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Vibration analysis of a vertical rotor immersed in fluid at extreme operating temperatures

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Contemporary fusion reactors, such as the International Thermonuclear Experimental Reactor (ITER), require deuterium and tritium to fuel the fusion reaction. Whereas deuterium can be easily extracted from seawater, tritium is very rare. Fortunately, a tokamak can produce tritium when neutrons escaping the plasma collide with lithium in its inner wall. This procedure is called tritium breeding. The search for the superior tritium breeding material is still ongoing, although it started more than 40 years ago. One of the potential candidates evaluated during the ITER project is lead-lithium (PbLi) eutectic material [5], a homogenous mixture of Pb and Li that solidifies at a single temperature. The mixture is produced in a *saturator* that continuously mixes involved substances under laminar flow conditions.

The saturator can be constructed as a vertical rotor with several thin discs, which is placed in a pressure vessel. The clearance between the saturator and vessel is occupied by PbLi, which is mixed as it flows through the vessel thanks to the saturator rotation. The temperature of the PbLi mixture reaches 550 °C, rendering the usage of rolling element bearings or petroleum-based lubricants in journal bearings impossible. However, the dynamic viscosity of PbLi is only $1.28\,\mathrm{mPa\,s}$ at 550 °C [5], which is suitable for providing hydrodynamic lubrication at low specific loads. On the other hand, the density of PbLi at 550 °C is ca. $9300\ \mathrm{kg\,m^{-3}}$ resulting in significant buoyancy and inertia forces acting on the saturator.

This paper explores the potential of using PbLi as a lubricant in the journal bearings of the saturator. PbLi has similar lubricating properties to water due to its low dynamic viscosity, which raises concerns regarding possible whirl/whip instability [2]. Moreover, the high temperature and density of PbLi influence the mechanical properties of the rotor and can cause problems in a thrust bearing due to centrifugal forces. As noted by Nishimura et al. [2], the influence of the pressure vessel (*stator*) on the rotor dynamics is also critical because its first natural frequency is close to the nominal speed of the saturator.

Fig. 1 shows a saturator assembly consisting of three flexible bodies (main rotor, drive shaft and stator), a rigid drive and couplings including a radial journal bearing, a conical bearing supporting the main rotor radially and axially, three deep groove ball bearings and two torsional couplings. Since the assembly contains several flexible bodies and nonlinear couplings, its motion is described utilising the Newton-Euler equations for multi-body dynamics. The exact form of these equations for body B reads [3]

$$\boldsymbol{M}\,\dot{\boldsymbol{v}} - \boldsymbol{p}^{\mathrm{rb}}\left(\boldsymbol{s},\,\dot{\boldsymbol{s}}\right) = \boldsymbol{p}^{\mathrm{gyr}}\left(\boldsymbol{s}\right) + \boldsymbol{f}^{j}(\boldsymbol{s},\,\boldsymbol{w}) + \boldsymbol{f}^{e}(\boldsymbol{s}) - \boldsymbol{D}\,\dot{\boldsymbol{q}} - \boldsymbol{K}\,\boldsymbol{q}, \tag{1}$$

