



Improved identification of squeeze-film damper models for aeroengine vibration analysis

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ABSTRACT

Numerical solution of the Reynolds equation imposes a prohibitive computational cost on the dynamic analysis of practical squeeze film damped turbomachinery. To surmount this problem, the present paper develops the use of Chebyshev polynomial fits to identify finite difference (FD) solution of the incompressible Reynolds equation. The proposed method manipulates the Reynolds equation to allow efficient and accurate identification in the presence of cavitation, the feed-groove, feed-ports, end-plate seals and supply pressure. The ability of Chebyshev polynomials to rapidly reproduce FD routines is demonstrated. The bearing models developed are experimentally proven to give more accurate results than alternative analytical bearing models.

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1. Introduction

Squeeze film damper (SFD) bearings are an effective solution to the problem of attenuating vibration and transmitted forces caused by rotor unbalance in gas turbine engines. Their simplicity of construction and consequent robustness mean they are commonplace in modern aircraft gas turbine engines. For example, the engines of a leading aero-engine manufacturer typically have 5–6 SFDs. However, their inclusion requires careful unbalance response calculations that take into account of the SFD's nonlinearity, to ascertain smooth running [1].

Methods used to analyse the nonlinear response of rotor-dynamic systems may be classified as either time domain or frequency domain techniques. A time domain method marches the equations of motion forward in time until a steady-state solution is reached. This solution may not necessarily be periodic. This process requires the numerical solution of the coupled nonlinear equations of motion at each of a very large number of time steps. A frequency domain approach like the harmonic balance (HB) method solves nonlinear algebraic equations to extract steady-state solutions that are assumed to be periodic of given fundamental frequency. The relative merits of these two complementary approaches are discussed in [1]. A typical engine model requires consideration of many hundreds of modes, posing

prohibitive demands on conventional time/frequency domain methods. The issue of the large number of modes was recently resolved through the development of novel fast time/frequency domain methods [1]. However, like conventional methods, these new methods still require a number of SFD force computations per time step (time domain) or iteration (HB). In the case of HB, one iteration requires the calculation of the SFD forces at each of an adequate number of time points over the period of vibration in order to obtain their Fourier coefficients. Moreover, the iterative process requires a Jacobian matrix that is obtained through repeated calculation of these Fourier coefficients. The SFD model can cripple a time/frequency domain solver unless the SFD forces are rapidly computed. This has led to a trade-off between the capability/reliability of the SFD model and its computational cost.

Della Pietra and Adiletta [2] provide a comprehensive review of SFD modelling techniques, where the above-mentioned trade-off is evident among most of the research works cited. Advanced SFD models based on the Navier–Stokes equation to account for fluid inertia were proposed in [3,4]. However, in order to enable efficient solution for just one SFD, the system dynamics had to be restricted to circular centred journal orbits or small-amplitude orbits about the bearing centre. These restrictions render such an approach unsuitable for real engines like that considered in [1].

The typical and more practical approach in SFD modelling is based on the incompressible Reynolds lubrication equation to describe the relationship between the pressure in a fluid film and the journal motion. Even then, analytical solution is only achievable if the equation is simplified. Most simplifications are based upon neglecting pressure gradients in one direction, resulting in solutions termed long and short bearing approximations and their

