typically approximated by semi-empirical approaches. Two works, which consider turbulence, are the work by Hagemann13 and the work by Vetter14 that use the approach by Mittwollen (Mittwollen91).

Thermal fluid representation

The so far discussed approaches use 1D or 2D representations of the fluid hydrodynamics between the journal and the shell surface. Some of the works do not consider thermal effects of the fluid at all, see for instance Stefani04, Profito15, Bartel10 and Krampl12. However, in many cases isothermal fluid properties cannot be assumed as the fluid viscosity is strongly influenced by the temperature of the oil (Allmaier13). Two basic approaches – 2D and 3D – will be discussed in the following.

Knoll10 and Andres12 use a 2D energy equation approach which considers only viscous heating, so heating due to local asperity contacts is not considered. Lorenz15 presents a thermo-elasto-hydrodynamic (TEHD) contact model, which computes the oil film temperature using a 2D energy equation. The 2D equation is derived from the equivalent 3D energy equation by integration over the clearance gap height. Besides component material properties such as specific heat capacity, density, heat conductivity for lubricant and structures, also heat transfer through mixed lubricated regimes and partly filled clearance gaps, as implied in cavitation regions, are being considered. Offner13 applies the 2D approach by Lorenz15.

Early works as for instance Priebsch97 and Bukovnik07 already consider a 3D temperature distribution in the oil film. However, these works do neither consider flow factors nor the possibility of metal-to-metal contact by employing an asperity contact model. Hagemann13 use a 3D energy equation for a steady, incompressible oil film. The approach considers both laminar and turbulent regimes, increasing effective viscosity and improvement of the radial heat flux. The method is also applied in Porzig14. Gentre01 use a 3D energy equation to model the thermal behavior in the oil film considering a transient heat equation for journal and shell. Bartel10 uses a 3D energy equation in some application cases and assumes isothermal conditions in others.

3D fluid dynamics modeling

Above listed approaches couple 3D structural dynamics with a 2D Reynolds equation. The step towards a fully coupled 3D representation of the oil film is a large one from the perspective of the applied modelling techniques but also from the perspective of required computation times. However, two works which consider 3D fluid dynamics have been reviewed.

Tucker95 employs a 3D CFD approach for thermo-hydrodynamic analysis considering cavitation. Emphasis is put on the case of a stationary journal center assuming a fixed eccentricity ratio at a predefined operating speed and static load. Both journal and shaft have static surface shape, so there is no dynamic interaction of structural deformations with the oil film. Pressure boundary conditions, as for instance oil supply grooves, are considered as an imaginary boundary. For non-isothermal problems, the lubricant temperature at the boundary is set to the supply temperature and the source term in the thermal transport equation is set to zero in the groove region. Usual diffusion equations are used for conduction of heat into the journal/shell surface, respectively. In addition a number of further boundary conditions have to be defined. These are no-slip, impermeability and pressure radial derivative conditions and others.

Shahmohamadi15 presents a full 3D CFD approach for thermo-hydrodynamic analysis of big-end bearings using a combined solution of the Navier-Stokes-, continuity- and energy equations for multi-phase flow conditions. A vapour transport equation is also included to ensure continuity of flow in the cavitation region for multiple phases as well as the Rayleigh-Plesset approach to take into account the growth and collapse of cavitation bubbles. The approach removes the need to impose artificial outlet boundary conditions in the form of various cavitation algorithms, as for instance those of JFO or Elrod, which are often employed to deal with lubricant film rupture and reformation. The bearing shell is modelled as a stationary wall, while the journal is modelled as moving component with an absolute rotational velocity. The oil film profile is calculated from the average clearance, the eccentricity ratio, the angular position and the non-circularity. The structural dynamics is represented by a two DOF representation of the piston-connecting rod-crank sub-system. The total transient load applied on the big end bearing at any instant of time is a function of the instantaneous connecting rod obliquity angle. At the oil supply orifice to the bearing the lubricant is fed into the contact at a constant pressure. Axially, the lubricant is assumed to flow to ambient (atmospheric) pressure. Transient conduction of the heat into the journal and steady-state conduction into the shell are taken into account. The heat generated in the lubricant is allowed to increase the journal's temperature while the heat transferred to the bushing can be cooled through ambient air. As a simplification no axial temperature gradient for the journal is assumed. A modified density model based on Dowson59 and a viscosity model based on

Houpert85 are utilized. In addition to the hydrodynamic load an asperity contact load according to Greenwood70 is considered. Similar to the applied load, a sum of a viscous component and a boundary component is considered as total friction force.