$$\dot{\boldsymbol{x}}_B = \boldsymbol{v}_B, \tag{2}$$

$$\dot{\boldsymbol{q}} = \boldsymbol{v}, \tag{3}$$

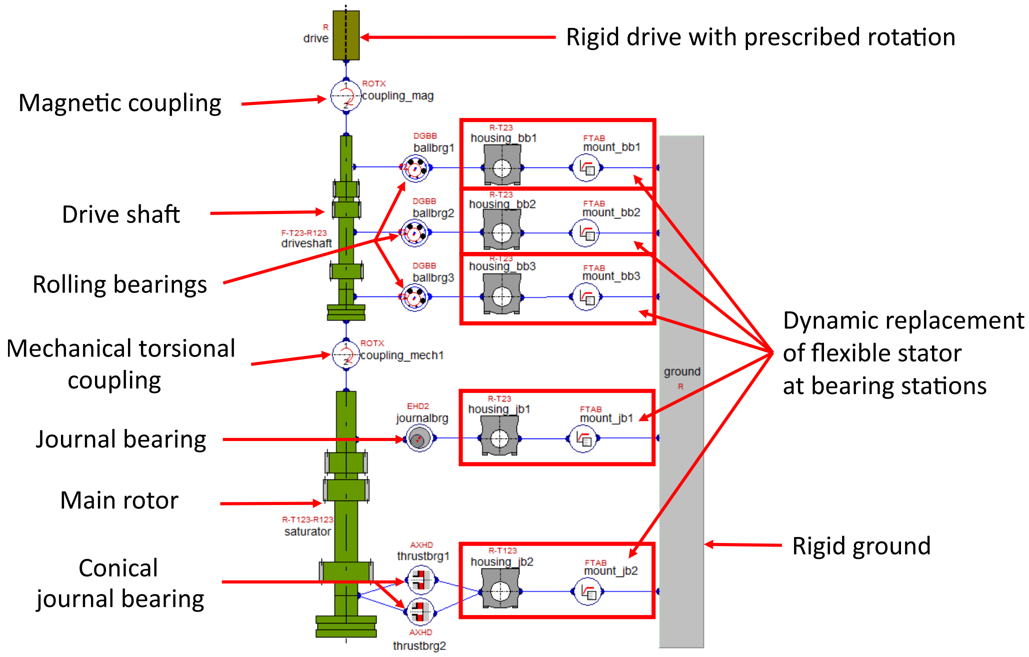

Fig. 1. Scheme of the computational model of the saturator

$$\boldsymbol{\theta}_B^{\mathsf{T}}\boldsymbol{\theta}_B = 1, \tag{4}$$

$$2\,\boldsymbol{S}\left(\boldsymbol{\theta}_B\right)\dot{\boldsymbol{\theta}}_B = \boldsymbol{\Omega}_B, \tag{5}$$

$$\boldsymbol{r}\left(\boldsymbol{q}\right) = \boldsymbol{0}, \tag{6}$$

where $\boldsymbol{x}_B \in \mathbb{R}^{3\times1}$ and $\boldsymbol{\theta}_B \in \mathbb{R}^{4\times1}$ hold the position of a body reference and its orientation given by four Euler parameters. $\boldsymbol{q} \in \mathbb{R}^{6N\times1}$ contains all elastic coordinates defined w.r.t. the body reference with N being a number of nodes that describe the body configuration. State vector $\boldsymbol{s} = \left(\boldsymbol{x}_B^{\mathsf{T}},\, \boldsymbol{\theta}_B^{\mathsf{T}},\, \dot{\boldsymbol{x}}_B^{\mathsf{T}},\, \boldsymbol{\Omega}_B^{\mathsf{T}},\, \boldsymbol{q}^{\mathsf{T}},\, \dot{\boldsymbol{q}}^{\mathsf{T}}\right)^{\mathsf{T}}$ contains both global and elastic coordinates and their respective time derivatives, and vector $\boldsymbol{w}$ consists of state vectors of all bodies coupled to B. $\boldsymbol{M}$, $\boldsymbol{D}$, $\boldsymbol{K} \in \mathbb{R}^{6N\times6N}$ are constant mass, damping and stiffness matrices, respectively. Vectors $\boldsymbol{p}^{\mathrm{rb}}$, $\boldsymbol{p}^{\mathrm{gyr}}$, $\boldsymbol{f}^{j}$, $\boldsymbol{f}^{e} \in \mathbb{R}^{6N\times1}$ incorporate forces resulting from inertia and gyroscopic effects, joint deformations and predefined external loading, respectively. An exact form of vectors $\boldsymbol{p}^{\mathrm{rb}}$, $\boldsymbol{p}^{\mathrm{gyr}}$ is derived in [3] and the form of vector $\boldsymbol{f}^{j}$ depends on governing equations of the coupling forces.

Eqs. (2) and (3) are substitutions required for a numerical integration. Eqs. (4) and (5) describe relations between the Euler parameters, body orientation and angular speed [3]. The system of Eqs. (1)–(5) does not necessarily have a unique solution. Therefore, Eq. (6), which ensures a unique separation of the global and flexible coordinates, has to be attached [3].

