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Regularised cavitation algorithm for use in transient rotordynamic analysis

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ABSTRACT

Journal bearings are still widely used, and – in combination with the omnipresent trend of light weight construction – there is an increasing focus on transient behaviour. Therefore, the structure, or more precisely, the aspect of rotordynamics has to be considered in conjunction with the hydrodynamics. Existing tools either address rotordynamics accurately while simplifying the bearings, or they model the bearings profoundly, including cavitation, but oversimplify the rotor to a point mass. The paper at hand addresses an approach to close the gap between both worlds. Therefore, a cavitation algorithm for the use in transient rotor dynamic applications is developed based on a regularised formulation of the Elrod–Adams model. Due to the achieved smoothness of the cavitation boundary, the solution of the hydrodynamic task yields to a more robust and faster calculation with respect to the classical approach. Therefore an application in transient simulations including rotor or multibodydynamic interactions under arbitrary conditions is realizable. For validating purposes, other approaches as well as experimental data were considered. The comparison shows an excellent agreement and additionally underlines the numerical advantages. As a result, the influences of rotordynamic aspects like the shaft's time dependent skew positioning can be considered and lead to a more precise representation of the system's behaviour, e.g. bearing damping or minimal fluid film gap.

1. Introduction

Due to their simple configuration, journal bearings are widely used in rotordynamics applications. For design and dimensioning purposes, numerical investigations concerning steady-state load conditions are state of the art. A lot of effort has been done in order to get cavitation constraints coincident with conservation of mass, leading to different implementations of the so-called JFO[1] boundary conditions: Elrod–Adams algorithm [1–5], bi-phase algorithm [6–9] and ALE[2] approach [10–13], wherein the first mentioned is commonly accepted and widely used.

In some cases, however, a steady-state investigation is insufficient – making a transient analysis become mandatory. For this purpose, and due to their rapidity, it is still common to use short bearing approximations as well as look-up tables listing bearing stiffness and damping coefficients [14–18]. Whereas the first approach is not able to handle arbitrary boundary conditions, the second is well developed based on the available cavitation algorithms. The drawback of both is the negligence of damping due to skew gaps generated by bending deflection of the shaft.

An alternative approach is based on a numerical solution of Reynolds equation within the time integration scheme [5,19–21]. Most applications following this approach draw on the EHL[3] theory, considering the deformation of the housing. In this regard, the rotor model is oversimplified to a mass point, but the procedure still results in high computational costs due to the necessity of a fine mesh for cavitation hand ling.

The approach presented here considers shaft and rotor mass properties, including moments of inertia as well as shaft elasticity. It utilises a numerical solution of the Reynolds equation with a modified Elrod–Adams algorithm in order to limit the computational costs to an acceptable amount.

The main idea of Elrod and Adams was to introduce a universal PDE[4] combining the two complementary unknown fields, pressure

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[1] Jakobsson–Floberg–Olsson.
[2] Augmented-Lagrangian–Eulerian.
[3] Elasto-Hydrodynamic-Lubrication.
[4] Partial differential equation.

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Nomenclature

Roman symbols

d	diameter
l	length
g	switch function, gravity
h	film width
H	nondimensional film width $(=\frac{h}{c})$
p	pressure
P	nondimensional pressure $(=\frac{p}{\eta\,\omega}\psi^2)$
t	time
T	nondimensional time $(=t\,\omega)$
u_m	effective surface velocity $(=\frac{\omega_\mathrm{shaft}-\omega_\mathrm{shell}}{2}\frac{d}{2})$
x	circumferential coordinate
X	nondimensional coordinate $(=\frac{2\,x}{d})$
y	axial coordinate
Y	nondimensional coordinate $(=\frac{2\,y}{d})$
$\mathbb{K}$	stiffness matrix
$\mathbb{M}$	mass matrix
$\mathbb{G}$	gyroscopic matrix
$\mathbb{D}$	damping matrix
$\mathbb{C}$	conductivity matrix
ψ	relative clearance $(=\frac{2\,c}{d})$
β	bulk modulus of oil
$\bar{\beta}$	nondimensional bulk modulus of oil $(=\frac{\beta}{\eta\,\omega}\psi^2)$
ω	rotational speed
η	dynamical viscosity of oil
ϑ	film fraction
Π	pressure-like function
Π^*	regularisation parameter
Ω	computational domain $(=\Omega_p+\Omega_\vartheta)$
ε	relative eccentricity $(=\frac{2e}{d})$
ψ_s	relative segment clearance $(=\frac{d_\mathrm{s}-d}{d})$
ρ	density of shaft
ν	Poisson's ratio
E	Young's modulus
$\mathbb{J}$	Jacobian
$\mathbb{r}$	right-hand side vector
$\mathbb{x}$	vector of unknowns
$\mathbb{f}$	force vector

p and film fraction ϑ, each of them associated with a part of the bearing area. The partition depends on load as well as time, and it is realised by a switch function $g\in[0,1]$, whereas the discretisation yields a stepwise nonlinear system of equations, which is usually solved by fixed-point iteration.