COUPLING METHODS

Coupling between structure dynamics and fluid dynamics

The coupled hydrodynamic and structural solution of EHD problems in journal bearing systems are traditionally addressed by using either the nodal or the modal approach, see Booker10 and the section *Displacement representation* in this work. Profito15 identifies two sub-methods of coupling for the nodal approach, namely indirect and direct methods. The nodal indirect (or monolithic) methods employ sophisticated implicit Newton-Raphson solution schemes, where the estimation of the solutions in the next steps are computed from a system of residual equations based on the perturbed Reynolds equation and on the applied external loads. All equations are solved simultaneously in this case. The nodal direct (or partitioned) methods are defined in terms of direct iterative schemes, in which the hydrodynamic and structural problems are solved separately. In this case a coupling algorithm is required to incorporate the interaction between fluid and structure, examples are presented in Profito15. The main advantage of the partitioned approach is the possibility of using optimized codes to solve hydrodynamic equations and structural equations separately.

Andres12 assumes steady state conditions; in this case a static variant of the Reynolds equation is used. Bushing and journal are represented by two translational degrees of freedom (DOFs) each. Misalignment as well as the flexibility of the housing is neglected. The fluid film reaction forces are obtained by the integration of the pressures around the bearing circumference. An iterative Newton-Raphson scheme is applied to find the balance between static loads and according reaction forces.

Both Stefani04 and Andres04 use a coupled approach. Center eccentricities of journal and bushing are again represented by two translational DOFs and are calculated using Newton's equation.

Mohammadpour14 also use a coupled approach based on applied loads that considers the journal eccentricity and pressure dependent bushing deformation using an iterative procedure.

Knoll10 represents the rotor using Newton's equations of motion. Both global motion and local shaft deformations are considered. A condensed FEM model is used for the rotor. Bushings and housing are assumed to be rigid.

Smolik15 and SmolikRendlova15 use an approach, where all involved components of the turbocharger model are considered with their elastic properties. The equations of motion of the components are strongly coupled with the Reynolds equations on an iterative basis.

Steinbatz10 present an approach, which couples several individual software parts. A commercial FEM program is used to compute the elastic deformations of components considering local pressure loads (load shape functions) by means of a component mode synthesis. The dynamics of the complete engine assembly are calculated using a commercial MBS program. The elastohydrodynamic simulation to describe the oil film is incorporated into the MBS program by user-written subroutines.

Ozdemir14 outlines a coupled approach involving a simple rigid component mechanism coupled via the journal eccentricity with the Reynolds equation. Cylinder pressure diagrams are used for a kinematic analysis and for computing bearing reaction forces for every degree crank angle in a pre-processing step. Using this data a crank angle based procedure is applied for the entire cycle in a second step. Within each crank angle, the equations of motion of the journal bearing, the Reynolds equation and the asperity contact model are solved iteratively. A similar approach is also outlined by Kim10.

Lang14 uses a weak coupling between component motions and hydrodynamics. In a first step, Newton's equation of motion is solved for all components considering current local and global forces acting on the elastic structures. The resulting local and global accelerations are integrated to velocities and positions (deformations) of the next time step. With these integration values, the lubricated gaps and their time derivatives are determined and together with the hydrodynamic boundary conditions the Reynolds equation is solved. The resulting radial and tangential bearing forces are then again transferred to the components.

Profito15 use a steady state Reynolds equation, which considers the eccentricity of the journal and the radial elastic displacements of the bearing shell.

Hagemann13 iteratively couples the Reynolds equation, the energy equation and the heat conduction equations. In addition, the structural mechanics are considered in terms of journal misalignment and pivot deformation. A Force balance is achieved by applying a radial stiffness coefficient based iterative approach. The radial stiffness is again derived via perturbation of the Reynolds equation.

