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Methods for linearized analysis of floating ring bearings

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Automotive turbochargers [1] are typically supported by floating ring bearings (FRBs), where a rotor journal is held by two oil films separated by a rotating ring. This construction brings the advantage of high damping capacity, however, the rotor tends to behave in the nonlinear way—during the operation, oil whirl/whip instabilities [3] occur in the inner or outer oil film or in both at the same time. This leads to *jump phenomena* [2, 5] in the rotor response during runup/rundown.

Although the rotors with the FRBs are by nature nonlinear, linear analysis can also provide important insight into the dynamical behaviour of such a system. Using the linear approach, the nonlinear forces generated in the fluid films are approximated by linear or linearized spring-damper couplings. If the modelling is done precisely enough, it can show the possible regions of instability and predict operation modes of the rotor in these regions. There are several methods to create the linear model, that are discussed in this contribution. In order from the simpliest to the most complex, the bearing model can be considered as follows: (i) constant isotropic approximation of the whole FRB, (ii) constant orthotropic approximation of the whole FRB, (iii) FRB with linearized outer oil film only [4], (iv) constant isotropic approximation of the inner and outer fluid film with neglected ring mass, (v) constant isotropic approximaiton of the inner and outer fluid film with ring mass considered, (vi) ring mass considered and both fluid films linearized. Moreover, the last method can be performed in several ways:

- (a) separated linearization for the inner and the outer fluid film,
- (b) coupled linearization for both fluid films,
- (c) same as (b) with ring speed ratio (RSR) variable with eccentricity,
- (d) same as (b) with RSR calculated based on its own degree of freedom in the static equilibrium (algebraic) equations.

In general and not distinguishing between the methods with and without ring degrees of freedom, the turbocharger with two floating ring bearings can be described by the matrix equation

$$\boldsymbol{M}\ddot{\boldsymbol{q}} + \left[\boldsymbol{B} + \boldsymbol{G}(\omega_R) + \boldsymbol{B}_B^{(M)}\right]\dot{\boldsymbol{q}} + \left[\boldsymbol{K} + \boldsymbol{K}_B^{(M)}\right]\boldsymbol{q}_R = \boldsymbol{0}, \tag{1}$$

where $\boldsymbol{M}, \boldsymbol{B}, \boldsymbol{G}, \boldsymbol{K} \in \mathbb{R}^{n,n}$ are global mass, damping, gyroscopic and stiffness matrices, respectively, without bearing couplings. Bearing couplings are expressed using coupling bearing matrices $\boldsymbol{K}_B^{(M)}, \boldsymbol{B}_B^{(M)} \in \mathbb{R}^{n,n}$. The particular form of these matrices depends on the chosen method $M = i, ii, \ldots vi$. For the methods with $M = i, ii, iv, v$, these matrices are constant and for $M = iii, vi$ (in all the considered variants), they are a product of linearization process and hence they are speed dependent, i.e. $\boldsymbol{K}_B^{(M)} = \boldsymbol{K}_B^{(M)}(\omega_R, \omega_{FR})$, $\boldsymbol{B}_B^{(M)} = \boldsymbol{B}_B^{(M)}(\omega_R, \omega_{FR})$. Number of degrees of freedom is also dependent on the chosen method – for the methods with neglected ring mass $n = n_R$ (number of degrees of freedom of the rotor) and for the methods with FRBs considered $n = n_R + 4$ (a,b,c) and $n = n_R + 6$ (d). Since the stability of the turbocharger is analysed using Campbell diagrams, right-hand side of Eq. (1) equals zero for modal analysis purposes.

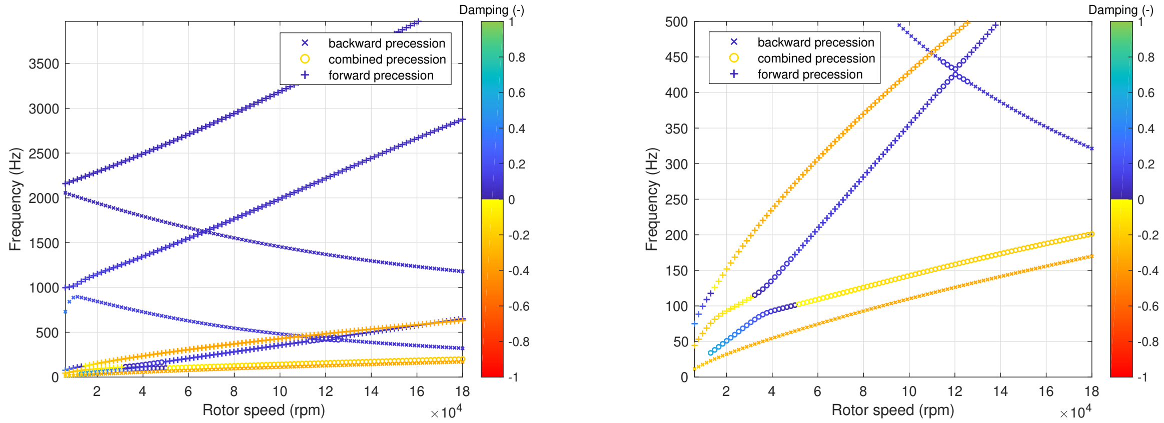

Figure 1. Campbell diagram of the turbocharger considering separated static solution for both fluid films – the diagram up to 3000 Hz (*left*) and the detail of unstable branches (*right*)

The results are analysed using the Campbell diagrams depicting together all important information about the system: they show change of natural frequencies with angular speed of the rotor, the precession (forward, backward and combined) is distinguished by different markers and the damping is shown using a carefully designed colormap. The results show significant differences between proposed methods with respect to the measure of abstraction. Slight differences in Campbell diagrams are obtained in case of method (vi) with all the considered subcases (a)-(d). However, the computational time is strongly affected by the chosen method. The contribution also shows a comparison of resulting Campbell diagrams with nonlinear run-up simulations.

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