In order to determine the cavitation boundary sufficiently accurate, the bearing area has to be meshed relatively fine, which leads to high numerical efforts. Nevertheless, oscillations can occur during the iteration process due to the heaviside-type switch function g [22–24].

The modified Elrod–Adams model presented here uses a regularised switch function to avoid oscillations and to reduce the number of unknowns. Furthermore, a Newton–Raphson algorithm can be utilised to solve the nonlinear equations more efficiently.

First, in Section 2 the original approach is described briefly, followed by an introduction to the modifications. Afterwards, the algorithm will be validated under steady-state load, and a convergence study is performed, Section 3. Since the scope of the paper is the dynamic behaviour of bearings, Section 4 addresses an additional validation under transient load in the case of a parallel gap. In conclusion, the modified Elrod–Adams model is applied to a rotor supported in journal bearings, leading to skew gaps depending on time and load, Section 5.

2. Hydrodynamic theory

2.1. Governing equations

The purpose of hydrodynamic models in a rotordynamic analysis is to provide bearing forces and moments resulting from the pressure distribution p, which can be calculated by solving the Reynolds equation

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial y}\right)=u_\mathrm{m}\frac{\partial(\vartheta h)}{\partial x}+\frac{\partial(\vartheta h)}{\partial t}. \tag{1}$$

To address the cavitation problem, the bearing area is divided into two domains – the pressurised Ω_p and the cavitational Ω_ϑ. The first one is associated with a completely filled gap $\vartheta=1$ and the unknown pressure p. In the latter, the pressure is supposed to be $p=p_\mathrm{cav}$, while ϑ has to be determined.

Elrod and Adams, by use of

$$p=p_\mathrm{cav}+\beta\ \ln\ \vartheta, \tag{2}$$

came up with the idea to establish the film fraction ϑ as unique unknown, valid in both regions in order to satisfy the JFO cavitation constraints [1]. In addition, they introduced the switch function $g(\vartheta)$, resulting in the nonlinear PDE

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\beta\,g(\vartheta)\frac{\partial\vartheta}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{h^3}{12\eta}\beta\,g(\vartheta)\frac{\partial\vartheta}{\partial y}\right)=u_\mathrm{m}\frac{\partial(\vartheta h)}{\partial x}+\frac{\partial(\vartheta h)}{\partial t}$$
$$\text{with}\quad g(\vartheta)\overset{!}{=}\begin{cases}0 & \forall\ \vartheta<1\\ 1 & \forall\ \vartheta\geq 1\end{cases}. \tag{3}$$

The drawback of this formulation is the weak dependency $p(\vartheta)$ in Ω_p, which can be avoided – following an idea of Shi and Paranjpe [5] – by introducing a universal unknown Π containing the nondimensional pressure P as well as the film fraction ϑ

$$\Pi(x,y)\overset{!}{=}\begin{cases}\vartheta(x,y)-1 & (x,y)\in\Omega_\vartheta\\ P(x,y) & (x,y)\in\Omega_p\end{cases}. \tag{4}$$

The switch function g then becomes

$$g(\Pi)\overset{!}{=}\begin{cases}0 & \forall\ \Pi<0\\ 1 & \forall\ \Pi\geq 0\end{cases}, \tag{5}$$

whereby Eq. (3) reads – under consideration of nondimensional relations according to the nomenclature – in terms of Π, as follows:

$$\left[\frac{\partial}{\partial X}\left(\frac{H^3}{12}\frac{\partial\Pi}{\partial X}\right)+\frac{\partial}{\partial Y}\left(\frac{H^3}{12}\frac{\partial\Pi}{\partial Y}\right)-\frac{\partial H}{\partial X}-\frac{\partial H}{\partial T}\right]$$
$$g+\left[\frac{\partial((\Pi+1)H)}{\partial X}+\frac{\partial((\Pi+1)H)}{\partial T}\right](g-1)=0. \tag{6}$$

The remaining boundary conditions can be stated as

$$\Pi=\Pi_\mathrm{inlet}\quad\text{on }\partial\Omega_p^\mathrm{inlet},$$
$$\Pi=\Pi_\mathrm{amb}\quad\text{on }\partial\Omega_p^\mathrm{amb},$$
$$\frac{\partial\Pi}{\partial Y}=0\quad\text{on }\partial\Omega_\vartheta^\mathrm{amb}. \tag{7}$$