

Calculation of the Unbalance Response of Whole Aero-Engine Models

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Summary. An aero-engine assembly is typically fitted with squeeze-film damper (SFD) bearings that are highly nonlinear in their performance. The calculation of its nonlinear vibration response to rotational unbalance is computationally prohibitive unless the analysis technique has the following qualities: (a) the ability to handle the structural complexity of two/three-spool engines; (b) the ability to rapidly compute the SFD forces. This paper summarises the authors' recent and ongoing contributions in this area.

Introduction

Aero-engine assemblies typically have at least two nested rotors mounted within a flexible casing via SFD bearings, as shown in Figures 1 and 2. SFDs are proven to be a very cost-effective solution to the problem of attenuating vibration caused by rotor unbalance. However, in order to achieve the best deployment of SFDs in an engine it is necessary to carry out calculations on the whole-engine structure that take due account of the SFDs' nonlinearity.

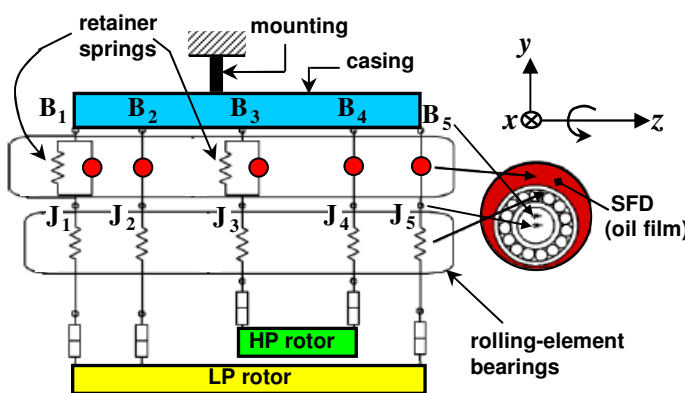


Figure 1: Twin-spool aero-engine [1]

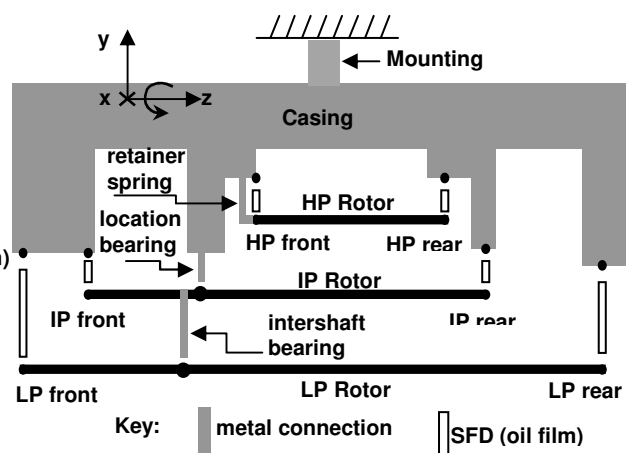


Figure 2: Three-spool aero-engine [2]

The nonlinear solvers used to calculate the unbalance response work either in the time domain or in the frequency domain. Time domain solvers progress forward in time until a steady-state response is obtained that may not necessarily be periodic. Frequency domain solvers extract steady-state solutions that are assumed to be periodic of given fundamental frequency. An efficient computational facility takes advantage of the relative merits of both approaches through an integrated strategy that makes effective use of both [1]. A time/frequency domain solver that is useful for realistic aero-engine models should be able to accommodate an essentially large number of assembly modes at minimal computational cost. This is achieved by the novel Impulsive Receptance Method (IRM) and the Receptance Harmonic Balance Method (RHBM), which efficiently solve the nonlinear problem in the time and frequency domains respectively [1, 2]. Prior to these methods, the analysis of realistic engine structures was computationally prohibitive even when done with relatively simple (analytical) SFD models. Having established these new methods, it is desirable that they operate efficiently even with advanced numerical SFD models. This can be achieved by the nonparametric identification of advanced numerical SFD models [3].

Theoretical overview

The typical approach with nonlinear rotordynamic solvers is to regard the complete nonlinear rotordynamic assembly as a non-rotating linear part acted on by "external" forces. By "linear part" is meant the structure left after all SFDs in the schematics of Figs. 1, 2 are replaced by 'gaps'. The equations of motion are then transformed into modal space using the modes (eigenvectors) of the undamped linear part of the assembly under non-rotating conditions. The unknowns are then contained in the vector (column matrix) of modal coordinates \mathbf{q} . This results in $2R$ first order differential equations that are nonlinearly coupled on the right hand side by the SFD forces that are nonlinear functions of the displacement and velocities of the journals (marked 'J' in Fig. 1) relative to their housings (marked 'B' in Fig. 1). The number R of modes necessary for the transformation is much less the number of physical coordinates (which runs into hundreds of thousands). Nonetheless, R is still a high number, typically 1000 for a two-spool engine and 2000 for a three-spool engine, covering a frequency range 0-500Hz (frequencies up to at least 500Hz need to be considered in the nonlinear response due to the presence of harmonics of the rotor speeds up to 14,000 rpm) [1, 2]. Conventional implicit

integration algorithms transform the 2R differential equations into an equal number of algebraic equations which then have to be solved iteratively at each time step to obtain the current modal state variables. Hence, for a whole-engine model, the time-marching process slows down to impractical levels.