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Nomenclature

a_j, b_j, c_j, d_j grouped constants—see Section 2.1

A_h area of feed-port, m^2

c clearance of a centred SFD, m

C_i Chebyshev polynomial coefficients

d gap between sealing plate and journal, m

e journal eccentricity, m

F unbalance force amplitude, N

h squeeze film thickness, m

I moment of inertia about pivot, $kg\ m^2$

l_p length feed-port, m

l_s effective length of end-plate seal, m

L land length, m

L_g groove length, m

p pressure, Pa

p_o outlet pressure, Pa

p_s supply pressure, Pa

$p_{i,j}$ pressure at node i,j in the mesh, Pa

Q squeeze film force, N

R radius of SFD journal, m

t time, s

T_i the Chebyshev polynomials

u fluid velocity in the x direction, m/s^2

U unbalance, $kg\ m$

v_f average fluid velocity in feed-port, m/s^2

w fluid velocity in the z direction, m/s^2

$\Delta x, \Delta z$ node spacing in the x and z directions, m

C_j, E_j, D_j, R_j FD matrices

ε non-dimensional journal eccentricity

θ angular location, rad , see Fig. 2

μ dynamic viscosity, $Pa\ s$

ψ attitude angle, rad

ω rotational speed, rad/s

$\hat{()}$ indicates a non-dimensional property

$\dot{()}$ differentiation with respect to time t

$\dot{()}'$ differentiation with respect to $\tau (= \omega t)$

variants (e.g. [5]). The most prominent of these variants involves the combination of the short and long pressure solutions through an ‘end-leakage factor’ λ that accounts for the degree of end-sealing [5,6]. This model is popular with industry and was used recently for whole-engine analysis with 5 SFDs [1]. The factor λ can be described as an empirical ‘fudge factor’ that in reality can only be related to the parameters of a given bearing when it is hosted in a given experimental setup. Attempts to theoretically relate it to the SFD parameters have proved unsuccessful [5].

To solve the Reynolds equation in its full two-dimensional form, with arbitrary boundary conditions, a numerical scheme like finite difference [4], or finite element (FE) and its variants [7], is necessary. The direct use of such a numerical scheme within a time/frequency domain solver is computationally prohibitive for practical applications. Hence, the recently added SFD routine in NastranTM is based on a one-dimensional approximation of the FD model that is restricted to long and short unsealed SFDs [8]. The full 2-D model can be rapidly deployed within a solver via an identification scheme, which usually involves an interpolation procedure [9,10]. Rodrigues et al. [9] use Chebyshev polynomial fits of a numerical bearing model to relate the three variables e , $\dot{e}$ and ψ to the two orthogonal output force vectors Q_R and Q_T (Fig. 1).

The assumption of a uniform boundary condition at the groove in [9] meant that, in the absence of fluid inertia effects, $Q_{R,T}$ were only functions of the aforementioned three variables. The inclusion of feed-ports into a bearing model necessitates the inclusion of ψ in the input space. Identification of functions with

more than three variables is possible, yet the computation becomes numerically cumbersome [9]. It follows that the inclusion of feed-ports would not be practical using the technique of Rodrigues et al.

Before the identification procedure may take place, the range of each of the input variables must be prescribed. This is simple in the case of e (bounded by radial clearance c) and ψ (if the feed-ports are equi-spaced by angle α around the circumference then $Q_{R,T}$ are periodic in ψ , period α). However, selecting a suitable range for $\dot{e}$ and $\dot{\psi}$ is more difficult, since these variables have no natural bounds. For example in [9] the upper limits of the magnitudes of these two variables were set arbitrarily to 0.7 m/s and 20,000 rad/s, respectively. The judicious prescription of the range of these two input variables is critical to maintaining accuracy of the fit. Widening their range lowers the quality of fit, all other things kept equal. A further consideration is that part of the ‘training’ data required for the calculation of the Chebyshev coefficients involves unrealistic combinations of the input variables, namely high e and high $\dot{e}$, for which $Q_R \rightarrow \infty$. This condition is unrealistic since Q_R increasingly acts to reduce $\dot{e}$ as e grows. The inclusion of such unrealistic input combinations skews the identification.

The work of Chen et al. [10], although using a crude identification technique based on linear interpolation, surmounts the problems associated with defining the input range by applying a reduction technique that exchanges $\dot{e}$ and $\dot{\psi}$ for a single variable with limits of ± 1 . Moreover, the ability to reduce the number of input variables by one makes it practical to use Chebyshev polynomials to identify SFD models that necessitate inclusion of ψ . The major shortcoming of the reduction technique, as applied by Chen et al., is that it cannot accommodate fluid film rupture at a non-atmospheric pressure (e.g. absolute zero) and cannot handle nonzero boundary conditions (e.g. oil supply pressure).

The research of the present paper addresses the need for reliable identification of numerical models. It combines the relative strengths of the work in [9,10] and uses two novel techniques to overcome their limitations. Firstly, Chebyshev interpolation, while using the reduction technique, is performed on the pressure. This allows post-interpolation truncation of the pressure to account for cavitation. Secondly, static and dynamic elements of the pressure function are separated. This treatment allows the identification of bearing models that have

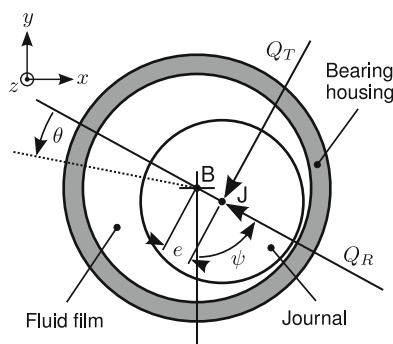


Fig. 1. Polar coordinate system.

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