Hydrodynamic forces acting in the journal bearings are evaluated employing the modified Reynolds equation complemented with the JFO cavitation model. Governing equations and a method for their solution based on finite volume and multi-grid methods are introduced in [4], and boundary conditions are shown in Fig. 2. Elastic forces acting in the rolling element bearings are estimated using the Hertzian theory. The dynamic characteristics of the stator are computed using a detailed finite element model, which is capable of evaluating frequency response functions (FRFs) between individual bearing stations and the ground. The multi-body model incorporates the FRFs that contain the first two natural frequencies, which correspond to first-order bending mode shapes.

The influence of PbLi on flexural vibrations of the main rotor is implemented in accordance

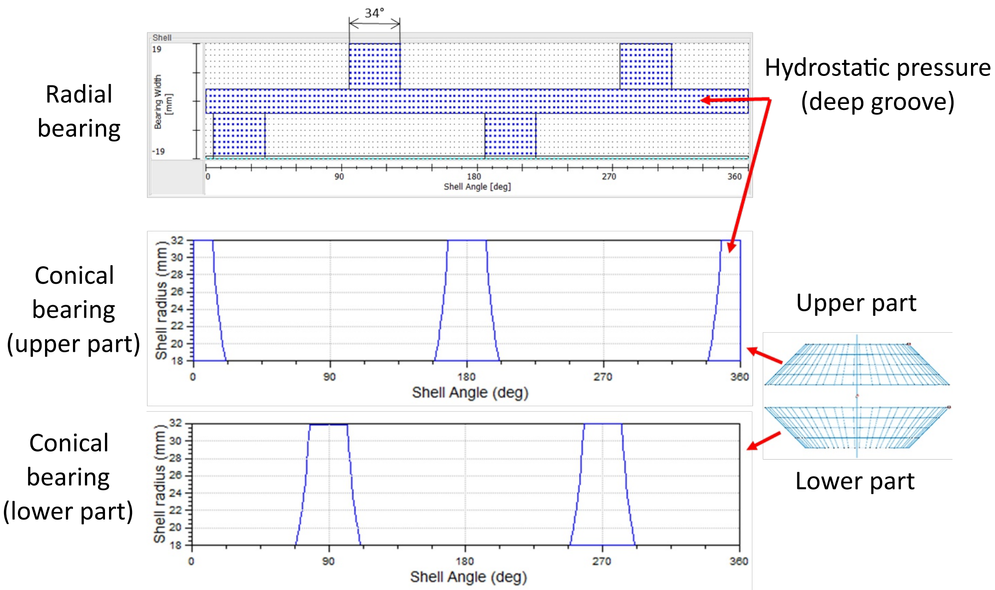

Fig. 2. Boundary conditions considered in the journal bearings of the main rotor

with [1]. Assuming that inertial terms dominate over viscosity terms, the resulting forcing is

$$F_{x,i} = -m_{\text{eff},i}\left(\ddot{x}_i - 2\,\omega_{\text{eff},i}\,\dot{z} + \omega^2_{\text{eff},i}\,x_i\right), \tag{7}$$

$$F_{z,i} = -m_{\text{eff},i}\left(\ddot{z}_i - 2\,\omega_{\text{eff}}\,\dot{x}_i + \omega^2_{\text{eff},i}\,z_i\right), \tag{8}$$

where x, z are horizontal displacements of the i-th node, $m_{\text{eff},i}$ and $\omega_{\text{eff},i}$ are the effective mass and angular speed of PbLi in the vicinity of the i-th node.

Due to the inherent nonlinearity, the system of Eqs. (1)–(6) is solved numerically employing a backward differentiation formula with the variable time step $\Delta t \in \langle 5\cdot 10^{-6},\, 5\cdot 10^{-4}\rangle$ s. The response is simulated in the interval $t \in \langle 0,\, 3\rangle$ s.

Fig. 3 shows hydrodynamic pressure fields in the conical bearing. Apparently, only the upper half of the bearing is active due to the buoyancy force. The hydrodynamic pressure in the lower half is determined only by the boundary conditions. Fig. 3 also reveals that the pressure field is asymmetric. The asymmetry generates a pressure gradient sufficient for hydrodynamic lubrication and is caused by the relative misalignment of the main rotor to the stator.