The IRM equations relate the instantaneous relative displacements and velocities at the nonlinear elements (SFDs) with the motion-dependent excitations (SFD forces, gyroscopic moments) and other excitations (unbalance, gravity) acting on the linear part of the system. It is these equations, totaling $4n$ (where n is the number of SFDs), that are solved at each time step. Hence, unlike conventional implicit integrators, the IRM's computational efficiency is largely immune to the number of modes [1]. Once the displacements and velocities at the SFDs are computed at a given time step, the modal state variables \mathbf{q} , $\dot{\mathbf{q}}$ are updated at no computational cost. The RHBM is the frequency-domain counterpart of the IRM. It assumes that the vibration is periodic with K harmonics of an assumed fundamental frequency ϖ . The RHBM uses receptances (frequency response functions) of the linear part to relate the harmonics of the relative displacements at the SFDs with the corresponding harmonics of the excitations (motion-dependent and otherwise) acting on the linear part of the system [1]. The unknowns to be determined are the harmonics of the relative displacements at the SFDs plus a few extra unknowns to cater for the zeroth harmonic (the equations at this latter harmonic describe the statically indeterminate equilibrium of the mean SFD forces with the distributed rotor weight). Since the SFD forces are known nonlinear functions of the relative displacements and velocities, the harmonics of the SFD forces are determined through a Fourier analysis of their time histories for an assumed solution. This allows solution by iteration [1]. The choice of ϖ depends on whether the unbalance is restricted to one rotor only (single frequency unbalance-SFU) or is present on more than one rotor (multi-frequency unbalance excitation- MFU). In the case of SFU, ϖ is simply the speed of the unbalanced rotor. In the case of MFU, $\varpi = \Omega_{LP}/N_0$ where N_0 is an integer that depends on the ratio of the rotor speeds and Ω_{LP} is the speed of the LP shaft (slowest shaft). A large N_0 inevitably leads to a large number of harmonics K . To avoid such a computational burden, the actual speeds are slightly adjusted so as to obtain the smallest possible N_0 as follows: (i) Let $\alpha = \Omega_{IP}/\Omega_{LP}$, $\beta = \Omega_{HP}/\Omega_{LP}$; (ii) Find whole numbers p , q , r and s with q and s smallest possible so that $|\alpha - p/q| \leq \zeta\alpha$, $|\beta - r/s| \leq \zeta\beta$ in which ζ is a prescribed tolerance; (iii) Find the least common multiple number N_0 of q and s ; (iv) Establish the fundamental frequency using the formula $\varpi = \Omega_{LP}/N_0$.

Simulation Results

Simulations were performed in Matlab on a standard desktop pc with Intel® Pentium® D CPU 3GHz processor. Figs. 3(a-f) show the excellent agreement between the IRM and RHBM results for the three-spool engine (Fig. 2) with unbalance on all three rotors. These figures show the trajectory of the centre of the journal of SFD 'LP rear' relative to the centre of its housing for different engine speeds. This trajectory is bounded by the annular clearance circle.

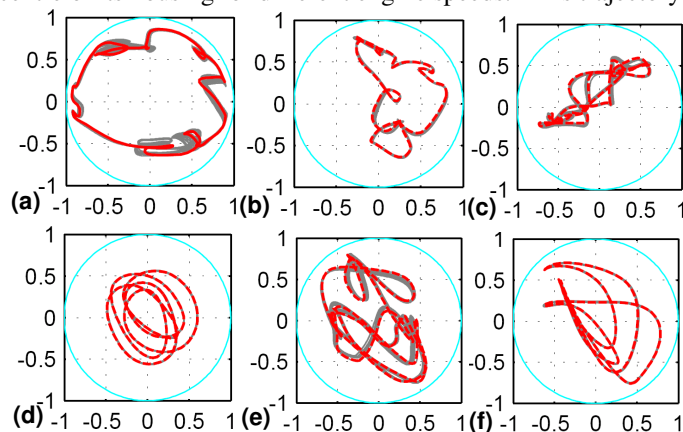


Table 1. RHBM solution parameters

Fig.	N_0	K
3a	1	38
3b	1	32
3c	1	23
3d	2	25
3e	4	36
3f	2	22

Figure 3: Comparison between RHBM (.....) and IRM (————) orbits of SFD 'LP rear' of the three-spool engine

Conclusions

Novel robust algorithms for the rapid calculation of the forced nonlinear response of whole aero-engine models have been discussed. The results presented here were for an analytical oil-film model. Current efforts are being directed towards the nonparametric identification of advanced numerical SFD models [3] (their inclusion within the solvers will be shown in the presentation).

References

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- [3] Groves, K.H., Bonello, P. (2010), Improved identification of squeeze-film damper models for *Tribology International* **43**(9): 1639-1649