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A multiphysics finite element model for hydrostatic guideway systems

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Abstract
This paper presents a coupled elasto-hydraulic finite element framework for the multiphysics simulation of hydrostatic guideway systems in large machine tool rotary tables. The model simultaneously solves the linear elasticity equations for structural deformations and the Reynolds lubrication equation for thin fluid films, enforcing bidirectional coupling through a monolithic system of equations. The approach is applied to a five-metre diameter rotary table with a maximum load capacity of 100 tonnes and is benchmarked against an equivalent rigid-body assumption. Results demonstrate that structural flexibility significantly alters oil film thickness distribution, reduces global guideway stiffness, and lowers energy consumption, effects that are not captured by conventional rigid-body design tools.

Hydrostatic bearings; fluid-structure interaction; Reynolds equation; finite element method; machine tools

1. Introduction

Hydrostatic guideways are widely used in precision machine tools because of their high load-carrying capacity, near-zero friction, and excellent damping characteristics [1]. In the specific context of large rotary tables for heavy-duty machining, these advantages become critical: workpiece masses can exceed 100 tonnes and diameter requirements surpass several metres, placing stringent demands on guideway design [2].

The design of such systems traditionally relies on analytical or numerical models that treat structural components as rigid bodies. This assumption simplifies the problem considerably, but increasingly thin oil films mean that the elastic deformation of the table can become comparable in magnitude to the nominal film thickness itself as a consequence of the high loads. Under these conditions, the pressure distribution generated by the fluid directly influences structural deformation, and vice versa, creating a strongly coupled fluid-structure interaction (FSI) problem [3].

Coupled elasto-hydrodynamic models have been applied to journal bearings and thrust pads in rotating machinery, yet their systematic application to large-scale hydrostatic guideways in machine tools remains limited. Michalec et al. [1] provide a comprehensive review of large hydrostatic bearing design but note the lack of tools that incorporate FSI effects. Direct coupling finite element approaches have been demonstrated for simpler guideway geometries [3], and multiphysics models are established in aerostatic bearing design [4], but a validated framework targeting large rotary table architectures is still needed.

This paper presents an in-house finite element framework that couples structural and hydraulic physics in a monolithic formulation. The model is described in Section 2. Its application to a 5 m diameter proof-of-concept table is detailed in Section 3. Key results — oil film distribution, guideway stiffness, and power consumption — are discussed in Section 4, followed by conclusions in Section 5. Summing up, the challenge of the work is the application of FSI simulation in an industrial case where currently the analytical rigid body calculations are the standard.

2. Coupled elasto-hydraulic formulation

2.1. Governing equations

The structural domain is governed by linear elasticity. Assuming quasi-static loading and small displacements, the equilibrium equation is:

$$\nabla \cdot \sigma + \mathrm{F} = 0 \tag{1}$$

where σ is the Cauchy stress tensor and F the body force vector.

The pressure field in the thin lubricant film is described by the Reynolds equation, derived from the Navier-Stokes equations under the assumptions of laminar, incompressible flow and negligible film inertia [5]:

$$\nabla_{xy} \cdot \left(\frac{h^3}{12\mu} \nabla_{xy} \cdot P \right) = \nabla_x \left(\frac{hU}{2} \right) + \frac{\partial h}{\partial t} \tag{2}$$

where h is the local film thickness, μ the dynamic viscosity, P the pressure and U the sliding velocity.

2.2. Monolithic coupling and FEM discretisation

The two physics are coupled bidirectionally: fluid pressure acts as a distributed load on the structural surfaces, while structural displacements modify the local film thickness h entering Eq. (2). This coupling is enforced simultaneously — rather than through sequential staggered iterations — yielding a monolithic system after finite element discretisation:

$$\begin{bmatrix} [K_S] & [K_{SP}] \\ [K_{PS}] & [K_P] \end{bmatrix} \begin{bmatrix} u \\ p \end{bmatrix} = \begin{bmatrix} f \\ q \end{bmatrix} \tag{3}$$

Here Ks is the structural stiffness matrix, Kp the hydraulic stiffness matrix, Ksp and Kps are the off-diagonal coupling matrices, u and p are the nodal displacement and pressure vectors, and f and q are the applied force and flow-rate vectors, respectively.

Because Kps and Kp depend nonlinearly on the film thickness h, which is itself a function of u, the system is solved iteratively. Fluid viscosity is additionally updated as a function of temperature at each iteration, introducing a further source of nonlinearity. Convergence is assessed on both displacement and pressure increments.