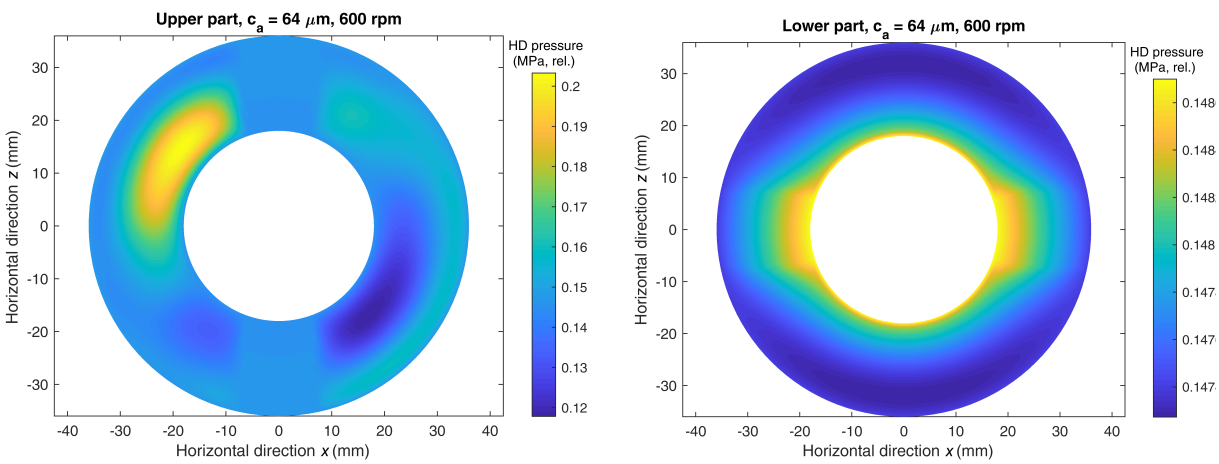

Fig. 3. Results – Hydrodynamic pressure fields in the conical bearing assuming axial (half) clearance $c_a = 64\ \mu\text{m}$ and speed $n = 600$ rpm

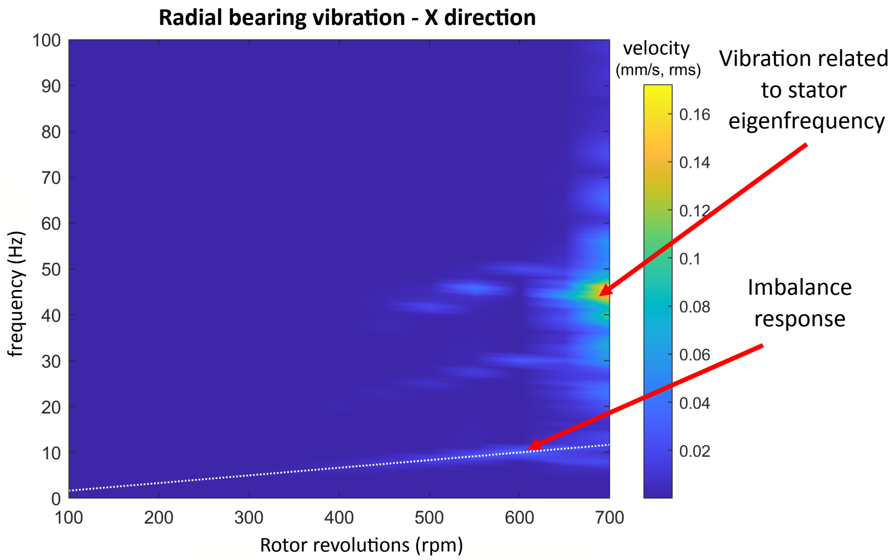

Fig. 4. Results – waterfall plot of absolute vibrations at the radial journal bearing

Fig. 4 depicts a waterfall plot of absolute vibrations of the stator at the radial journal bearing. The vibrations are dominated by the synchronous component caused by rotating unbalance up to 600 rpm. The stator experiences asynchronous vibration related to its natural frequency at speeds from 650 rpm. This asynchronous vibration is surprisingly associated with the conical bearing, which is very lightly loaded because the buoyancy force is almost equal to the gravitational force. The vibration can be attenuated by introducing an additional thrust load.

In conclusion, PbLi is suitable for journal-bearing lubrication, provided the sliding velocity is low to moderate. The low viscosity of PbLi guarantees acceptable power losses, but its high density can be problematic at higher sliding velocities due to innegligible inertial forces.

Acknowledgments

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