Bartel10 uses a quasi-stationary coupling between structure and fluid. The compliance matrices are derived using FEM in a preprocessing step. Eccentricity and misalignment are statically described.

Bukovnik07, Lorenz15, Offner13, Offner13-2, Offner15 use strong coupling of mechanical and hydrodynamic equations on an iterative basis. A Newton-Raphson approach is utilized for this purpose, see Offner13. The approach is generic so that both center and surface couplings are possible.

Hydraulic coupling between radial slider bearings

Beside others, the performance of radial slider bearings is influenced by the amount of available lubricant and its distribution within the bearing. In internal combustion engines, grooves and bores serve as oil supplies for radial slider bearings and define transient boundary conditions. The bores of different radial slider bearings, like for instance of a main bearing and a neighboring big end bearing, may be connected via drillings (lines) in the crank shaft. The task of these lines is to transfer lubricant between connected bearings. Furthermore, due to relative motion of different boundary conditions within a bearing, these may overlap geometrically so that several neighboring bearings are interacting at certain times. This results in the requirement to consider the oil supplying lines within a network approach with adaptive connectivity. Furthermore, the flow in lines may be influenced by the motion of the embedding component.

In the special case of turbochargers, the rotor is supported by two radial journal bearings. Each of these bearings connects three bodies – a journal, a floating bushing and the housing. Thus there are two oil films in each radial bearing – between journal and the bushing is one oil film and between the bushing and the shell is a second oil film, see for instance Knoll10 and SmolikRendlova15. The oil supply of the floating bushings is realized with housing drillings, where the outer and the inner oil film are connected by a number of oil feed holes. This case requires a coupled simulation of the oil supply drillings, the two oil films and the structural dynamics. More generally this requires the coupled simulation of a number of oil films all interconnected by a network of drillings.

Chun03 describes models for oil flow through camshaft bearings, hydraulic tappets and oil jets of a four cylinder gasoline engines. The radial slider bearings are considered as simplified engine cycle averaged flow rates. The objective of the work by Yuan07 is a methodology for predicting lubrication flow in the connecting rod bearings and oil circuits. The oil flow in the system is subject to the influence of centrifugal forces, surface acceleration and viscous shear. Soltani10 uses an approach to investigate the lubrication system of a four-cylinder engine, where journal bearings and oil supply lines are offline coupled.

Offner13 shows a generic and flexible network approach. The methodology targets strong interactions between component dynamics, lubricated contacts and connecting straight oil supply drillings. The oil in the drillings is assumed to be incompressible with isothermal viscosity. The flow in each line is described by a steady-state one-dimensional Euler equation, i.e. Bernoulli equation. Fictitious force effects, which result from the motion of the enclosing component, and cavitation effects are considered.

A coupled simulation of the inner and the outer oil film of a turbocharger can be found in Knoll10. Unfortunately, the work does not outline more details on the hydraulic coupling between the inner and the outer oil film. Smolik15 discusses an application, where the inner and the outer oil film are connected with four feed-holes.

Porzig14 presents a simple approach considering the hydraulic interface of the radial lubricant feed holes in the floating ring circumference. A pressure drop across the oil feed holes, which is proportional to the rotational speed of the ring, results from centrifugal forces. The feed pressure is adjusted by an iterative algorithm.

Summary

As this work attempts to summarize recent developments in the simulation of journal bearings, only a brief summary is given. Despite their long use in internal combustion engines, journal bearings are still widely used and face new challenges. New modes of engine operation and lubrication motivate the ongoing development of new and more accurate methods to describe lubricants, coatings, complex surface structures and mixed lubrication in general. An overview of the current developments of the journal bearing simulation methodology is given, which is certainly not complete – given the large number of works published in this field – but hopefully gives a valuable impression on the current developments.

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Definitions/Abbreviations

FEM	Finite element method
MBS	Multi-body simulation
ICE	Internal combustion engine
EHD	Elastohydrodynamic
TEHD	Thermoelastohydrodynamic
DOF	degree of freedom
CDA	cylinder deactivation technology