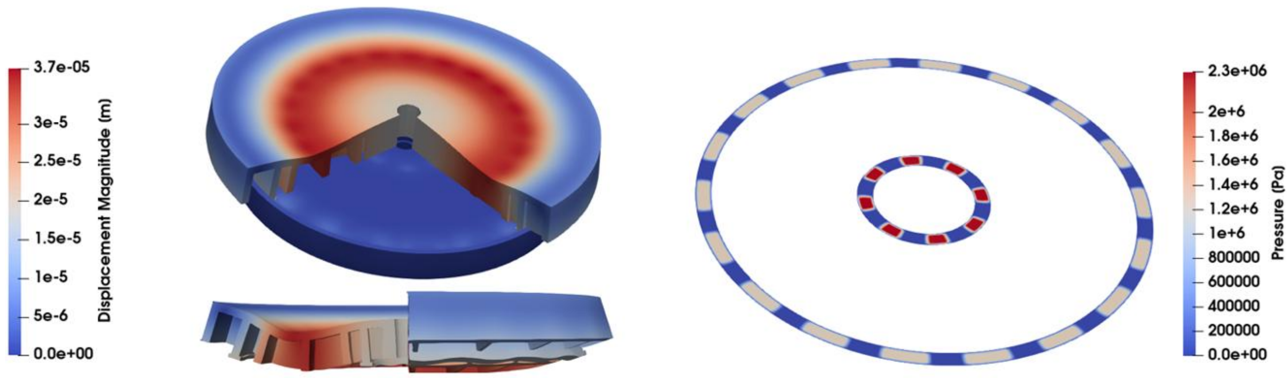

Figure 1. Displacement and pressure fields of the rotary table and fluid film, respectively.

3. Application to a large rotary table

The framework is applied to a 5 m diameter rotary table designed for heavy-duty machine tools, with a nominal load capacity of 100 tonnes and a supply pressure of 30 bar regulated through capillary restrictors. The table architecture consists of radial and circumferential ribs supporting two concentric hydrostatic rings, each equipped with multiple bearing pads. The structural material is FG-30 grey cast iron. This design is selected as representative case of heavy machin tool application.

The base structure is constrained at discrete mounting points, and the radial bearing is modelled by restraining in-plane rigid-body degrees of freedom. Simulations are performed across the full admissible load range (25–125 tonnes). To quantify the importance of structural flexibility, each load case is also run with the solid stiffness artificially increased by a factor of 10^4, producing a quasi-rigid reference solution equivalent to conventional rigid-body design tools.

4. Results and discussion

First at all, mention that the simulation tool and solvers where validated by simplified models and analytical solutions. The final model shown in Figure 1 has 348.072 DoF, it's solved in a i9-12950HX (12ª gen) with 16 cores and 24 threads with an average iteration time of 9,5 minutes.

The coupling mechanism is shown in Figure 1, which depicts the deformation of the solid bodies as a result of the pressure of the oil film. The difference between the inner and outer ring pressures is induced by the deformation of the rotary table, which creates very different film thicknesses.

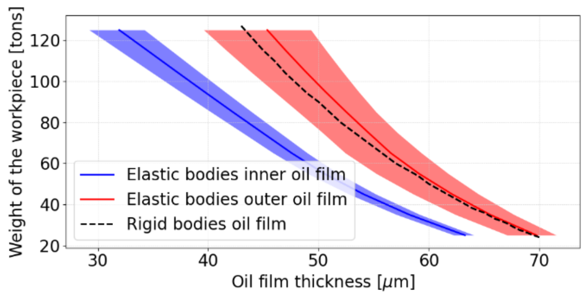

Figure 2. Load-gap curve for elastic body and rigid body rotary tables.

Across the studied load range, the mean oil film thickness decreases monotonically with load for both elastic and rigid models, as depicted in Figure 2. However, the divergence between the two solutions grows with load: the elastic table exhibits a lower mean film height in the inner ring and a higher one in the outer ring compared to the rigid case. This also means that the actual stiffness of the table is lower than in the rigid case. Regarding the energy consumption, Figure 3 shows that the pumping power is also different when coupling mechanism is considered, with lower energy consumption compared to the ideal rigid models.

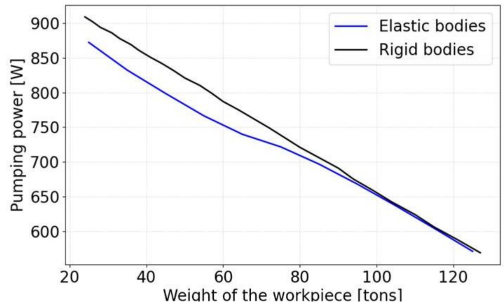

Figure 3. Pumping power as a function of the workpiece load.

5. Conclusions

A monolithically coupled elasto-hydraulic finite element framework has been presented and applied to a large rotary table with hydrostatic guideways. The key findings are:

– Structural deformation produces non-uniform oil film distributions within and between bearing rings.

– Global guideway stiffness is overestimated by rigid-body models because the differential film response of the two bearing rings reduces the effective parallel stiffness.

– Energy consumption may vary in coupled models compared to their rigid counterparts due to the gap variability.

These results demonstrate that structural flexibility must be accounted for in the design of large hydrostatic guideway systems. Future work will extend the model to dynamic (rotating) load cases and will incorporate thermal coupling to capture viscosity gradients across the film.

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