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An efficient approach to model dynamics of a small engine crankshaft

V. Houdek[a], L. Smolík[a]

[a] Faculty of Applied Sciences, University of West Bohemia, Univerzitní 8, 306 14 Plzeň, Czech Republic

This paper deals with the computational modelling of a crankshaft. The main aim is to provide the overview of an efficient approach to model dynamics of a small engine crankshaft. The introduced computational model includes flexibility of bodies and employs a non-linear model of hydrodynamic forces in journal bearings. The whole model is assembled and analysed in MSC Adams software for multibody system dynamics.

The presented approach is based on the division of the crankshaft into a system of rigid bodies shown in Fig. 1 which are coupled with massless beam elements in accordance with Fig. 2. Properties of these elements, such as the second moment and polar moment of the cut area, are tuned in accordance with the results of a structural analysis with 3D finite elements.

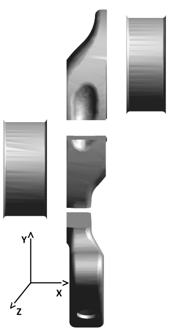

Fig. 1. Crankshaft division

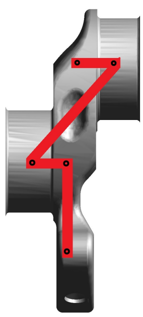

Fig. 2. Beam elements

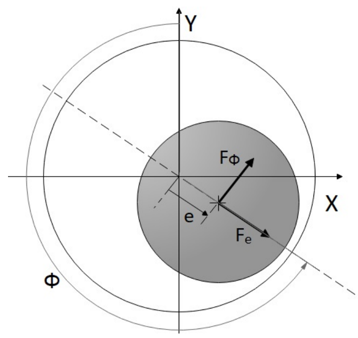

Fig. 3. Hydrodynamic forces

The beam properties has been tuned only in one part of the crankshaft. The tuned beams are depicted in Fig. 2 as red lines. Three types of static loads were defined for the system: a main bearing journal was fixed and forces in the main directions were applied in the centre of mass of a rod bearing journal. Resulting displacements of the center of the rod bearing journal are shown in Table 1.

Table 1. Displacement of the rod pin after tuning of the model parameters

force direction	**displacement – Ansys [μm]**	**displacement – MSC Adams [μm]**
X	2.265	2.275
Y	2.324	2.337
Z	2.018	2.040

The hydrodynamic forces (HD) acting in journal bearings of the engine are set as forces in the infinitely short bearing. The analytical solution of these forces is considered in the form of

$$F_e(\epsilon, \dot{\epsilon}, \dot{\Phi}, \omega) = -\mu R L \frac{L^2}{c} \cdot \left[|\omega - 2\dot{\Phi}| \frac{\epsilon^2}{(1-\epsilon^2)^2} + \frac{\pi(1+2\epsilon^2)\dot{\epsilon}}{2(1-\epsilon^2)^{2.5}} \right], \tag{1}$$

$$F_\Phi(\epsilon, \dot{\epsilon}, \dot{\Phi}, \omega) = \mu R L \frac{L^2}{c} \cdot \left[(\omega - 2\dot{\Phi}) \frac{\pi\epsilon}{4(1-\epsilon^2)^{1.5}} + \frac{2\epsilon\dot{\epsilon}}{(1-\epsilon^2)^2} \right], \tag{2}$$

where F_e and F_Φ are forces acting on the journal in radial and tangential directions respectively (Fig. 3), μ is the oil dynamic viscosity, R is the rotor radius, L is the bearing length, c is the radial clearance, ω is the rotor speed, ϵ is the rotor eccentricity and Φ is the angle to the eccentricity [1]. Validation of the forces was performed using a simplified model which only consists of a rotor pin supported on a journal bearing. The motion of the pin was compared with the motion of the Jeffcott rotor which is given in [2] and that has been simulated for the considered HD forces in journal bearings and for rotational speeds $\omega = \pi\, n/30$ rad/s, where $n = 2000 - 2900$ RPM.

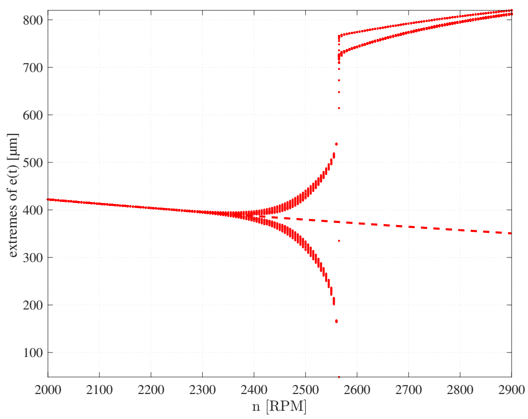

Fig. 4. Local extremes of displacement of the rotor pin [2]

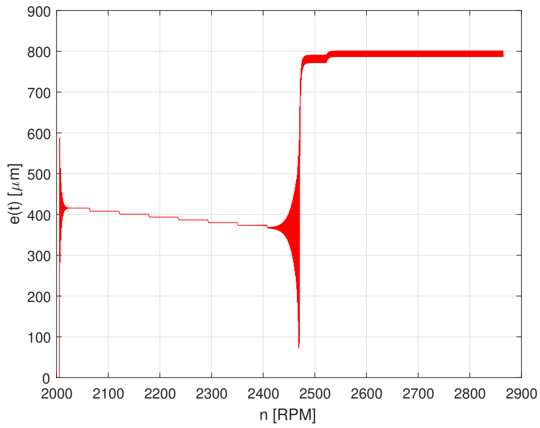

Fig. 5. Displacement of the rotor pin (MSC Adams)

The introduced computational model presents a powerful tool for a powertrain vibrations analysis. It is capable of incorporating not only linear but also non-linear behaviour of powertrain components and subsystems. The complex computational model should be verified by suitable experimental methods.

Acknowledgement

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References

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[2] Dyk, Š., Byrtus, M., Smolík, L., Steady-state behaviour of the rigid Jeffcott rotor comparing various analytical approaches to the solution of the Reynolds equation for plain journal bearing, DSTA 2017: Dynamical Systems in Applications. Cham, Springer, 2018, pp. 95-103.